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Explanatory Note.

Illustrated articles are marked with an asterisk (*). Book notices are denoted by a dagger (†). The cross references condense the matter and assist the reader, but are not to be regarded as conclusive. So, if there were a reference from "Boiler" to "Furnace" and if the searcher failed to find the required article under the latter word, he should look through the "Boiler" entries, or others that the topic might suggest, as he would have done had there been no cross reference. A reference from "Oil" to "Lubricating" would apply equally to "Lubrication," "Lubricator," etc. Letters are indexed under title or subject, general articles under writer's name as well. Not all articles relating to a given topic necessarily appear under the same entries.

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POWER

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Fixed Prices

IN the good old days you bought your suit of clothes for the lowest price you could induce the merchant to accept. If you were shrewder than he you came out ahead. It was the universal custom to wrangle and haggle over practically everything you bought.

One day a retail merchant came along who had the courage of his convictions. He set a definite value on every suit he had for sale and marked the price in plain figures. If you found a suit you liked and the price fitted your pocketbook, you bought. If not, there was the end of it.

People liked his plan; it saved time and it bred confidence. This man was successful and it was not long before all his competitors were following his lead. Today, the fixed-price plan is practically universal in many lines of trade. Automobiles are sold, not for what they will bring, but for a definite, predetermined price based upon the cost of manufacture, the quantity manufactured and the cost of selling.

The fixed price is not extensively used in the sale of many pieces of power-plant apparatus, although it could be used to advantage both for the manufacturer and the purchaser. Engines, turbines, dynamos, motors, boilers, pumps, stokers, and other similarly well standardized pieces of apparatus, all could be sold for a fixed price. There are few good reasons indeed why the manufacturers of such apparatus should not publish a set of net prices just the same as the clothing man, the furniture man, et al.

We know of dozens of cases where the engine man, the boiler man or the stoker man, as the case might be, has deliberately added 20 or 25 per cent. to the price he hoped to get, simply for the purpose of cutting it off again at the vigorous urging of the purchaser, thus permitting the latter to feel that he had done a fine stroke of business and saved himself some money. What a piece of monkey-shine for male adults to be indulging in!

The fault for such a state of affairs lies with both the manufacturer and the purchaser, although neither likes it over much. The former is reluctant to put in his best bid first for fear that if he does so, and by some lucky chance he has not the usual brisk competition, he may have to take less for his product than he could have obtained. The latter is so accustomed to find that he can play one bid against another to his own apparent advantage that he has never stopped to reason out the true state of affairs.

We predict that some manufacturer, growing tired of the present barbarous selling methods, will come out with a fixed-price policy. And we further predict that he will be so successful that others will rapidly fall in line.

Limiting Efficiencies of Steam Plants

By Prof. R. C. H. Heck

The influence of feed-water temperature upon absolute efficiency and calculations showing that the best efficiency attainable is that based on the steam consumption of the main engine with feed water at the temperature of its exhaust.

The low thermal expenditure of operating steam pumps is shown when their exhaust can be used to heat the feed water.

In every heat-engine plant there are two efficiencies to be considered—one absolute, the other relative. Absolute efficiency is the ratio of work output to heat input. It has two values in every case, first, the actual or realized value; second, the ideal. The latter represents the limit of the attainable (never reached), or the performance of an ideal heat engine under the controlling conditions. Dividing the actual by the ideal performance, gives the relative efficiency, sometimes called the efficiency ratio. These general ideas may well be illustrated by an example.

Example 1. Suppose that a condensing plant consumes 12.5 pounds of dry saturated steam per horsepower-hour at 165 pounds absolute pressure corresponding to 366.1 degrees Fahrenheit, from feed water at 149.1 degrees. From the steam tables, the total heat of one pound of dry steam at 165 pounds absolute is found to be 1195.4 B.t.u., while the heat in one pound of water at 149.1 degrees is 116.9 B.t.u.—both these quantities being measured above 32 degrees. Then the heat of formation, or that received from the fire per pound of steam is

$$1195.4 - 116.9 = 1078.5 \ B.t.u.$$

As to output, one horsepower for one hour equals

$$60 \times 33{,}000 = 1{,}980{,}000 \ foot\text{-}pounds$$

of work done. Reducing this to heat units by dividing by 778, the amount of heat converted into work is 2545 B.t.u. This work being credited to 12.5 pounds of steam, the output per pound is

$$2545 \div 12.5 = 203.6 \ B.t.u.$$

Then the absolute thermodynamic efficiency of the plant is

$$E = \frac{203.6}{1078.5} = 0.1888$$

or 18.88 per cent.

To obtain the ideal efficiency, the pressure or temperature of the exhaust steam must be known. Let this temperature be 90 degrees, corresponding to a pressure of 0.696 pound absolute, or to a vacuum of 28.50 inches when the barometer is at 29.92 inches. With heat received from the fire after the general manner of the ordinary boiler, or with the Rankine cycle of ideal steam-engine operation, the output would be 334.6 B.t.u. per pound of steam. (This quantity is the difference between the total heat in one pound of the steam at 165 pounds absolute and the heat after adiabatic expansion to 0.696 pound absolute.) If the ideal output, or the best that could be done under perfect conditions of working, is 334.6 B.t.u. per pound of steam, while the actual output is 203.6 B.t.u., the relative efficiency will be

$$E_r = \frac{203.6}{334.6} = 0.608$$

or 60.8 per cent.

It will be noted that only that part of the plant is considered which has to do with the use and application of heat, after it has been received from the fire. Boiler and furnace efficiency, in generating heat by combustion and in transferring it to the working medium (water and steam), is another matter.

Limit of Absolute Efficiency

The present purpose is not to show how ideal performance and efficiency are derived by logical processes, but to keep much nearer the practical side of the problem and consider how actual efficiency is related to feed-water temperature, and how the latter is determined by the combined influences of the main engine and of auxiliaries. Assuming the steam consumption to be fixed by operating conditions which are outside of this discussion, the following proposition is to be established:

If the heat of steam formation be determined from feed water of the temperature of the main exhaust, and if the steam consumption of the main engine alone be used in calculating the output, the resulting thermodynamic efficiency is the highest of which the plant is capable.

The best that can be done with any combination of auxiliaries and feed-water heater is to approach this upper limit. As stated, the proposition meets the simpler case in which the main engine can return heat to the boiler only from its exhaust; a broader specification, covering such action as the return of hot water from steam jackets, will be dealt with later.

Plant with Steam-driven Pumps

Example 2. Continuing the controlling data of the first example, let the 12.5 pounds of steam per horsepower-hour be divided into 11.8 pounds consumed by the main engine and 0.7 pound by the feed and condenser pumps. Then the limiting efficiency as just defined would be calculated for a steam consumption of 11.8 pounds, from water at 90 degrees. The heat input per pound of steam is the total heat minus the heat in the feed water, or

$$1195.4 - 57.9 = 1137.5 \ B.t.u.$$

while the output is

$$2545 \div 11.8 = 215.7 \ B.t.u.$$

then the best efficiency is

$$E_{max} = \frac{215.7}{1137.5} = 0.1896$$

or 18.96 per cent.

Suppose, first, that the exhaust from the pumps is turned into the main condenser or otherwise wasted, and that water is fed to the boilers at a hotwell temperature of 90 degrees. Under this condition the whole 12.5 pounds of steam must be charged, cutting the output to 203.6 B.t.u., while the input remains 1137.5 B.t.u. The resulting efficiency then is only

$$E = \frac{203.6}{1137.5} = 0.1790$$

or 17.90 per cent.

Of course, no properly arranged plant would operate in such a fashion; instead, the exhaust from the auxiliaries would be used to heat the feed water. Suppose that these pumps consume 45 pounds of steam per horsepower-hour of their own development, or convert into work

$$2545 \div 45 = 56.5 \ B.t.u.$$

per pound of steam. Assume further that 25 B.t.u. are lost by radiation from each pound of steam passing through the pumps. Then the total heat carried by the pound of pump exhaust will be the original heat, 1195.4 B.t.u., less 56.5 turned into work and 25 lost, or

$$1195.4 - 81.5 = 1114 \ B.t.u.$$

Temporarily disregarding the need of removing cylinder oil, let 0.7 pound of this exhaust combine with 11.8 pounds of hotwell water at 90 degrees; the resulting water will have some temperature t, to be found by equating the heat from the steam with the heat absorbed by the water, or by the equation

$$0.7\,[1114 - (t - 32)] = 11.8\,(t - 90)$$

whence

$$12.5\,t = 1864, \text{ or } t = 149.1 \ degrees$$

This is the feed temperature which was used in the first example.

An instructive comparison can be made by considering the heat input per horsepower-hour under different conditions. Thus for the main engine alone, which uses 11.8 pounds of steam and is capable of returning feed water at 90 degrees, or

which requires 1137.5 B.t.u. per pound of steam, the heat received is

$$1137.5 \times 11.8 = 13{,}420 \ B.t.u.$$

For the combination of engine, pumps and feed-water heater, which uses 12.5 pounds of steam from water at 149.1 degrees, the corresponding heat supply is

$$1078.5 + 12.5 = 13{,}480 \ B.t.u.$$

From these numbers can be obtained the same efficiencies that have already been worked out with heat quantities per pound of steam, namely;

$$E_{max} = \frac{2545}{13{,}420} = 0.1896$$

and

$$E = \frac{2545}{13{,}480} = 0.1888$$

as in the first example.

The difference of 60 B.t.u. between the numbers 13,480 and 13,420 represents the thermal cost of operating the auxiliaries. It is about 0.7 of the 81.5 B.t.u. taken as the combined useful conversion and radiation loss of the steam. Comparing it with the supply 13,420, the percentage cost of the auxiliaries is only

$$60 \div 13{,}480 = 0.0045$$

or 0.45 per cent., as against an apparent steam cost of

$$0.7 \div 12.5 = 0.056$$

or 5.6 per cent.

In calculating these results, several minor sources of loss have not been taken into account. First of all, water from the condenser at full exhaust temperature—whether condensate or new water passed through a surface heater ahead of the condenser—is hardly attainable; there will probably be a deficiency of at least five degrees, and it may be much larger. An oil separator will remove from the pump exhaust considerable water, which will carry some heat with it. Radiation from the feed pump and feed piping will also cause a small loss.

Power-driven Pumps

The scheme which has been under discussion may now be compared with that of using power-driven pumps. These are either coupled to the main engine, as in many pumping-engine and marine plants, or are driven by motors as in some steam-electric plants. Whatever power they absorb is either directly or indirectly subtracted from the useful output of the unit. So far as the engine side of the plant is concerned, elimination of steam-driven pumps means the return of feed water at or below exhaust temperature. In the second example the first case, that of 11.8 pounds of steam per horsepower-hour, would correspond to the feed water at a maximum temperature of 90 degrees; but, as just stated, the efficiency of 0.1896 or a little less at the engine would be diminished to the extent of the power consumed by the pumps. The use of a live-steam heater or of an injector, either one giving the boiler the advantage of hot feed water, would not appreciably affect the thermodynamic relations, since both simply take heat from the boiler and put it back again. If an economizer be included in the plant, it would be considered a part of the boiler, sharing in the function of abstracting heat from the fire.

The contrasting points of the two types of arrangement may be summarized as follows:

With steam pumps, provided that they are not excessively wasteful, all their exhaust may be condensed by the main feed, and the heat rejected by the pumps may thus be returned into the main circuit. Consequently, the power developed and applied by these auxiliaries costs only its own thermal equivalent, plus secondary losses in oil removal and in radiation. If the latter could be reduced to zero, these minor members of the plant would have unit efficiency.

With power pumps, the percentage of power consumed will be much less than the percentage of steam used by the other type of auxiliaries; but since this power is produced at only the efficiency of the main engine, say from 17 to 22 per cent. in good practice, its thermal cost is much higher than in the other case.

The conditions assumed in the example correspond with a pump power equal to about 1.6 per cent. of that of the engine, as may be shown by the following example:

With independent steam-driven pumps, the pump output per engine horsepower requires 0.7 pound of steam, at the hourly rate of 45 pounds per pump horsepower. Using the power developed at the engine rate of about 12 pounds per hour, the work of the pumps would be done by

$$\frac{12}{45} \times 0.7 = 0.19 \ pound$$

of steam, which is 1.6 per cent. of 12 pounds.

One and six-tenths per cent. of 2545 B.t.u. is 40 B.t.u.; and with a thermodynamic efficiency of 0.19 for the engine, this would require a heat supply of more than 200 B.t.u. This is high compared with the 60 B.t.u. deduced in the second example for steam-driven pumps with complete utilization of the pump exhaust.

Feed Temperature

In short, what has been discussed is the influence of feed-water temperature upon thermodynamic efficiency. The point emphasized is the determinative influence of the main engine, or of the temperature which it alone (with its condenser) can give to the water delivered to the boiler. In the simpler cases the limiting or ideal feed temperature from the engine (never quite attained) is that of the exhaust steam. If the engine has steam jackets or reheaters in its receivers, and can therefore return hot jacket water, the ideal feed temperature is higher.

Example 3. With the engine using 11.8 pounds of steam per horsepower-hour, let 10 per cent. be condensed in the jackets and reheater, for simplicity at the full initial temperature of 366.1 degrees, the exhaust temperature remaining 90 degrees. What will be the ideal feed-water temperature due to the engine?

Solution. At 366.1 degrees the heat in one pound of water is 338.1 B.t.u. above 32 degrees, or

$$338.1 - 57.9 = 280.2 \ B.t.u.$$

above water at 90 degrees. The 1.18 pounds of jacket water, if there were no lowering of its temperature by radiation, would carry 330.6 B.t.u. above 90 degrees. This would raise the whole 11.8 pounds of feed water (including that from the jackets) through

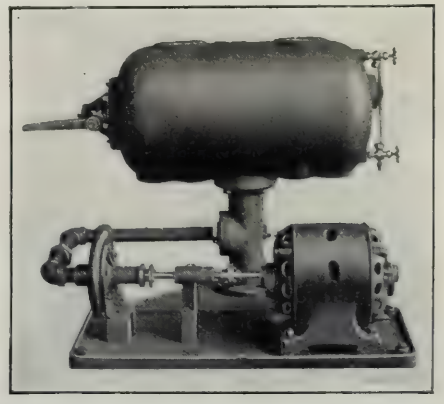
$$330.6 \div 11.8 = 28.1 \ degrees$$

or to 118.1 degrees Fahrenheit, which is therefore the ideal feed temperature sought. Diminishing the input by 28.1 B.t.u., or from 1137.5 to 1109.4 per pound of steam, while keeping the output at 215.7, the limiting efficiency is raised to

$$E_{max} = \frac{215.7}{1109.4} = 0.1945$$

Motor Driven Pump and Receiver

On the motor-driven centrifugal pump and receiver shown herewith the lever arm at the end of the cast-iron receiver is balanced on the inside of the receiver by a float and the position of the arm is changed with the rise or fall of the water



Pump and Receiver

level. The outside end of the arm is attached to the electrical device controlling the motor. When the water has risen in the receiver to a predetermined point the motor is started automatically and the water is pumped out. The set is manufactured by the Advance Pump & Compressor Co., Battle Creek, Mich.

Substituting Gas and Oil for Coal

By C. A. Tupper

The Fort Smith Light & Traction Co., which took over the properties of two competing local companies in Fort Smith, Ark., some years ago, offers an interesting example of systematic improvement in power-plant operation, particularly as the results obtained were secured without heavy additional investment. Among these changes the most interesting to the practical operating man is the use of natural gas under the boilers, displacing coal, and the installation of an auxiliary oil-burning system.

Furnaces

Each of the furnaces as now equipped has 15 burners of the Kirkwood type, consuming about 25 cu.ft. of gas per second when turned on full. The supply is brought to the furnace by 8-in. piping, from which 2-in. tubes branch off to the burners. Here the required heat is regulated to a certain extent by the length of the flame, but principally by the number of burners in service. Air mixture is effected by a rotating chamber, forming part of the burner, through which the gas passes, and by means of a suitable gate opening in the mixing chamber, the admission of air is regulated so that the gas burns with a bright blue flame. Provision has also been made for automatic regulation of the gas supply from the boilers.

Between and underneath each five gas burners is an oil spray nozzle. This system, taking about ¼ gal. of oil per horsepower, is utilized whenever the demand for the gas, which is largely consumed locally, causes the pressure to fall off, or anything else occurs tending to interrupt the supply. The oil burners are also of the Kirkwood type, manufactured by Tate, Jones & Co., of Pittsburg. Live steam from the boilers, reduced through a valve to 35 lb. pressure, is mixed with the oil in an annular chamber and from the latter the jet emerges through a flattened nozzle, giving a "fish-tail" effect to the flame.

Boilers

The five boilers are of the Aultman & Taylor type, the furnaces being fitted with brick checkerwork. In the boiler room only one head fireman and an attendant are now required for a 12-hour shift, whereas four or five times that number would be necessary if coal were used as fuel. Boiler-feed water is taken from the city mains as any supplied from the Arkansas River would have to be filtered, and this also dispenses with feed pumps. In the steam system a pressure of 125 lb. is maintained.

Interior of Boiler Room, Showing Gas and Oil Burners

Engine Room

In the engine room are installed non-condensing engine-driven units aggregating 5200 hp. These are divided between two-phase, 60-cycle, 2300-volt generators for commercial power and lighting service, and direct-current railway generators wound for 550 volts. There is also a 350-kw. motor-generator set, through which current for either system can be taken from the other.

The great gain effected by the introduction of gas and oil in place of coal has been not so much in the lower cost of the fuel as in the simplicity and convenience of operation, with greatly decreased attendance. There have also been eliminated the expense of handling coal and ashes, of which the cost of bringing the gas and oil to the furnaces is but a fraction. Furthermore, deterioration has been reduced to a minimum and there is a total absence of dust and smoke.

Oil System

The oil used in the plant is kept in a ventilated underground tank, surrounded by coils through which exhaust steam can be passed in winter when necessary to maintain the proper temperature. Duplex pumps are used for forcing the oil to the furnaces, with suitable regulation of pressure; and relief valves in the piping system, set at 45 lb., prevent an oversupply, with an automatic return to the tank, including all drips or overflow. The natural gas comes in directly from the field pipe line and enters through a reducing valve at 15 lb. pressure. Both the oil and gas are metered in transit.

In supplying and controlling the fuel care has been taken to avoid danger from fire or explosions, including a steam blanketing system and arrangements for quickly emptying the oil tanks in case of hazard.

Boiler Explosions at Sea

The danger of boiler explosion on board ship is not as serious as the same danger ashore, in power houses or factories and office buildings, says Capt. Hobson in the *Engineering Magazine*.

Due precautions, particularly in the efficiency of fire-room personnel, will reduce this danger to a minimum of only secondary importance, in most cases. But this cannot be said of the "Titanic"; her boilers were out close to the sides of the ship, with no inner skin to protect them. The boiler spaces extended for over a third of the ship and the bodies of the boilers themselves offered a target covering a considerable part of this space. A craft of comparatively small size, colliding perpendicularly or even obliquely in this space, would have easily reached a boiler.

In collision, the "Titanic" ran about as much danger from being blown up by a boiler explosion as from sinking from the inflow of water. No boiler located in the wings should be left without the protection of a partial longitudinal bulkhead, or at least an inner skin on the side. This risk on the "Titanic," though not great, was unnecessary. At very little cost and no difficulty, the inner skin could have been carried up the sides in the boiler spaces as far as the lower deck. In fact, it would have been good protection to take the same precaution in the engine-room space.

Combustion and Flue Gas Analysis

By Charles M. Rogers

Brief outline showing theoretical relation between percentage of CO_2 and boiler efficiency.

Some charts giving actual results of flue-gas analyses.

The predominating factor in the economical operation of a steam-generating station is combustion, and on the average, over 50 per cent. of the money spent for the production of power is consumed in supplying the boiler room. It would seem, then, that this is the end to "get after," if for no other reason than a monetary one. A prominent power superintendent told the writer not long ago that if he built any more power plants he would put the engineer's office in the boiler room.

The greatest loss in the operation of a steam boiler is made up of two factors, viz.: (1) the sensible heat of the escaping flue gases, and (2) the potential heat of the combustible constituents of these gases. This loss varies from 10 to 50 per cent. of the total heat contained in the fuel, and as a large percentage of this loss is avoidable, it would seem that too much study and attention could not be given to the subject of economical combustion.

Loss Due to Excess Air

It is readily seen that by increasing the air supply above the amount theoretically required for perfect combustion, the number of heat units carried up the stack by the waste gases also increases. The number of units lost in this way in the hot gases is determined by multiplying the difference between the temperatures of the stack gases and the air supplied to the furnace by the number of pounds of air supplied per pound of coal, plus the weight of combustible burned per pound of coal, times the specific heat of the gases. The loss varies, then, directly as the stack temperature. The excess air reduces the percentage of CO_2 in the gases, and makes it equal to the maximum possible percentage of CO_2, divided by the number of times the theoretically required amount of air is supplied, when no CO is present. This loss varies therefore inversely as the percentage of CO_2 in the gases.

The loss caused by the gases carrying off potential heat contained in the combustible constituents would naturally not depend upon the temperature of the gases. And the stack temperature alone can hardly be relied upon as indicative of high or low efficiency in that so many factors, such as dirty heating surfaces, air infiltration, etc., enter as functions of its rise and fall.

Since the sensible heat lost in the waste gases decreases as the percentage of CO_2 increases, then, other things being equal, a high percentage of CO_2 is conducive to high efficiency.

The curves in Fig. 1 were plotted from actual results obtained with a boiler equipped with a water meter, coal weigher and CO_2 recorder. They serve as a good illustration of how the boiler efficiency varies directly as the percentage of CO_2, and are but one case of many that show this relation distinctly.

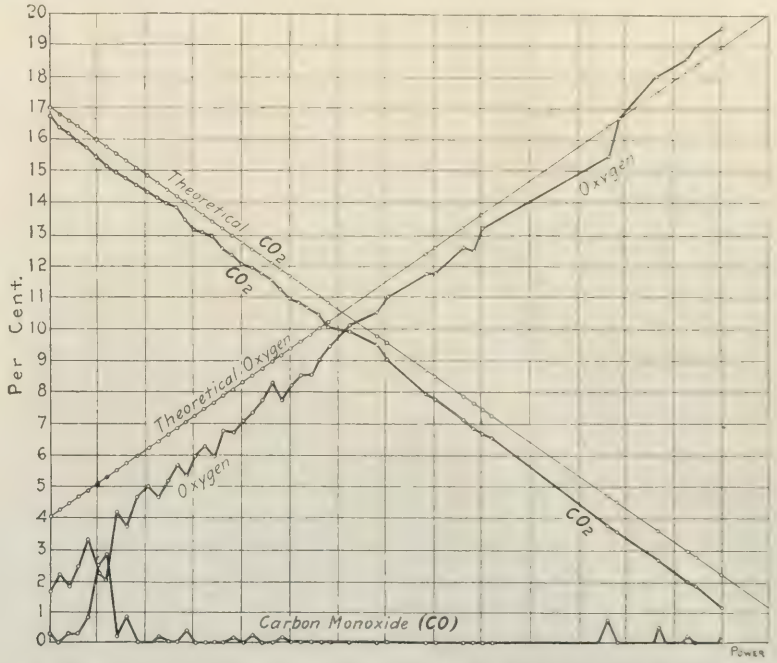


Fig. 2. Series of Flue-gas Analyses Showing Relation between Theoretical and Actual Percentages

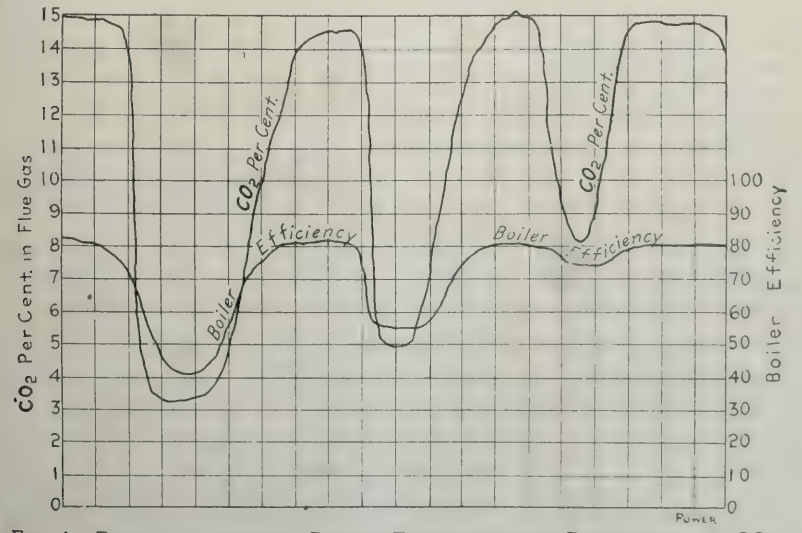


Fig. 1. Relation between Boiler Efficiency and Percentage of CO_2 in Flue Gases

Incomplete Combustion

It is well known that with a high percentage of CO_2 there is danger of an accompanying loss from incomplete combustion due to the formation of CO. This depends to a great extent upon the furnaces, fuel, draft and other general conditions. Apropos of this, the proper thing to do is to attain a comparatively high percentage of CO_2, which represents a gain that more than offsets any loss due to incomplete combustion.

Unfortunately, the writer has not been able to find any definite relation existing between the CO_2 or O and CO. The sum of these three constituents varies with the completeness of combustion, composition of the fuel, and to some ex-

tent with the temperature at which the gases are analyzed, as moisture is calculated as nitrogen and therefore brings the sum below 21. Since all the oxygen which combines with hydrogen forms water vapor, which is condensed, for a fuel which contains hydrogen, as does coal, the sum of the oxygen components will always be less than 21 per cent.

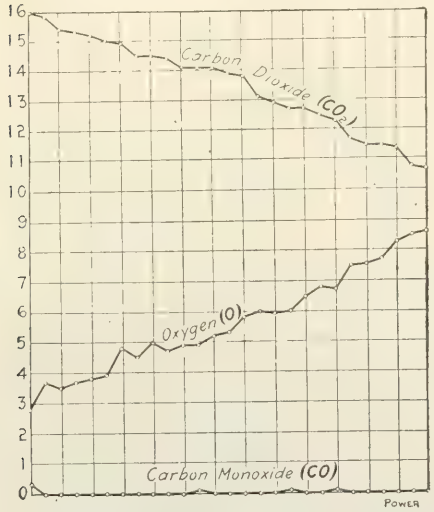


FIG. 3. TWENTY-SEVEN ANALYSES FROM 500-HP. BOILER

Thus, it will be noticed upon referring to Figs. 2, 3, 4 and 5, that by adding together the percentages of the oxygen components at the left of the chart, the sum rarely exceeds 19, while on the right in Fig. 2 it approaches 21, the percentage of oxygen in the air. This is what would be expected, as the readings on the right were taken while the fires were banked; the moisture, hydrogen and other constituents, which unite with part of the free oxygen of the air, having been expelled, and therefore leaving the total percentage of oxygen available for combining with the carbon to form CO_2 and CO. This checks up surprisingly close with the results of other tests not shown here.

Figs. 2 and 4 indicate that there is more danger of the formation of CO with the percentage of CO_2 above 13. These analyses were made with an improved type of Orsat hand analyzer. It is of interest to note that in Fig. 3, which gives the results obtained with furnaces in another plant but of the same make where the conditions are about the same, with the exception that the dampers and stoker engines are controlled automatically by the steam pressure, a trace of CO is found only four times out of 27 trials, even when the CO_2 ran as high as 15 and 16 per cent.

The results plotted in Fig. 4 were obtained from boilers where live steam was admitted through 6 jets distributed along the front of the furnace just above the coking grates. The results plotted in Fig. 5 are from the same boilers with the jets shut off. A study of these two charts shows that the possibility of CO formation is greater with the jets on when the CO_2 approaches the maximum; but while getting the normal percentage of CO_2, which averages about 12 per cent. with these boilers, there is little danger of the jets causing the formation of CO.

At the time each sample was taken, from which the results plotted in Fig. 3 were obtained, the conditions of the fire, draft and smoke were noted, and a comparison of these data reveals the fact that the CO (on the left-hand side) was found, during the emission of smoke only, when the fires were too heavy, or when the draft had been suddenly checked or green coal had been admitted in excess.

The straight lines in Fig. 2 give the theoretical percentages of CO_2 and O. That is, for any given percentage of CO_2 the theoretical O will be found on the

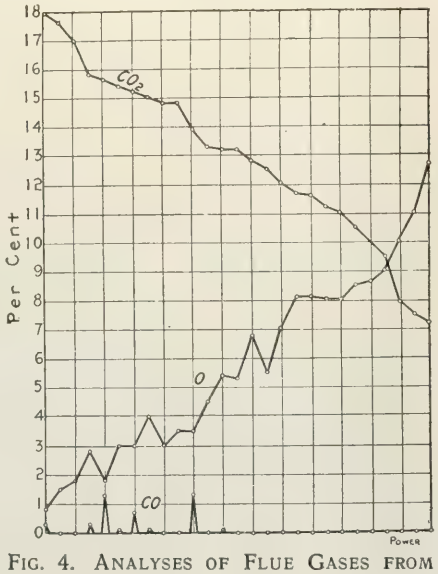


FIG. 4. ANALYSES OF FLUE GASES FROM BOILER USING STEAM JETS

same ordinate, and their sum will be equal to the percentage of O in the air.

LOSS DUE TO CO

It has been calculated that the average theoretical loss approximates 4.8 per cent. for every 1 per cent. of CO found in the flue gases. This and the following will illustrate the importance of guarding against the formation of CO.

The loss due to the formation of CO in reality amounts to considerably more than the foregoing theoretical figure indicates; as hydrogen and hydrocarbons usually escape unconsumed under the same conditions as does CO. On account of the analytical difficulty in determining the small quantities of these high heat-value combustibles in the flue gas, no practical examples are given which represent the exact losses due to their escape. The government, however, has found by several series of tests that the efficiency of a boiler drops 20 per cent. for every 1 per cent. of CO found in the waste gases. About three-quarters of this 20 per cent. loss is accounted for by the escape of combustibles other than CO. This figure is sufficiently high to be questioned by some, but it seems to be the case nevertheless, and furnishes an incentive to secure furnace operation free from the production of CO.

CONCLUSION

The foregoing tends to prove, then, that (1) the percentage of CO_2 in the escaping gases may be relied upon as an index of (a) high or low combined efficiency, and (b) the approximate percentage of heat carried off by the gases. (2) A high or low stack temperature is not always indicative of low or high efficiency. (3) A large part of the sensible and potential heat lost, ranging from 10 to 50 per cent. of the total heat contained in the coal, and in some cases even more, is avoidable. (4) No easily fixed relation exists between the percentages of CO_2 or O and CO, but knowing the percentage of CO_2, that of the O is easily determined approximately, or *vice versa*. (5) The occurrence of CO in the waste gases is a decided danger signal; not so much on account of the loss which it represents, but because it is usually accompanied by hydrocarbons and hydrogen, which are high in heat value, ranging from 21,000 to 62,000 B.t.u. per

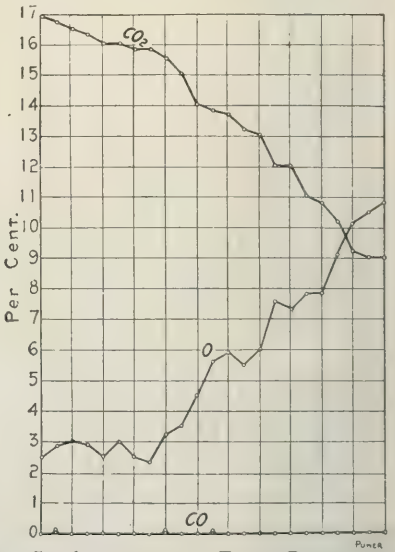


FIG. 5. ANALYSES OF FLUE GASES FROM SAME BOILER AS IN FIG. 4 WITHOUT STEAM JETS

pound. (6) Certain equipments and conditions, other things being equal, permit the regulating of a furnace so as to produce a comparatively high CO_2 without the formation of CO while others work adversely. (7) With a normal percentage of CO_2, the CO caused by steam jets is negligible. (8) When dense smoke is emitting from a stack, the loss is not indicated by the small percentage of precipitated carbon which makes the smoke visible, but by the loss of the unconsumed combustibles which usually accompany such conditions; a smokeless stack may be carrying away large amounts of high heat-value combustibles.

Heating Surface, Boiler Horsepower

By F. L. Johnson

Brief directions for computing the heating surface and rated horsepower of horizontal return-tubular boilers and instructions for using the tables which have been prepared for the purpose of materially shortening the operation.

At a recent engineers' examination in Ohio, this question was asked one of the candidates for a license: "What is the heating surface and horsepower of a horizontal return-tubular boiler 60 in. in diameter, 16 ft. long, having forty-four 4-in. tubes?" The heating surface of a boiler is that portion which is exposed to the heat of the furnace and to the hot products of combustion on one side and is covered by water on the other.

In the horizontal return-tubular boiler the water-covered surface exposed to heat is made up of the lower half of the shell, the inside area of all of the tubes and the area of one head less the area taken up by the tubes in both heads.

In these calculations one-half the area of the shell is taken because that part of the shell below the line where the brickwork is closed in and comes in contact with the shell and limits the height to which the fire or hot gases may rise closely approximates one-half of the circumference of the shell.

In calculating the heating surface of the tubes, the interior or fire surface is taken because the water will take away from the outside and greater area all of the heat that can be transmitted through the tube from the inside and smaller area.

The area of only one head is reckoned because the total surface of both heads which are exposed to heat is but little, if any, more than this. At the rear the setting is closed in just above the upper row of tubes, cutting off nearly one-half the area of the head, and at the front the surfaces below the tubes and above the water line transmit no heat to the water. This is practically equal to the area of one head. From this area must be subtracted the cross-sectional area of the tube ends in both heads as they present no surface whatever to absorb the heat of the passing gases.

Horsepower is a rate of work and in steam engines is measured by the foot-pounds of energy developed in a given time. In a steam boiler the energy developed cannot be measured or expressed in foot-pounds; for the work done is the conversion of water into steam and must be measured in terms of heat units.

In 1876, the committee of judges of the Centennial Exposition, in order to correctly compare the performance of many competing boilers, formulated what is known as the Centennial rule for the horsepower unit of steam boilers. This unit is the evaporation of 30 lb. of water per hour into steam at 70 lb. gage pressure from feed water at a temperature of 100 deg. F., an equivalent of the transfer of 33,305 heat units from the furnace to the water. This rule was accepted in 1884 by a committee of the American Society of Mechanical Engineers, which recommended its adoption by the society as a standard for use in all boiler trials.

As the value of the heating surface of a boiler depends on the quantity of heat generated in the furnace per square foot of heating surface of the boiler, the number of square feet required to develop a boiler horsepower will depend on the conditions of operation in each case.

From the earlier days of steam-boiler practice until comparatively recent times, the rates of combustion were moderate, and 15 sq.ft. of heating surface was the common allowance made for one boiler horsepower. By common consent, it was dropped to 12 sq.ft. for horizontal return-tubular boilers and 10 sq.ft. for water-tube boilers.

Some years ago, the New England Boiler Builders Association adopted 10 sq.ft. as the area required to develop one horsepower in horizontal return-tubular boilers.

Taking the boiler given in the question, the total area of the shell is found by multiplying the diameter by 3.1416 and by its length; 60 in. is 5 ft., hence the surface of one-half the shell is

$$\frac{5 \times 3.1416 \times 16}{2} = 125.664 \text{ sq.ft.}$$

The inside diameter of a standard 4-in. boiler tube is 3.74 in., which multiplied by 3.1416 gives the inside circumference

$$3.74 \times 3.1416 = 11.75 \text{ in.}$$

Hence each foot of length of a 4-in. boiler tube will contain

$$\frac{11.75 \times 12}{144} = 0.979 \text{ sq.ft.}$$

of heating surface, and in 44 tubes, each 16 ft. long, there will be

$$16 \times 44 \times 0.979 = 689.216 \text{ sq.ft.}$$

The area of a 60-in. (5-ft.) boiler head is

$$5 \times 5 \times 0.7854 = 19.635 \text{ sq.ft.}$$

The cross-sectional area of a 4-in. boiler tube is 12.566 sq.in., and in 88 tube ends there will be

$$\frac{88 \times 12.566}{144} = 7.679 \text{ sq.ft.}$$

Therefore, the heating surface of the boiler will be, in the shell,

$$\tfrac{1}{2} \times 5 \times 3.1416 \times 16 = 125.664 \text{ sq.ft.}$$

In the tubes

$$16 \times 44 \times 0.979 = 689.216 \text{ sq.ft.}$$

In the heads

$$19.635 - 7.679 = 11.956 \text{ sq.ft.}$$

adding

$$125.664 + 689.216 + 11.956 = 826.836 \text{ sq.ft.}$$

If 10 sq.ft. of surface are allowed per horsepower, the boiler will be rated as

$$\frac{826.836}{10} = 82.68 \text{ hp.}$$

TABLE 1. PRINCIPAL DIMENSIONS OF STANDARD BOILER TUBES FROM 2 TO 4 IN. OUTSIDE DIAMETER

External Diameter	Standard Thickness	Inside Diameter	Inside Circumference	Outside Circumference	Heating Surface per Foot of Length	Length per Square Foot of Heating Surface	Cross Section in Square Feet	
							Outside Diameter	Inside Diameter
2	0.095	1.810	5.6858	6.2832	0.4739	2.11	0.0218	0.017870
2¼	0.095	2.060	6.4717	6.7066	0.5393	1.86	0.0276	0.023112
2½	0.109	2.282	7.1691	7.8540	0.5974	1.67	0.0341	0.028402
2¾	0.109	2.532	7.9545	8.6394	0.6629	1.51	0.04120	0.034870
3	0.109	2.782	8.7399	9.4248	0.7283	1.37	0.04909	0.041451
3¼	0.120	3.010	9.4562	10.2102	0.7880	1.27	0.05761	0.049412
3½	0.120	3.260	10.2416	10.9950	0.8535	1.17	0.06681	0.05796
3¾	0.120	3.510	11.0270	11.7810	0.9189	1.08	0.0767	0.067195
4	0.134	3.732	11.7244	12.5660	0.9790	1.02	0.0872	0.075270

TABLE 2. AREAS TO BE DEDUCTED FROM AREA OF BOILER HEAD FOR ANY NUMBER OF TUBES FROM 2 TO 4 IN. DIAMETER

External Diameter of Tubes	1	2	3	4	5	6	7	8	9
2	0.0218	0.0436	0.0654	0.0872	0.1090	0.1308	0.1526	0.1744	0.1962
2¼	0.0276	0.0552	0.0828	0.1104	0.1380	0.1656	0.1932	0.2208	0.2484
2½	0.0311	0.0682	0.1023	0.1364	0.1705	0.2046	0.2387	0.2728	0.3069
2¾	0.0412	0.0824	0.1236	0.1648	0.2060	0.2472	0.2884	0.3296	0.3708
3	0.0491	0.0982	0.1473	0.1964	0.2455	0.2946	0.3437	0.3928	0.4419
3¼	0.0576	0.1152	0.1728	0.2304	0.2880	0.3456	0.4032	0.4608	0.5184
3½	0.0637	0.1274	0.2011	0.2548	0.3185	0.3822	0.4459	0.5096	0.5733
3¾	0.0767	0.1534	0.2301	0.3068	0.3835	0.4602	0.5369	0.6136	0.6903
4	0.0872	0.1744	0.2616	0.3488	0.4360	0.5232	0.6104	0.6976	0.7848

TABLE 3. HEATING SURFACE IN HALF SHELL OF CYLINDRICAL BOILERS

Length in Feet	Diameter of Shell, in Inches																							
	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	84			
14	76 96	80 63	84 29	87 96	91 64	95 30	98 95	102 63	106 29	109 96	113 62	117 28	120 95	124 62	128 28	131 94	135 61	139 28	142 94	142 94	150 27			
14.5	79 71	83 51	87 30	91 10	94 91	98 70	102 48	106 29	110 09	113 88	117 68	121 48	125 27	129 07	132 86	136 66	140 46	144 25	148 05	148 05	155 64			
15	82 46	86 39	90 31	94 24	98 18	102 10	106 02	109 96	113 88	117 81	121 73	125 66	129 58	133 52	137 44	141 37	145 30	149 23	153 15	153 15	161 00			
15.5	85 21	89 27	93 33	97 38	101 46	105 51	109 55	113 62	117 68	121 73	125 80	129 85	133 90	137 98	142 03	146 09	150 14	154 20	158 26	158 26	166 37			
16	87 96	92 15	96 33	100 53	104 73	108 91	113 09	117 29	121 47	125 66	129 85	134 04	138 23	142 42	146 61	150 79	154 99	159 17	163 36	163 36	171 71			
16.5	90 71	95 03	99 34	103 67	108 00	112 31	116 62	120 95	125 27	129 58	133 91	138 23	142 54	146 86	151 19	155 51	159 83	164 15	168 47	168 47	177 11			
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19	104 45	109 43	114 39	119 37	124 37	129 33	134 29	139 28	144 25	149 23	154 20	159 17	164 14	169 12	174 10	179 06	184 05	189 02	193 99	193 99	204 02			
20	109 94	115 19	120 42	125 66	130 90	136 14	141 36	146 61	151 84	157 08	162 32	167 55	172 78	178 02	183 26	188 48	193 73	198 97	204 20	204 20	214 67			
Area of head, sq.ft	9 62	10 56	11 54	12 57	13 64	14 75	15 90	17 10	18 35	19 63	20 97	22 34	23 76	25 22	26 73	28 27	29 87	31 50	33 18	34 91	38 49			

Tables 1, 2 and 3 have been prepared for the purpose of shortening the process of calculating the areas of heating surface. In Table 1 will be found all of the principal dimensions of standard boiler tubes from 2 to 4 in. in diameter. Table 3 gives the heating surface of one-half the shell and the area of the heads of boilers from 42 to 84 in. in diameter and from 14 to 20 ft. in length. Table 2 gives the areas to be deducted from the area of the boiler head for various sizes and numbers of tubes.

To illustrate the use of the tables, take the same example, Shell, 5x16 ft.; forty-four 4-in. tubes.

Opposite 16 and under 60 in Table 3, the number 125.66 is found. To find the total cross-sectional area of the eighty-eight 4-in. tube ends to be deducted from the area of one head, use Table 2. Under 8 and opposite 4, 0.6976 will be found. This is the area in square feet of eight 4-in. tube ends. By moving the decimal point one place to the right, the number is multiplied by 10 and becomes 6.976, which is the area of 80 tube ends, which added to the other number gives the total area desired.

$$0.6976 + 6.976 = 7.67 \; sq.ft.$$

Or the area of one tube end 0.0872 square may be multiplied by the number of tube ends.

$$88 \times 0.0872 = 7.6676$$

Adding the area as found in the tables, the total heating surface of a 5x16-ft. boiler having forty-four 4-in. tubes is found to be:

In one-half the shell, 125.66 sq.ft.
In forty-four 4-in. tubes, 689.21 sq.ft.
In the heads, 11.96 sq.ft.

making a total of

$$125.66 + 689.21 + 11.96 = 826.83 \; sq.ft.$$

Wrong Gage Connection

By Victor White

The following incident shows how needless delays are sometimes caused, occasioning great inconvenience, through a comparatively simple and easily avoidable mistake.

A certain power house was laid out and run on the plan of each engine having its own condensing plant, there being an alternative exhaust into the atmosphere through an automatic valve on each engine exhaust. An additional unit had just been installed and had passed through its preliminary slow- and full-speed runs on atmospheric exhaust in order to make sure that everything was working properly. Before putting the load on, it was decided to run the engine light with the condenser to see that this part of the equipment was in order. It should be explained that for the convenience of the attendant there was mounted on the wall at the back of each engine a 12-inch dial showing the vacuum; these were connected by means of copper tubes to each condenser in the basement.

Before throwing this particular set onto the condenser, a reading was taken on the vacuum gage on the wall, but, although circulating water was flowing freely through the condenser and everything to all appearances appeared to be in order, no vacuum was indicated. The engineers studied this problem for some time; the engine was shut down and the pipe leading from the condenser to the gage was taken apart and tested to see whether there was a leak or if it were choked anywhere, but everything seemed in good order.

The pressure gage was next taken off the wall and put directly on to the condenser itself, and when the engine was started and the air and circulating pumps attached to it were working, the gage immediately gave a reading of 28 inches. It was taken off again, put on the wall and the connecting pipe put back into position, when, as before, it registered nothing.

Three days were spent in trying to solve the mystery when finally a fitter's helper, hunting around underneath the engine, discovered that the plug and nipple attached to the connection between the condenser and the engine had been put in on the engine side of the condenser valve instead of next to the condenser. The result was, of course, that as the steam of the engine was passing to the atmosphere before the condenser valve was opened, and as the engineers had not ventured to open the condenser valve until they were sure everything was correct, the gage was simply registering the pressure of the exhaust steam to the atmosphere.

The gage pipe was disconnected, the hole in the exhaust pipe was plugged, a new hole was drilled and tapped next to the condenser, the pipe put into position, and then the readings on the gage were found to be satisfactory.

The Electric Light in 1878

In a report of the Bombay Gas Co., 34 years ago, says the *Scientific American*, a member who shortly before attended an exhibition of the electric light stated that this light was not one "which any man could put about his house and leave in the care of his servants. There could be no store of electricity, and the instant the engine ceased to work from any cause or accident all the lights would be gone."

Another attendant at the meeting "was quite ready to admit that at some distant day electric lighting might become general; but it might be that at some remote period men might fly. . . . All these things might be possible, but they were a long way off."

Testing an Engine under Difficulties

By H. B. Lange

A brief description of the methods followed to make sure that only the correct quantity of water should be charged to an engine which was being tested for steam consumption.

Often before an engine is paid for by the buyer a test is made to determine its efficiency and steam consumption. On such a test both the buyer and builder are represented. The builder in submitting his proposal guarantees that the engine will consume not more than a certain amount of steam per indicated horsepower or kilowatt-hour with a given steam pressure at the throttle and with a certain back pressure or vacuum. It is, therefore, necessary that an economy test be made to determine whether the engine meets the guarantee, and it is essential that the conditions under which the test is run be satisfactory to both parties involved and that the data and measurements be accurate within reasonable limits.

It is generally thought that it is an easy matter to make such an economy test; that all that is necessary is to weigh or meter the feed water, indicate the engine and read the switchboard instruments, and possibly to account for the steam used by the feed pump in some simple way and make corrections in the feed-water calculations. The testing engineer, however, knows from experience that this is by no means the case; that the first requirement is that his measuring instruments, such as gages, water meters, scales, thermometers, indicator springs, electrical meters, etc., must be correct, or if they are not he must know to what extent they are out. Few, if any, instruments are 100 per cent. correct and therefore they must be proved and calibrated in connection with standard instruments which are known to be correct. This calibration of instruments, while a more or less difficult task, is the least of the engineer's troubles when it comes to making a commercial test. The special and local conditions connected with each test which give the greatest trouble and make the most work are never the same, yet they occur in as many different cases as there are plants or engines to be tested. The most serious difficulties that come up are due to the condition of the piping systems; they are generally well laid out and erected accordingly, but the layouts are usually made for the best operating conditions and do not provide in any way for testing, and when it comes to setting up the various measuring instruments for the test the greatest difficulties are met in this part of the plant. The problems that confront the engineer in this connection are in some cases difficult to solve, yet they must be met and overcome in some way. This is illustrated by a test in connection with the plant in a large office building which consisted essentially of three direct-connected units, three water-tube boilers and two direct-acting duplex feed pumps. The boilers and engines were placed in the relative positions shown in the diagram. Engines Nos. 1, 2 and 3 were connected to boilers Nos. 1, 2 and 3 by the loop header *ABCD* with valves and drips as indicated.

Engine No. 3 was to be tested, and the first problem was to devise a means to accurately weigh the feed water used for steam in the engine alone. To accomplish this, boiler No. 2 was cut off entirely and boiler No. 1 was used to supply the engine. Boiler No. 3 was

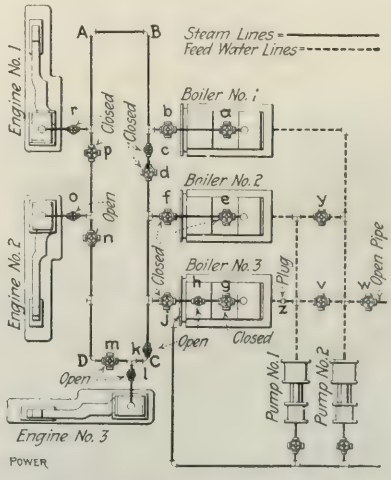


DIAGRAM OF THE ARRANGEMENT OF THE EQUIPMENT

used to supply the feed pumps and sump-pit ejector and to supply any other demand for steam. Engines Nos. 1 and 2 were shut down. All this was possible because the test was run on Sunday when the building was only partly occupied. Under these conditions it was necessary to isolate boiler No. 1 from the others by closing valves *g, j, e, f* and *p*. If it had been justifiable to assume that all the closed valves were not leaking, then it could also have been assumed that all steam generated in boiler No. 1 would have gone directly to the engine and that no steam would have gone to the header. Inasmuch as all the steam supplied to any part of the header came from boiler No. 1 and as this boiler was being fed with weighed water which was intended for the engine, it follows that if any steam had passed through leaky valves and then condensed, this weight of steam would have been charged to the engine and would have indicated a larger steam consumption than actually occurred. The problem therefore was whether any steam went astray, and, if so, how much. All doubt would have been removed if valves *j, d* and *p* could have been blanked.

The valves *n* and *m* were left open and the drips *k, l* and *o* were open and all condensation was caught in pails and returned to the feed-water barrel without being reweighed. That any condensation could pass through the valves *f* and *e* was out of the question, as whatever condensation there was would follow the line of least resistance and escape at the drip *k*. If any steam escaped through the valves *j* or *g* it would pass through the open drip *h*, but there was no leakage at this point. In this way the steam connections were proved to be tight and the condensation that did occur was reclaimed and accounted for.

The remaining observations and data were obtained without difficulty; diagrams were taken at frequent intervals. The back pressure, which was about ¾ lb., was measured by a U-tube connected to the exhaust line. The load was furnished by lights in the building and by a rheostat.

The next difficulty, after the main steam connections were taken care of, was the steam connection to the feed pumps. Under working conditions steam for the pumps was taken directly from the header, but during the test it was necessary to cut this off and a direct connection from boiler No. 3 had to be made. This was done by making a connection to the top of the water column. When steam passed out this way it was found that water was carried along from the lower part of the gage-glass and the pump cylinders knocked so hard that it was necessary to stop the pump at times. In order to overcome this the cock on the lower pipe to the water column was shut and the water level guessed at. As very little steam was drawn from this boiler this was permissible. However, when it was desired to know the water level the pump was shut down and the lower cock opened.

Thus all the difficulties in connection with the steam lines were overcome, but there was just as much question in regard to the feed-water connections. The lines were connected substantially as shown in the diagram. Pump No. 2 was used to supply boiler No. 1 with

water while pump No. 1 fed boiler No. 3. The question was whether any of the weighed water passed through the valves *v* or *y* and to boiler No. 3, or whether any of the feed water from pump No. 1 escaped into boiler No. 1. No water escaped into the idle boiler No. 2. It then remained to prove that *v* and *y* were tight in both directions. To do this pump No. 1 was stopped and pump No. 2 fed boiler No. 1. The clean-out plug *z* was taken out and it was found that no water escaped there. Then the plug was replaced, and the pump stopped; pump No. 1 was then started and the valve *w* was opened. If there had been any leakage in the valves *v* and *y* water would have come out of the open pipe outside of the valve *w*, but this did not occur and in this way valves *v* and y were shown to be tight; all difficulties were overcome and the test was run off in two four-hour periods.

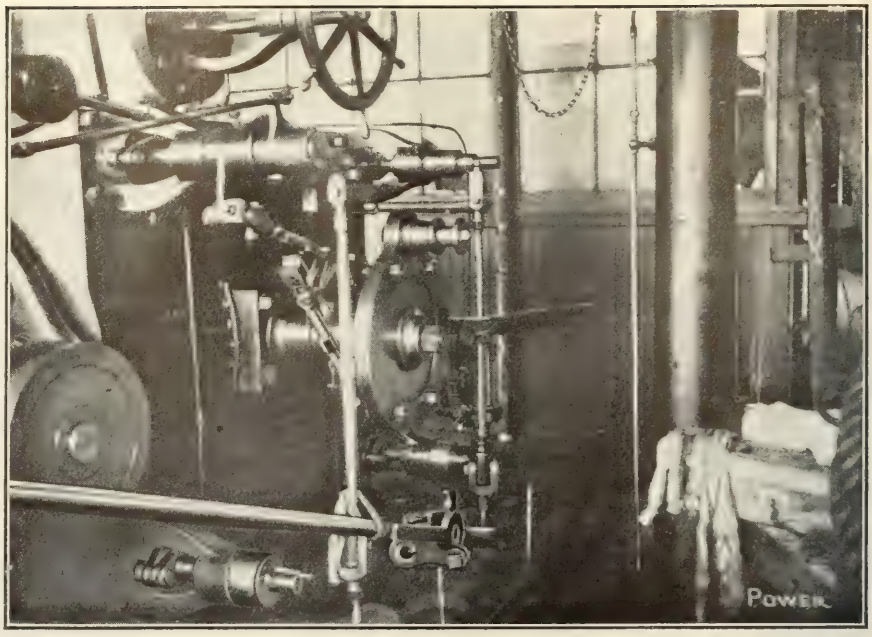


FIG. 2. SHOWING BROKEN VALVE GEAR

An Unusual Accident

BY H. E. OSGOOD

In shutting down a 32x42-in. single-eccentric Corliss engine running at 90 r.p.m. on 15 lb. pressure and 27 in. vacuum, water was drawn from the condenser into the cylinder which resulted in bringing the engine to a sudden stop. The arms of the flywheel were cracked, four at the hub and two at the rim, the main-bearing pedestal was demolished, the cylinder shifted ¾ in. on its base, the steam arm on the head end and the exhaust arm on the crank-end valves were broken, the piston rod was bent at the taper fit in the crosshead and the ¾x2½-in. key at the crosshead jammed ½ in. on the edge. The connecting-rod, crankpin and shaft sustained no damage, except for a light cut being necessary to smooth up the shaft at the main bearing, which was somewhat battered.

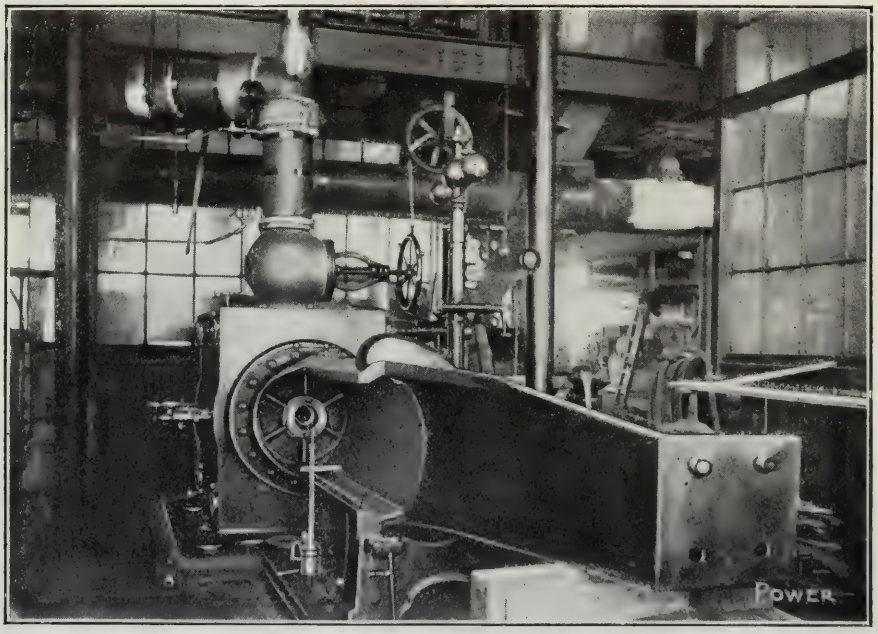


FIG. 1. ENGINE AFTER REMOVAL OF BROKEN PARTS

Fig. 1 shows the condition of the engine following the removal of the broken parts. The flywheel weighed 24,000 lb., was 16 ft. in diameter with a 36-in. face in two segments with four 2½-in. hub bolts and six 1½-in. rim bolts at each joint. There were six arms, each having 66 sq.in. of cross-sectional area at the hub and 45 sq.in. at the rim.

The accident resulted directly from water being drawn into the cylinder, before the engine had been brought to a stop. The steam was supposed to have been all but shut off, the clutch on the jackshaft was reported as having been thrown out, and, except for closing the exhaust valve, the engine was practically cut out of service.

Examination proved that the disk on the automatic check valve at the condenser had been removed some years

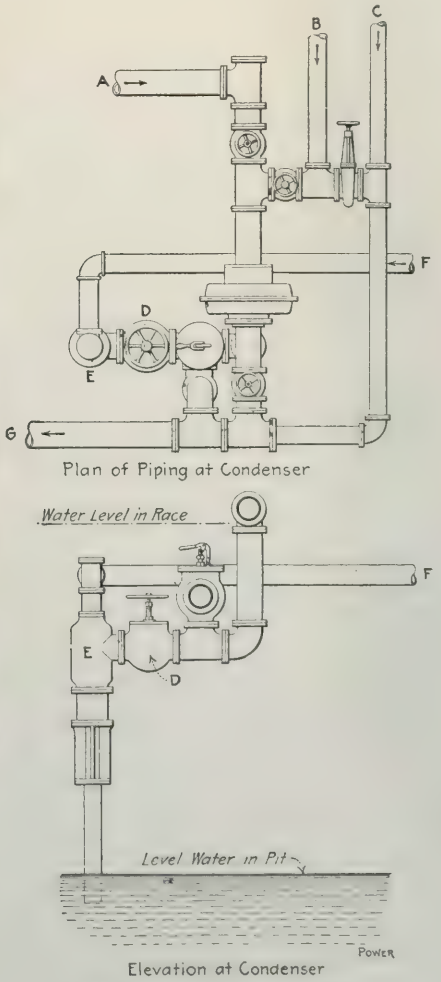


FIG. 3. CONDENSER PIPING

previous to the accident and never replaced; consequently, even if steam was entirely shut off, the momentum of the parts was sufficient to continue the motion, and there was nothing to prevent water from the condenser being drawn into the cylinder by the suction of the piston. The valves closed and the water thus entrapped brought the piston to a standstill, with the results as above described. Closing the main steam valve relieved the cylinder of any pressure to oppose an inrush of water. The water was drawn into the head end of the cylinder, as indicated by the compression of ½ in. of the key at the crosshead. It is surprising that the cylinder head was not broken.

Fig. 3 shows the exhaust-piping connections *A*, *B* and *C* from the 32x42-in. Corliss engine and Nos. 2 and 3 engines respectively and the location of the defective stop-check valve *D* and its relation to the condenser *E;* the water levels in the hotwell and also the race from which the injection water supply was taken through the pipe *F*. That both the race and hotwell are below the engine cylinder shows that the water was drawn into it by suction. The continuation of the exhaust line to the atmosphere is shown at *G*. The surplus water escaped while the exhaust valve was open on the return stroke, but the remainder formed a solid body resisting further motion of the piston.

Fig. 2 shows the valve mechanism, with the head-end steam-valve arm and the crank-end exhaust-valve arm, which were broken, removed. What caused these arms to break and the others not is an interesting point for speculation; the only likely supposition is that their valves were at the point of their most rapid movement, considering that the water was compressed in the head end, and the other valves were at a point of least motion, the momentum of the moving parts being sufficient to fracture the respective valve arms.

The Vibragraph

Vibrations in a power plant are always undesirable, but especially so when they occur in large urban power stations closely surrounded by other buildings. Although reciprocating engines are the chief offender, sometimes steam turbines also, owing to imperfect balance of the rotating parts, set up vibrations reaching to some distance.

The only instruments heretofore available for measuring vibration effect have been generally inconvenient to use, have left too much to the judgment of the observer, and have not been exact in tracing the source of the vibration.

Siemens Brothers & Co., London, have placed on the market the "vibragraph," which cuts out the personal equation and renders vibration visible in magnitude and direction. The vibragraph, Fig. 1, is a case inclosing the measuring apparatus proper. The vibrations are optically magnified so as to be clearly visible to the eye. A little mirror floating on a mercury drop and held against sidewise displacement throws a ray of light from

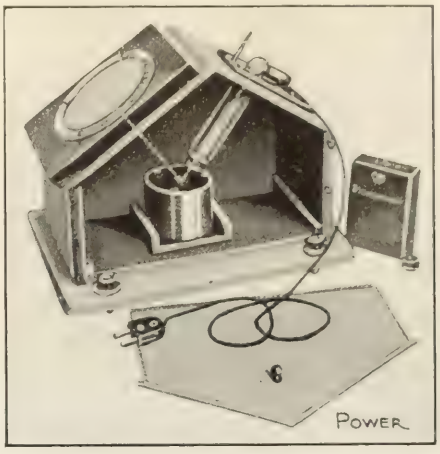


FIG. 1. THE VIBRAGRAPH

an incandescent lamp upon a sensitized paper or a ground glass. At complete rest the beam of light will form a point upon the paper, while a vibration in one plane will trace a straight line, and irregular and complicated vibrations curves or loops.

Each instrument has three measuring vessels of different sensitiveness, whose constants are laid out by the manufacturers in charts accompanying the apparatus so that a vibragraph leads to an

FIG. 2. VIBRAGRAM OF A TURBO-GENERATOR

easy determination of the actual magnitude of a vibration. Although the measuring vessels are very sensitive, they are so constructed as to need no special care in transportation. The apparatus ready for use weighs about 26 lb., and measures in its packing box about 14x10x8 in.

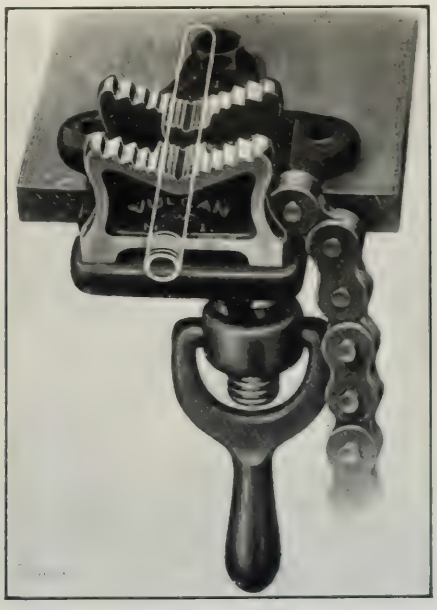
Fig. 2 shows the vibragram of a turbo-generator whose actual vibration is 0.005 in., and Fig. 3 the vibragram of a steam

FIG. 3. VIBRAGRAM OF A RECIPROCATING ENGINE

engine driving a 1250-kw. dynamo at 188 r.p.m., where the vibration amounts to 0.008 in.

Vulcan Chain Vise

Two new sizes have been added to the line of chain vises manufactured by J. H. Williams & Co., Brooklyn, N. Y. The larger size takes pipe ranging from ¼



VULCAN CHAIN VISE NO. 1

to 8 in. in diameter; the smaller has a pipe capacity of from ⅛ to 2 in. in diameter. The illustration shows the extended tooth which prevents bending or injuring the smallest pipe sizes when gripped in the vise.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Rotary Converters and Their Operation—II

Starting Rotary Converters

There are three ways of starting a rotary converter: by using a starting motor; by direct current, the converter starting as a shunt motor, and by alternating current, the converter starting as an induction motor.

The advantages claimed for the first are that heavy copper dampers can be used on the pole pieces, so proportioned that even under the most adverse conditions hunting is prevented. The small starting current required avoids any disturbance on the alternating-current system and when necessary all the converters can be started simultaneously.

When direct current is available the converter can be started as a direct-current shunt motor, and switched in on the alternating-current lines when it is in synchronism.

Rotary converters can also be started from the alternating-current side but this is not recommended except in special cases. Like the synchronous motor, it only develops a normal torque when in synchronism. To obtain the best operating performance the converter should have heavy copper dampers; but these interfere with starting on alternating-current without a starting motor, as it greatly increases the starting current. The power factor of the converter, while starting in this way, is low; and this, with the heavy current drawn from the line, causes flashing at the commutator. Also any such sudden demand for current will affect the voltage regulation of the supply circuit.

Where conditions likely to cause hunting are absent, self-starting will much less disturb the supply circuit if less effective dampers of smaller sections are used.

The starting motor is usually of the induction type having two poles less than the converter, and mounted on the armature shaft of the latter just outside of the bearing. This enables the motor to bring the converter up to and above synchronous speed and regulating devices are arranged for accurate synchronizing. Four-pole rotary converters usually have alternating-current, series-wound, commutator type starting motors, as there is too great a difference between its speed and the speed of a bipolar induction motor.

After a converter has been in service a short time the friction of the bearings will decrease, and the starting motor speed will have to be readjusted, or it will over-run the synchronous speed. Bearing friction also varies with changes of temperature, lubrication and other conditions impossible to accurately control. A resistance can be inserted between the alternating-current terminals of the converter, which will act as a load upon the starting motor, to check its speed and secure synchronism. When the converter is connected to the line this synchronizing rheostat must be out of circuit.

When a converter is started from the direct-current side or by a motor it must be brought into synchronism with the alternating-current supply before being connected to the line. Lamps are often used for synchronizing, but are objectionable because only a considerable difference of phase makes them glow and they do not indicate whether the converter speed is too fast or too slow. The ideal synchronizer should show the latter, the difference in speed, and the exact time of synchronism; these requirements are met by the synchroscope. The automatic synchronizer is an instrument designed to eliminate the uncertainty and trouble which occasionally arise when hand-thrown switches are used and insures that no machines will be thrown on the system unless conditions are right. When either a starting motor or direct-current starting is used, self-synchronizing can be obtained by closing the switches on the alternating-cur-

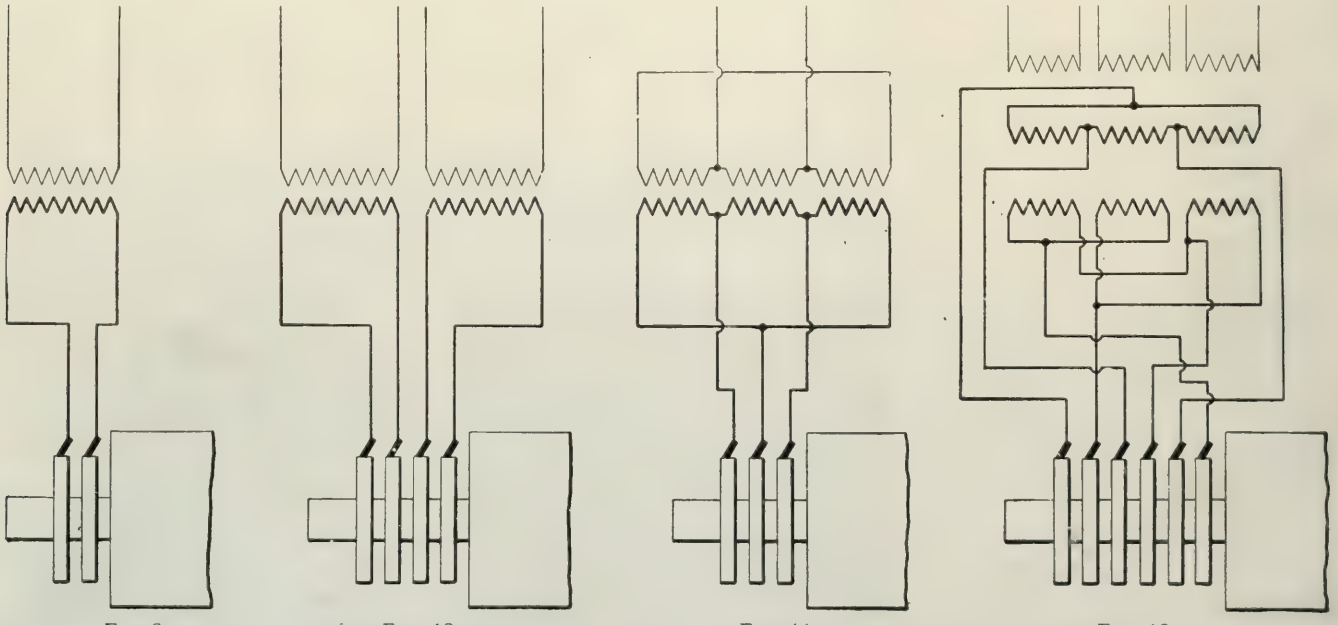


Fig. 9 Fig. 10 Fig. 11 Fig. 12

Transformer Connections for One-, Two-, Three- and Six-Phase Rotary Converters

rent side through proper reactance coils when the converter is above synchronous speed. The converter will lock in step as it passes through synchronous speed.

Parallel Operation of Rotary Converters

If several converters are to supply the same direct-current system they can be connected in parallel like shunt- or compound-wound direct-current generators. With compound-wound converters an equalizing connection must be used. The load will be divided between the different converters according to their direct-current voltages. The voltage can be regulated by changing the alternating-current voltage; or by regulating the field current, which changes the direct-current voltage by increasing the drop in voltage in the armature; or by shifting the direct-current brushes.

It is not advisable to have rotary converters, supplying the same direct-current system, connected on the alternating-current side; that is, supplied from the same transformer secondaries. This forms a closed local circuit in which heavy cross currents may be established and makes impossible adjusting the alternating-current voltage so as to compensate for the slightly different voltages of the several converters.

Rotary converters at different stations can be used to feed into one set of mains, if their voltage is properly adjusted. The direct-current sides can be handled almost exactly as independent direct-current generators, except that so wide a variation of voltage cannot be secured by adjusting the field current. Two rotary converters taking alternating current from two separate secondary circuits can be operated in series on a direct-current three-wire system. A single converter taking current from a pair of transformers can also be operated on a three-wire system, the neutral wire of the direct-current system being taken from the interconnected point of the transformer secondaries.

Oscillators and Overspeed Devices

The armature of a horizontal rotary converter will normally take a fixed position relative to the field frame and revolve without end movement. When running in one position the brushes are likely to wear grooves in the commutator and collector rings, and the bearings do not operate favorably. The oscillator is designed to produce a periodic longitudinal motion of the shaft, and may be either mechanical or magnetic, depending upon the source of the required impulse.

Where rotary converters may be operated at times from the direct-current side, running away due to a weakened field should be guarded against. This can be done by special individual exciters or a spring-closed knife switch operated by a centrifugal tripping device.

Inverted Rotary Converters

When driven by direct current and supplying alternating currents, the speed of a converter is not held constant by the alternating-current circuit, but, like a direct-current motor, is governed by the field strength and, therefore, by the armature reaction. If the alternating load is inductive, changes of load may cause wide variations in speed and frequency.

Inverted rotary converters have a special exciter, directly connected to the converter or driven by an induction motor receiving current from the alternating-current side. This exciter has an unsaturated magnetic field so that its voltage is considerably increased by a slight increase in speed. The resulting increase in the exciting voltage of the converter produces a stronger field and prevents the speed from being unduly increased.

Figs. 9 to 12 show the connections of substation transformers for single-, two-, three- and six-phase rotary converters respectively.

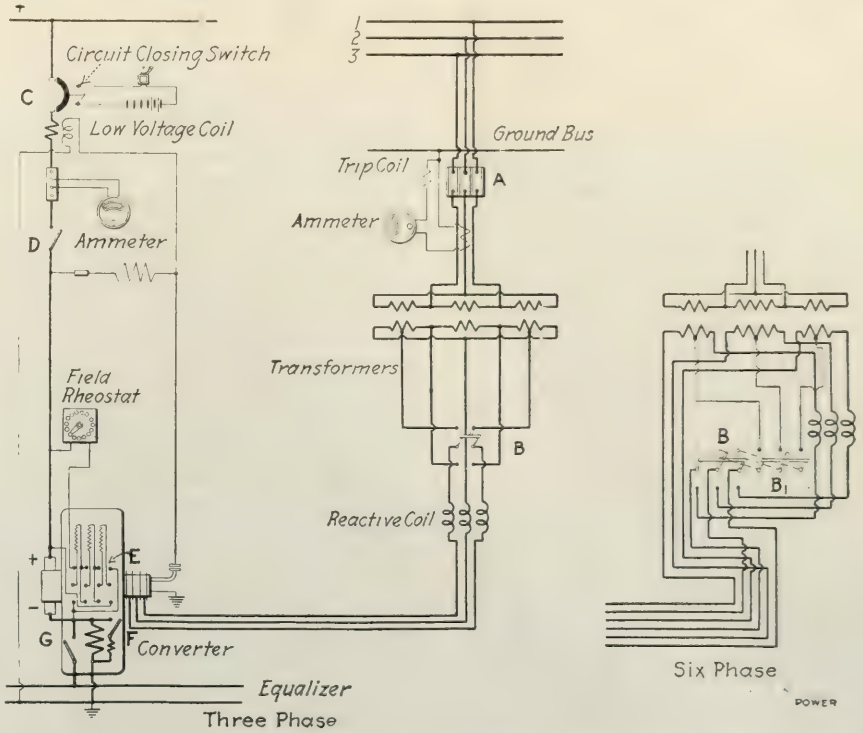


Fig. 13. Starting and Running Circuits for Three- and Six-phase Converters

Operating Hints

On heavy loads best results will be obtained with the brushes set at a slight lead from the neutral position, which can be determined by trial. To prevent sparking when starting from the alternating-current side, the brushes usually have to be shifted back to the neutral position. Occasionally a machine will not start from the alternating-current side if the voltage is low. This can often be overcome by shifting back the rocker-arm. Bolts should be occasionally tightened on the field winding connections, brush-holder studs and similar points. A good quality of oil should be used in the bearings and care should be taken to use a heavy oil on large machines. Direct-current brush springs should be set in the first notch if the commutators are in perfect condition; otherwise in the middle notch for ordinary loads. The speed-limiting devices should be tested at regular periods.

Using compressed air to blow out internal passages is strongly urged. Particular attention should be paid to the spaces under the collector rings; copper cuttings must not be allowed to accumulate in grease or oil around these parts.

Grinding or turning commutators should not be required if they are carefully attended to, unless heavily used or unless some fault develops such as a flat or high bar. In case the commutator condition becomes bad, grinding has certain advantages over turning; it removes less material, requires less skill and leaves the commutator in better condition.

In turning, care should be taken to block the shaft to prevent end play, and the speed should be as high as possible without burning the tool; this is usually not above 300 ft. per minute. Before turning the commutator, means should be provided to prevent chips or dust from working into the armature. After turning, the end of the commutator should be rounded off with a file to a radius of at least $\frac{1}{8}$ in. The commutator should then be examined bar to bar to see that no small particles of copper are em-

bedded in the mica. Under no circumstances should emery be used for grinding or polishing a commutator.

SWITCHING

The switch gear supplied with converters varies so much that complete instructions, covering all cases, cannot be given. The following applies to the usual connections and to the switchboard equipment of compound-wound converters in railway service. Referring to Fig. 13 a machine should be started as follows, assuming that all switches are open before starting and that the plug is inserted in the potential receptacle on the direct-current switchboard panel to connect the machine to the voltmeter:

Close the main high-tension switch *A*, then the starting switch *B* upward. For six-phase machines both switches *B* and B_1 should be closed upward. The machine will then run up to speed, which will be indicated by a cessation of the beats of the direct-current voltmeter. Close the equalizer switch *G* and the series-shunt switch *F*. Close the switch *E* into the top position. Throw the starting switch *B* quickly from the top to the bottom contacts. Adjust the direct-current voltage to approximately that of the busbars. Push up the low-voltage release of the circuit-breaker and close th circuit-breaker *C*. Close the main switch *D*. Adjust the division of the load between the machines, if more than one are in service, by means of the field rheostats.

To shut down a rotary converter, open the circuit-breaker *C*, pull out and turn the circuit-closing auxiliary switch to stop the ringing of the alarm bell; open the main switch *D;* open the high-tension oil switch *A;* allow the machine to retard until the voltage falls off to about 100 before opening the field switch *E* or the starting switch *B;* open the field switch *E*, equalizer switch *G*, series-shunt switch *F* and starting switch *B*.

If other machines are carrying load when a compound-wound converter is started, the correct polarity may be insured by closing the equalizer switch *G* when the machine locks in step. By watching the swings of the direct-current voltmeter as the machine approaches synchronism, switch *G* may be closed just previous to the last two or three swings, thus insuring proper locking on the first trial, if there is current for the series field from the other machines. If the machine locks with the wrong polarity as indicated by the direct-current voltmeter needle going down off the scale, the field switch *E* must be closed first into the down position, which will cause the voltmeter to return above zero, when the switch *E* must be pulled out and closed into the top position. The reversal of polarity should be made while the machine is running on the first starting tap.

Do not open the field switch *E* until the machine voltage has run down; otherwise the arc on opening the inductive field circuit may be blown by the armature blast from point to point, causing a short-circuit, or the strain may injure the insulation of the field winding.

Do not open the starting switches until the voltage has decreased to zero. This leaves the transformers without a heavy residual magnetism, which tends to cause an extra rush when they are subsequently thrown into circuit.

Do not open the oil switch with the converter on the starting tap and the field closed; otherwise the converter may build up to full voltage, tending to run up the transformers to considerable above normal voltage.

Do not parallel the machines on the direct-current side without closing the equalizing switches.

Do not start the converters with the field switch *E* closed either in up or down position or with the series-shunt switch closed.

Do not close the alternating-current starting switch slowly into the top contacts, for if this is done, the rush of current may pit the tips of the clips and thus prevent the switch blades from going in.

Do not close any switch slowly.

Do not close the circuit-breaker with the main switch *D* closed.

Do not close the circuit-breaker after a heavy short, without plugging the voltmeter to the converter to make sure that the polarity is not reversed.

LETTERS

Reversed Polarity

Our plant is equipped with two 125-volt, 720-ampere direct-current dynamos wired to be run in parallel. A few days ago the electrician stopped No. 2 and gave it a good cleaning, also sandpapering the armature and brushes. When ready to start it, he threw in the circuit-breaker and the three-pole switch, and told his helper to start the dynamo. The latter started to do so very carefully, but before the machine had attained full speed the electrician threw in the paralleling switches.

The circuit-breaker immediately flew out and the voltmeter and ammeter on No. 2 dynamo tended to go backward. We changed the meter wiring accordingly and they indicated all right.

We again tried to run the dynamos in parallel, when both circuit-breakers came out with a bang. We then decided that the polarity had been reversed and the question was how to change it back again. The master mechanic came to our aid and ran two wires from No. 1 machine to the field connections of No. 2, placing a cutout with 30-ampere fuses in circuit as a precaution. This changed the polarity back again. We then changed the meter wires back to their original connections, started the dynamo and threw in the paralleling switches and they ran all right.

JAMES MITCHELL.

North Adams, Mass.

Home Made Welding Device

Upon taking charge of our shop, I found a number of broken grids in the resistance for the crane controllers. The broken parts had been connected by splicing copper wire on each side, but this was very unsatisfactory, as they became loose and began to burn. I removed the broken parts and repaired them with a home-made electric welding device. This was made by connecting some No. 4 wire to the crane circuit, the positive wire to an old resistance, and the negative wire to the carbon holder. From the resistance the wire led to a plate upon which was placed the grid to be welded.

The carbon holder is made of ¾-in. pipe holding the wire on one end and two clips which hold ¾-in. carbon blocks on the other end; a handle was placed on the end to which the wire is connected.

Striking the arc at the brake and keeping it moving back and forth, it took but a few seconds for the metal to weld. Then feeding a little of the same metal in while moving the arc back and forth, I soon had the grid a little thicker at the fracture, repeating the same operation on both sides. It is now as good as new and has been in use several months. I protected my hands with a pair of gloves and my face with ⅛-in. ruby glass.

C. A. BLOMBERG.

Detroit, Mich.

What Made Transformers Heat Up?

We had occasion to disconnect one of our switchboard wattmeters, and left the secondaries of the potential and current transformers in open circuit. These transformers were operating on a three-phase 2300-volt circuit and I noticed after one night's operation that one of the current transformers became so hot that some of the insulating compound ran out.

I therefore put a jumper across the transformers and the next night that one ran cool but the other became hot again. I then did likewise with the second transformer and ran both with jumpers until the meter was back on the board. Why did the current transformers get hot while running with the secondaries open?

C. J. BRINDLEY.

Lorain, Ohio.

Before July 1, the National Electric Light Association hopes to have 12,000 members. There were 11,732 on May 15.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Small Gas Power Plant

By A. R. Maujer

Unless, for some special reason, the central station can and will sell current at or very close to actual cost, or unless the load factor is low, it is usually possible for the consumer to generate power for less than it would cost to purchase it. This is fairly demonstrated by the following description and operating data of the plant of Frank Holton & Co., 2638 to 2646 Gladys Ave., Chicago, Ill., makers of band instruments.

The factory is three stories high and covers a plot 86x100 ft. It is heated by a low-pressure gravity-return heating system, steam for which is generated in a small locomotive-type boiler.

The numerous lathes, drill presses, buffing wheels, etc., are driven in groups from motor-driven jackshafts. The motor load is about 40 hp. and in addition there is a lighting load consisting of about 200 incandescent lamps. To supply the original requirements a 15-kw. direct-current dynamo belted to a 25-hp. two-cylinder Nash gas engine was installed in 1907 (see Fig. 1). By 1911 the load had grown beyond the capacity of this unit and the equipment shown in Fig. 2 was installed. This consists of a Crocker-Wheeler 220-volt direct-current dynamo of 25 kw. rated capacity driven by a 40-hp. Nash gas engine.

The engine is of the two-cylinder type, operating on the four-stroke cycle, equipped with a throttling governor, and running at 275 r.p.m. Ignition current is supplied from the small storage battery seen on the side wall in Fig. 2. This battery is kept charged by a special igniter generator, mounted on the frame of the engine and driven by a belt from the main shaft. Illuminating gas is used as fuel.

The engine is started by compressed air at 180 lb. pressure, a small motor-driven single-cylinder vertical compressor being used to store air in two small steel tanks.

After passing through the cylinder jackets the cooling water is run through a transverse-current water heater made by the Williams Tool Co. which utilizes the heat of the engine exhaust to raise the temperature of some 200 gal. of water per hour from about 140 to 180 deg. F. All of this water is used for manufacturing purposes.

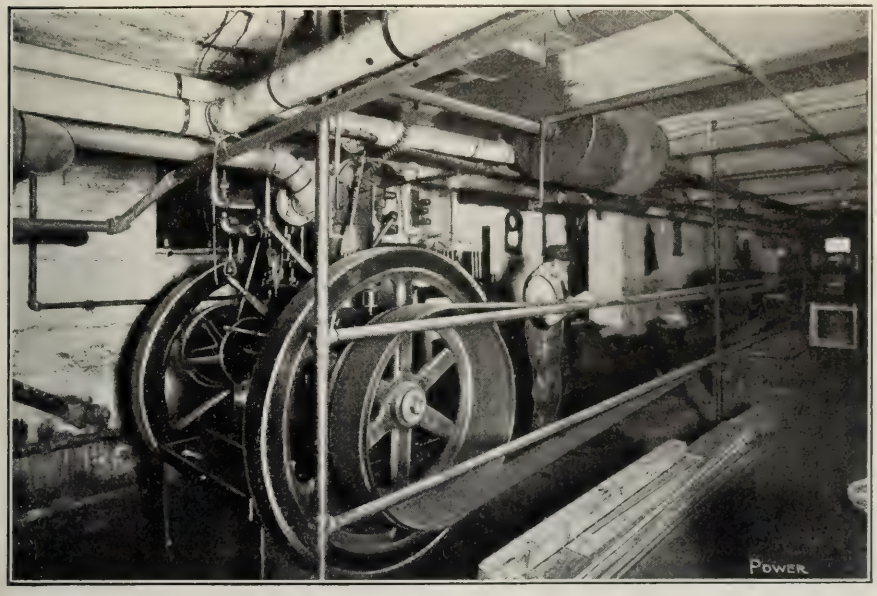


Fig. 1. Original Installation

Operating Costs

Table 1 gives the gas consumption and electrical output for November, 1911, which is a typical month. The engineer, in addition to looking after the generating equipment, runs the heating plant and takes care of all the motors, shafting, wiring, piping, plumbing, etc., throughout the building. Even if the generating apparatus were not installed his services or their equivalent would be required 75 per cent. of the time; hence, in estimating the cost of power, but 25 per cent. of his salary, which is $100 per month, can properly be included. As all of the cooling water is used throughout the factory, the cost of the water is not chargeable against the engine. In fact, this water would have to be heated by other means and hence the engine really should be credited with this item. The average consumption of lubricating oil is 0.9 gal. per day and costs 36c. per

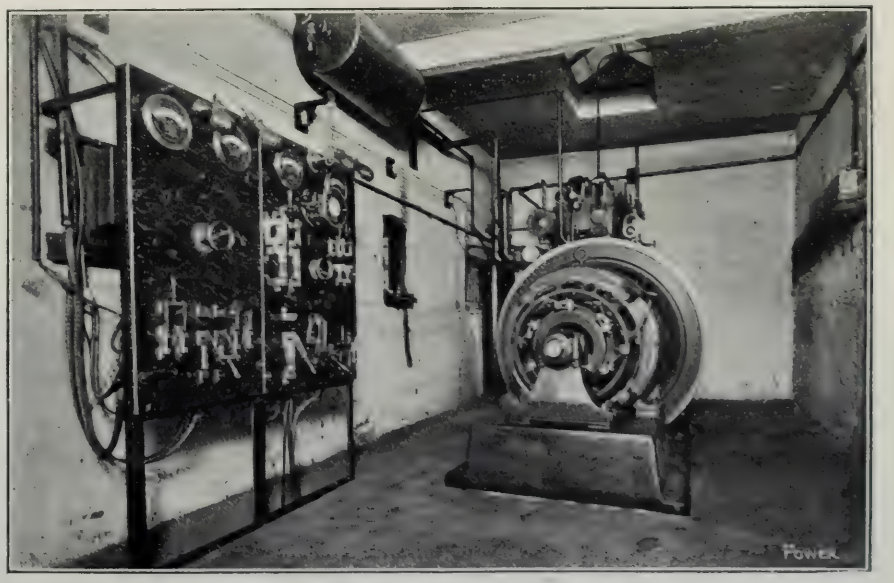


Fig. 2. 25-kw. Unit and Switchboard

gallon. At this writing, a year after installation, there have been no repair charges against the power equipment. The cost of the 25-kw. unit with all accessories installed complete was $3250 and that of the 15-kw. unit was $1775.

Based on the foregoing consideration and on the data given in Table 1, the following estimate of the cost of power for Nov. 1, 1911, is fairly accurate:

Fuel (121,100 cu.ft. of gas@80c. per 1000 cu.ft.)	$96.88
Labor (25 per cent. of engineer's time)	25.00
Oil	8.10
Interest, depreciation, etc. (12 per cent. per annum on $5025)	50.25
	$180.23

$$Cost\ per\ kw.\text{-}hr. = \frac{\$180.23}{4542} = 3.97c.$$

Cost of Heating

The heating does not affect the power cost one way or the other as it is exactly the same regardless of whether the power is generated in the building or purchased from some outside source. It may, however, be interesting to know what this cost is. During the winter of 1911-12 the heating plant consumed 54 tons of Pocahontas coal costing $3.75 per ton and 108 tons costing $4, making a total fuel cost for the heating plant of $634.50.

TABLE 1. DAILY FUEL CONSUMPTION AND ELECTRICAL OUTPUT FOR NOVEMBER, 1911

Date	Gas Consumed, Cu.Ft.	Output, Kw.-hr.	Fuel Cost per Kw.-hr., Cents
1	5,200	192	2.16
2	5,300	193	2.20
3	5,100	188	1.27
4	2,900	106	2.19
6	5,000	179	2.24
7	4,900	186	2.11
8	4,900	184	2.13
9	5,100	186	2.19
10	4,900	184	2.13
11	3,000	105	2.29
13	5,000	198	2.02
14	5,200	202	2.06
15	5,100	195	2.09
16	5,100	194	2.10
17	5,600	196	2.29
18	3,100	116	2.12
20	5,400	204	2.12
21	5,300	205	2.07
22	5,500	207	2.12
23	5,400	207	2.08
24	5,600	206	2.18
25	3,200	124	2.06
27	5,400	205	2.10
28	5,000	195	2.05
29	4,900	185	2.12
Total.	121,100	4542	2.13

Fuel Supply Troubles

By James H. Beattie

An engine, to give steady, uninterrupted service, must have constant fuel supply; furthermore, the fuel must be clean. Greater care is necessary perhaps to keep an engine using liquid fuel in constant operation than one using gas, as the former is more subject to trouble through dirt or water in the fuel, than an engine using natural or artificial gas as these are reasonably constant in their composition and not, as a rule, subject to interruption in flow. The following instances may be of interest as they are good examples of the kind of difficulties the average engine operator meets.

A small vertical engine suddenly developed a bad case of backfiring, misfiring and loss of power. The ignition system was in good condition and the spark properly timed. Opening the needle valve (the engine operated on gasoline) did not improve matters. Throttling the air pipe helped somewhat but did not effect a cure. A section of the cylinder and mixing valve of this engine is represented

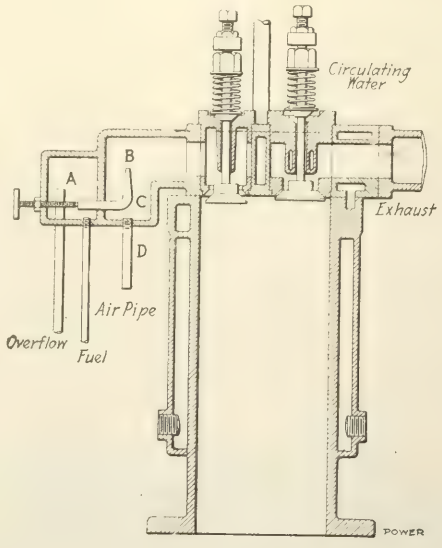


Fig. 1. Section through Cylinder

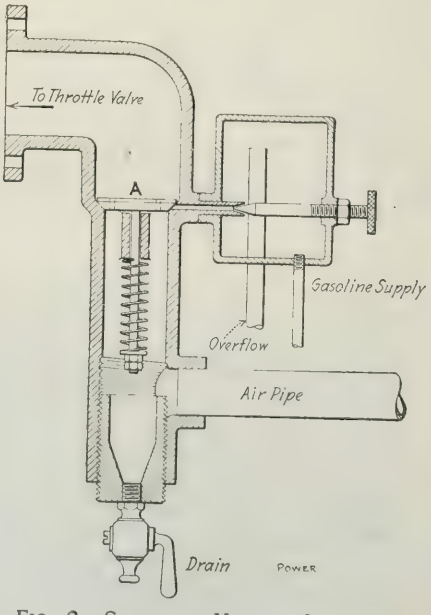


Fig. 2. Showing Mixing Chamber

in Fig. 1. The mixing valve of the common overflow type is indicated at *A*, the gasoline being kept at a constant level by the overflow pipe. On the suction stroke of the engine the gasoline is drawn through the small pipe *C*, the amount entering being determined by the setting of the needle valve. Dirt had entered the small nozzle and lodged near the end where the mixture of the fuel and air occurred. The pipe was not completely stopped up but very nearly so; hence the amount of fuel that could pass was regulated by the obstruction and not by the needle valve. Throttling the air supply increased the suction and thus more fuel could pass the obstruction in a given time. The whole trouble disappeared when the obstruction was removed. Lack of care in filling the fuel tank was responsible for this difficulty, which goes to show that too great care cannot be taken to keep the fuel free from dust, dirt or water.

Another engine operated very satisfactorily for several weeks after it was installed, and then began stopping frequently, for no apparent reason. It proved to be another case of fuel-supply failure, this time due to the fuel tank being too far from the engine and too far below the gasoline pump. While the pump was new and in perfect condition it brought the fuel to the engine without trouble, but when the packing began to wear the pump was not equal to the task. Pumping by hand before starting the engine would bring up enough fuel to keep the engine going for a short time, as the mixing valve held considerable, then the engine would stop again. The trouble was remedied by placing the fuel tank nearer the engine and at a higher level.

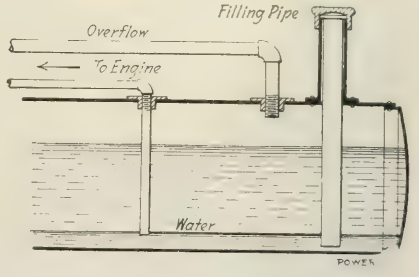


Fig. 3. Showing Water in Bottom of Tank

In another case a small engine was rapidly getting a bad reputation by using more fuel than necessary. It was rated at 15 hp. and consumed nearly 25 gal. of gasoline per day, although not operating at full load. The trouble proved to be looseness in the lock lever that holds the inlet valve shut when the engine is coasting. This caused the inlet valve to chatter (due to the partial vacuum in the cylinder when the engine was running idle) allowing fuel to enter and go to waste. The fuel consumption dropped to about 12 gal. per eight hours after proper adjustments were made.

The field man frequently gets more calls to imaginary troubles than to real ones. One man who had just purchased an engine wrote the general agents that the engine would not run and to send a man at once. If the trouble proved to be due to the engine the company agreed

to pay the expenses of the trip, but if the fault was the owner's, he was to pay them. The pipe leading from the supply tank to the carburetor extended about half an inch into the tank so that any dirt which might be in the tank would not get into the carburetor. There was only enough fuel in the tank to cover the bottom, its level being below the end of the pipe. The man was sure he had fuel in the tank for he could see it. After 5 gal. of gasoline had been placed in the tank and the engine started he paid the bill without a murmur.

In a similar case the fuel tank was outside the building, and, although there was plenty of fuel in it, a cock in the pipe to the engine was closed. This man had been trying for two days to get his engine started.

Fig. 2 is a cross-section of the carburetor and air valve of a throttle-governed engine operating a small electric-light plant. After operating satisfactorily for a day the engine refused to start the next evening. As will be noticed from the sketch, the opening from the carburetor to the port leading to the throttle valve is covered by the edge of the disk *A*, which is held on its seat by a light spring. The disk is perforated so that air may pass through whether the opening from the needle valve is open or not. This valve was stuck on its seat, the result being that the engine could get no fuel.

Another puzzling difficulty was with an engine employing a pump feed for the gasoline. This engine was served by a tank such as illustrated in Fig. 3, the pipe to the fuel pump extending to within an inch of the bottom of the tank. Water had accumulated in the bottom of the tank until it had reached the end of the pipe, and the pump could get nothing but water; hence a stop with considerable difficulty resulted before the trouble was remedied.

The Gas Turbine

Zeitschrift des Vereines deutscher Ingenieure for Mar. 30 of the current year prints a contribution by Dr. A. Stodola, of Zürich, in review and criticism of a recent book by Hans Holzwarth on the theory and construction of the gas turbine, together with operating results obtained from two of that type.

He says that in the Holzwarth design the working cycle is a periodic release and explosion of a predetermined quantity of fresh charge which then, under the effect of expansion, is directed through a nozzle to the rotor of the turbine. When the expansion has continued to nearly the back pressure, air is admitted to the explosion chamber until the latter is scavenged of the burned gases and the temperature is lowered sufficiently. The nozzle valve is then closed and gas or a gas-air mixture under pressure is injected into the chamber for explosion and a repetition of the process.*

The inventor decided on compression of the gas only, whereas the air is supplied to the chamber by exterior pressure and to this end an exhaust fan maintains in the turbine a vacuum of about 3 in. of mercury. As the compression in the explosion chamber rises only to about 18 to 25 lb. the motor belongs to that class of explosion turbines having only very low advance compression; that is, compression for the fuel delivery into the combustion chamber. The designer chose the almost insignificantly low compression of 1.5 to 2 atmospheres because he endeavored to offset the necessary power requirement by the recovery in a regenerator of the heat of the gases of combustion, for which purpose these are conducted to a vertical tubular boiler and the steam thus generated supplies a steam turbine operating the centrifugal compressor and the exhaust fan. A theoretical calculation shows that this energy requirement amounts to 10 or 15 per cent. of the available duty of the turbine, which, calculated as heat, certainly means 35 per cent. of the calorific value of the gas used.

Notwithstanding the considerable heat recovery from the exhaust gases at relatively low temperatures the required heating surface is not impracticably large, but rather comparable to the cooling area of a surface condenser for a steam turbine of like duty.

In so far as the described arrangement is concerned, the Holzwarth turbine does not differ from well known designs, whereas its behavior in the matter of temperature shows a peculiarity of design. Thus the designer absolutely excludes all cooling by either jacket or spray and intends to accomplish the required reduction of combustion chamber and bucket temperatures by means of the scavenging air alone. Professor Stodola says: "In the face of this proceeding, one is involuntarily reminded of Diesel who also sought to build a motor devoid of cooling but who after years of battle against it had to accept the water jacket as a necessary evil." He says further that it is deserving of thanks that the Holzwarth book frankly presents and critically speaks of all the unfavorable experiences and the disillusionments of the early investigations.

Originally the gas-air reservoir was built around the combustion chamber as a jacket and the fresh change was thus heated before its admission. No wonder that there was early or pre-ignition.

In explanation of the great difference observed between the theoretical and the actual duties the author brings forward the difficulties of scavenging the chambers, inasmuch as the jet of gas issuing from one chamber interferes with the free exit of the air from the neighboring one, although the gap had been materially widened and a bypass around the buckets provided for the scavenging air. Generally, however, it seems that the nozzle is to blame as it does not yield a sufficient drop in the pressure of explosion, and thus the issuing jet may not expand to the pressure induced by the exhaust fan and the expansion established outside of the nozzle, in the widened gap, may not operate to the benefit of further acceleration of the jet.

During all the tests unusually low service demand was made on the turbine and it is evident that a 10 to 15 per cent. load requires less cooling.

The admissible temperature for the buckets at full load remains the main question with the gas turbine and this the investigations leave unanswered. It thus seems idle to make conjectures as to its ultimate efficiency, for only in the light of this can a determination be made of its economy as compared with the reciprocating gas engine and the no less serious competition of the steam turbine.

Though the gas turbine problem has not been solved by the Holzwarth design, Professor Stodola acknowledges indebtedness to it for valuable disclosures as to numerous details of the development. Of particular prominence is the series of tests of the first turbine relative to the use of the heavier fuel oils, including coal-tar oil.

In a word, Professor Stodola says that the Holzwarth investigations are a powerful incentive to renewed activity in the direction of the gas turbine, for it is and will continue to be an ideal of the designing engineer.

*See Power, Feb. 6, 1912.

LETTER

Igniter Points

For the make-and-break igniters on our producer-gas engine we had been using points that cost 25c. apiece. This meant $1.50 for points for each set of three igniters, so I began looking for a substitute. After using various metals I tried a brass point in the terminal rod together with a cold-rolled steel point in the igniter stem (or movable stem), and have been getting good service from this combination. For this purpose I turned down some old ¼-in. brass rods to the proper diameter for the points and sawed off pieces as they were wanted, filling a taper on one end to drive into the hole of the igniter terminal. I did the same with the cold-rolled steel points for the igniter stem.

With both points of cold-rolled steel the igniter would miss when running on the batteries, but would run all right direct from the battery-charging generator. However, as that gave such excessive ignition current it caused backfiring.

J. W. Fries.

Middleboro, Mass.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Automatic Refrigeration an Ideal Central Station Load

By Porter L. Swift

Supplying electric current for the operation of automatic electro-mechanical refrigerating plants furnishes one of the largest and most profitable fields for the extension of electric service into commercial use. From the viewpoint of the progressive central-station manager, it is a most desirable addition to the load on lighting mains. The present perfected apparatus will produce no undue line disturbances through the automatic starting and stopping of the motor.

The reasons for the foregoing statements are numerous and self-evident if one will prepare a small chart divided longitudinally into 12 equal divisions representing the months of the year; at the left divide vertically for the degrees of temperature as on an ordinary thermometer scale and at the right divide vertically for the number of hours of darkness per day, which will range from about seven in the summer to nearly fifteen in the winter. On this chart plot the temperature curve for any locality for the entire year and the hours of darkness per day for the same period. The refrigerating duty will follow the temperature curve and the lighting load the hours-of-darkness curve.

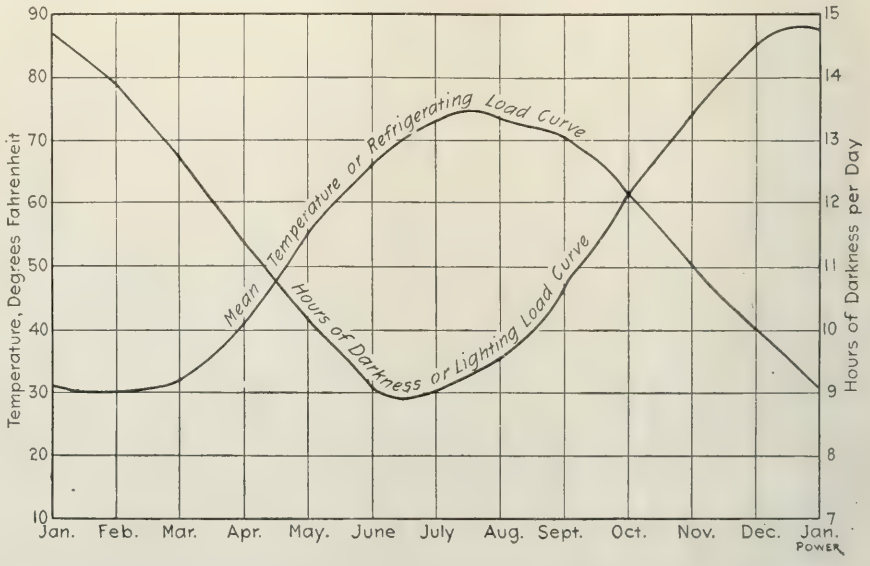
Herewith is a chart for New York. It may be seen that the dark periods range from a minimum of 8 hr. 55 min. in June to a maximum of 14 hr. 42 min. in December. These figures are from sunset to sunrise and the actual dark period in summer is shorter on account of the long light periods before sunrise and after sunset. The temperature curve is the average curve, while the actual refrigerating load will follow the maximum and minimum curve each day. The curve shows an average temperature around 75 deg. F. during July and August, while the maximum temperature for several days reaches 85 deg. and even 100 deg., and in this time there are only six to eight hours of actual darkness. During January and February temperatures as low as 20 deg. are reached for several days when lighting requirements extend 14 hr. or more.

A refrigerating load is the exact reverse of a lighting load, both in the months of the year and the hours of each day. It is heaviest in summer when the lighting requirements are small, and it is a long load, especially with automatic refrigerating plants, when the operation may reach 18 to 24 hr. each day. It is a day load at all times during the year because in the daytime the cooled supplies are removed from the refrigerators and replaced by new supplies. At the time the electric-lighting load is increasing it becomes less because the refrigerators are usually closed and well cooled during the later afternoon.

It would appear then that this class of power consumer ought to be encouraged by a low rate. In one Eastern city of 100,000 inhabitants the power company has found it advantageous to make a reasonably low flat meter rate for its refrigerating business. It was supplying some 21 of these refrigerating plants, last year realizing over $12,000 from this class of service alone. More plants are being installed and the management contemplates still further reducing an already low rate to encourage the installation of refrigerating plants, the majority of which are automatically controlled and driven by electric motors ranging from 2 to 25 hp. This business is nearly clear profit as the peak load for this service does not add to the general peak but follows it and comes at a time when the valley period is greatest. A load that is off the peak is a good one to cultivate, and in order that commercial departments may work up this new business a few words for the automatically controlled and electrically driven refrigerating plant may be of interest.

Any refrigerating load is desirable, but the automatic refrigerating load is doubly so because the automatic plant derives its motive power solely from an electric motor and for this reason alone is worthy of first consideration. The ordinary small refrigerating plant is usually operated during the daytime only and sufficient refrigeration is stored in brine tanks to hold the temperature in the refrigerators within the desired limit; but in extremely hot weather uniform temperatures are difficult if not impossible to maintain. The automatic type of refrigerating plant with its thermostatically controlled motor is available for operation during the whole 24 hr. and will maintain an even temperature in the refrigerator. This feature is especially attractive from a central-station viewpoint in that smaller motors are required to operate the automatic plants; this results in the longer use of a smaller maximum demand than is obtained with the ordinary type of machine, and in turn requires a smaller central-station investment to meet the consumers' demands.

There are few central-station loads which will show a better return per connected horsepower than by motors driving these automatic plants, for it is the rule that this plant will do the work with a 5-hp. motor which a nonautomatic plant will not do as satisfactorily when driven by a 10-hp. motor.



Mean Temperature and Lighting Load Curves for New York

Automatic refrigerating plants are now built having a refrigerating capacity equivalent to the melting of 700 lb. to 16 tons of ice in 24 hr. and they are driven by motors, using either direct or alternating current ranging from 1 to 30 hp. One plant in New York City operates two of these 16-ton units driven by two 30-hp. direct-current motors.

The devices which control the operation of the automatic plant and their functions may be of some interest. The automatic expansion valve is so constructed that, for any given temperature to be maintained around the expansion coils, there will be within the coils a density of vapor proportioned to and governed by the pressure of the vapor itself. This gives a better regulation of the ammonia supply to the expansion coils than is possible with the closest personal attention and a hand expansion valve.

Another type of automatic expansion valve is so constructed that the temperature of the returning vapor from the expansion coils acts on a given volume of liquid and gaseous ammonia confined in a chamber which is connected with a diaphragm controlling the action of the expansion valve. The expansion or contraction of this confined liquid in the chamber, caused by the varying temperature of the vapor from the expansion coils, exerts a pressure on the diaphragm which controls the feed to the pipe coils. Thus the supply of ammonia to the refrigerator coils is governed by the temperature of the suction gas from the same coils after the heat in the refrigerator is absorbed, and the supply will be throttled and cut off entirely as conditions in the refrigerator require. The frost line will also be prevented from reaching the ammonia compressor.

The automatic water valve is constructed to regulate the supply of condensing water corresponding to the temperature and pressure of the compressed gas. The thermostatic control for starting and stopping the motor is so arranged that the motor will stop and start when the temperature in the cooled rooms reaches the two predetermined limits. Another automatic controlling device is a high-pressure safety cutoff and a circuit-breaker.

The subject is worthy of attention and the new business ought to be encouraged by a low rate. A flat meter rate is good, but the machines show up well for both the consumer and the central station if on a sliding scale, depending on the number of hours' use of the maximum demand. The aggregate profit will be large, though made up of small items, when a few of these plants are on central-station lines.

It is the intention of the Massachusetts Institute of Technology to add mechanical refrigeration to its curriculum.

Fundamentals of Refrigeration*

Cold is a relative term, and the degree is measured by the thermometer. The temperature at which water freezes and ice melts is 32 deg. F. To melt ice it is necessary to apply heat, although in the process of melting the thermometer shows no rise in temperature until after the ice is converted into water. The heat necessary to produce this conversion from a solid to a liquid state is called the latent heat of fusion, and is expressed in heat units. The heat unit—that is, the energy necessary to raise the temperature of 1 lb. of water 1 deg. from 62 deg. to 63 deg. F.—is called a British thermal unit, and is abbreviated B.t.u. The latent heat of 1 lb. of ice is about 142 B.t.u. A ton of ice contains therefore

$$142 \times 2000 = 284{,}000 \text{ } B.t.u.$$

This figure has been adopted as the standard of a ton of refrigeration per day, equivalent to the amount of heat necessary to change the ice from a solid to a liquid state. Machines rated at one-ton capacity are able to extract 284,000 B.t.u. during a run of 24 hours, or 11,833 B.t.u. for each hour of operation, this heat-extracting capacity increasing in direct proportion with the increase in capacity of the machine.

Two systems are in use for artificial refrigeration, the direct-expansion system and the brine-circulation system. In the former, the refrigerant piping is exposed within the space to be cooled; in the brine-circulating system the brine is forced through the pipes and returned to the point where the refrigerant extracts the heat from the brine. In the latter case, it is necessary to use a pump to circulate the brine.

To determine the refrigerating capacity of the machine required in any particular case, it is necessary to consider the size of the box, the character of insulation in the walls and the amount of material put into and taken out of the box in a given period of time. A simple and fairly accurate calculation can be made by comparison with the ice consumption.

Suppose a ton of ice is required three times in two weeks during the warmest weather. This is equivalent to 6000 lb. in 14 days or about 450 lb. per day, including Sundays. Assuming that the conditions justify eight hours' daily operation, a one-ton machine operating eight hours daily, or one-third of a day, would give refrigerating results equivalent to one-third of a ton, or 666 lb. of ice. A one-ton machine, although slightly in excess of the requirements, would be the proper size to give results equivalent to the present ice use. Very frequently, however, a lower temperature is desirable, and a larger machine should therefore be recommended.

In the case of a new installation or where no direct comparison with ice can be made, other calculations must be used. This is roughly done as follows: Assume that a butcher with a box 10x10x10 ft., handling 500 lb. of meat per day, delivered to him at about 90 deg. F., desires the box maintained at 34 deg. F. The box is new and the insulation fairly good, the latter having a valuation constant equal to 3. Use this insulation formula:

$$H = cA(t - t')$$

where

H = Total heat loss, B.t.u., in 24 hr. = ?;
c = Insulation constant = 3;
t = Temperature outside = 85 deg. (summer heat);
t' = Desired temperature inside = 34 deg.;
A = Outside area of box = 10 × 10 × 6 sides = 600 sq.ft.

$$H = 3 \times 600 \times (85 - 34) = 91{,}800 \text{ } B.t.u.$$

Use this contents formula:

$$F = ws(t - t')$$

where

F = Total heat in B.t.u. = ?;
w = Weight in pounds = 500;
s = Specific heat of contents = 0.8 (for beef);
t = Temperature of material put into box = 90 deg.;
t' = Desired temperature inside = 34 deg.

$$F = 500 \times 0.8 \times (90 - 34) = 22{,}400 \text{ } B.t.u.$$

The total B.t.u. to be removed by the machine is therefore 91,800 for insulation added to 22,400 for contents is equal to 114,200 B.t.u. each 24 hours. If the machine is to perform the work in 10 hours out of each 24, what size of machine will be necessary? In the foregoing it was learned that a one-ton machine will remove 11,833 B.t.u. each hour, or 118,330 in 10 hours. Therefore a one-ton machine will do the work in this case, if operated between 9 and 10 hours daily in the hot weather, and a lesser number of hours, in proportion to the temperature prevailing, at other times of the year.

In the formula given, the value of the insulation c will vary from 1 up, 3 being an average condition for fairly good butcher boxes. Where old and poor boxes are used or many windows open into the refrigerator, a higher value must be used, perhaps about 5. The specific heat s varies with different substances. Milk is 0.9, beef 0.8, vegetables 0.9, fish 0.85, eggs 0.75.

*Excerpt from report of committee on refrigeration, ventilation and laundry machinery, read before the National Electric Light Association, at Seattle, Wash.

With her engines of 70,000 hp., the 900-ft. "Imperator," of the Hamburg-American Line, will have a speed of 22½ knots.

A Question in Operation

If a line pump is shut down for repairs, should the compressor be shut down or kept running?

If the expansion of the ammonia takes place in coils submerged in a brine or a freezing tank, the machine may be operated for a short time without any inconvenience. As the temperature of the brine in the tank will be somewhat lowered the operation of the machine should accordingly not be continued long enough to freeze the brine in contact with the expansion coils.

In the case of double-pipe brine coolers the expansion valve should be closed and the machine be stopped practically as soon as the brine pump is shut down. The precaution of stopping the machine in the latter case is necessary because of the limited brine volume surrounding the expansion pipe; freezing it would effect a complete stoppage in the brine circuit and possibly burst the pipe.

In shutting down the compressor for two or three hours, or, in fact, for a much shorter time, the expansion valves should always be closed. This is because the higher pressure behind the liquid in the receiver would continue to force through into the expansion coils, and eventually flood the coils. This ammonia would probably make trouble when the machine is again started by freezing back to the compressor, which means that on account of the abnormally large quantity of ammonia in the coils, some of the liquid will be entrained or drawn back with the ammonia gas returning to the compressor.

It is well to close the expansion valves a few minutes before the machine is stopped when the plant is to be shut down for any considerable length of time. This will reduce the back pressure in the coils to something below normal, so that the rise in pressure which always takes place because of the evaporation of the liquid remaining in the expansion coils will not be much above the usual working pressure.

The suction valve of the machine need not be closed unless the ammonia pressure in the expansion coils rises sufficiently to cause the stuffing-box to leak when the packing contracts on cooling. If the discharge valves of the compressor are not tight, leakage of high-pressure ammonia gas back into the cylinder may also give rise to the same difficulty, when the valve in the discharge line must be closed as well. It is advisable not to close this valve unless it becomes absolutely necessary because the starting of the machine without opening it would probably result disastrously to the machine. There being no outlet for the compressed gas, the pressure might readily rise sufficiently to blow off the compressor cylinder head.

Electric Refrigeration

The committee on refrigeration, ventilation and laundry machinery of the National Electric Light Association presented at this year's convention a classified list of industries using electric refrigeration. Table 1 shows principally the variation in the particular classes of business and the averages in the industries specified. The average tons of refrigeration are 6¾ with an average horsepower connected of 15½, or an average ratio of 1:2.3. The average kilowatt-hours per horsepower connected equals 1241, and the average kilowatt-hours per ton equals 2935.

TABLE 1. CLASSIFIED LIST OF INDUSTRIES

Class	No. of Installation	Total Hp.	Total Tons	Total Kw.-hr.	Kw.-hr. per Hp.	Kw.-hr. per Ton
Butter	2	17	9	16,013	942	1779
Candy	1	7	4	3,402	486	851
Dairy	11	177	73	230,494	1302	3157
Florist	3	30	13	38,638	1288	2972
Fruit	1	7	4	7,654	1093	1913
Hospital	1	75	20	103,376	1378	5168
Ice cream	7	45	19	83,162	1848	4382
Meats	14	285	137	339,561	1191	2471
Poultry	2	25	11	22,423	897	2038
Soda fountain	1	8	3	13,674	1709	4558
Saloon	1	8	3	10,418	1302	3472

Table 2 shows the capacity in tons of refrigeration with the horsepower of the motor required to operate, as found in practice.

TABLE 2. MOTOR CAPACITY REQUIRED

Capacity, Tons of Refrigeration	Hp.
1	3
2	5
3	7½
4	7½
5	10
6	12
7	15
8	15
9	15
10	20
12	20
15	25
30	50

Refrigeration in South America

"Refrigeration is not only a business issue but it has a great moral influence," said John Barrett, director-general of the Pan-American Union, at the recent convention of the American Association of Refrigeration at Washington. "I believe today that refrigeration means a new life for a large portion of Latin America and the tropical countries instead of endless apathy and lack of development.

"The whole Caribbean Coast, the Gulf of Mexico and the Mosquito coast, which a few years ago were the home of yellow fever and malaria, through the influence of refrigeration are being converted into countries where men, women and children can live in comparative comfort."

"What is a vacuum?" said the examiner.

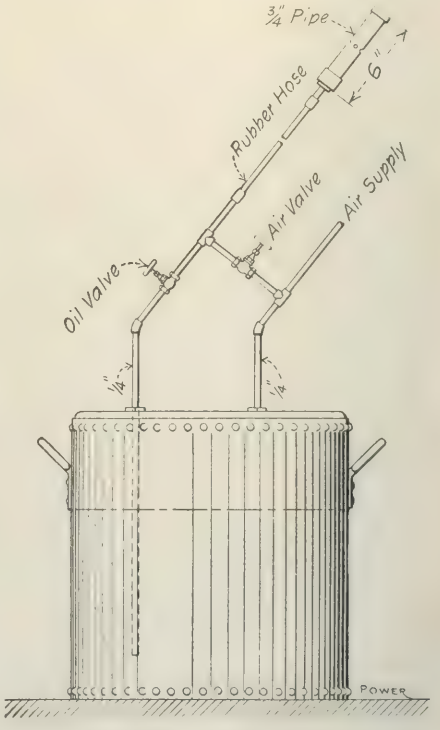
"A vacuum," replied the candidate, as he drew in his breath with a hissing sound, "is a thing that goes like that, and when you have 25 you are all right."

CORRESPONDENCE

Spraying Oil in Absorber Tubes

Every refrigerating engineer has been troubled with leaking absorber tubes while the machine is in operation and in getting the scale out of the tubes when the machine is shut down for winter repairs, but much of this trouble can be avoided by spraying the inside of the tubes with raw linseed oil. They should first be thoroughly cleaned with a flue scraper; then the oil may be applied to good advantage with an oil spray using air. I use the apparatus shown in the sketch. The tank can be placed on top of the absorber and the air piped to it.

This apparatus can be made very cheaply, using any tank that will stand



OIL SPRAYING OUTFIT

a pressure of from 50 to 60 lb. The sprayer is made of ¾-in. pipe for 2-in. tubes and ½-in. pipe for 1½-in. tubes, drilled with four ⅛-in. diametrically opposite holes. To obtain a good spray and not waste the oil, the oil and air valves must be properly adjusted.

After spraying the tubes let water flow slowly into the bottom of the absorber to give the oil on top of the water a chance to cover any unoiled place on the tubes as the water rises.

If this treatment is employed there will be fewer leaks during the season, the scale will not form so thickly on the tubes and they may be more easily cleaned when the machine is shut down for repairs.

GAY A. ROBERTSON.

Louisville, Ky.

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POWER

The Growing Use of Superheated Steam

The rapidly increasing use of superheated steam indicates a keener realization of its merits than has hitherto prevailed. The development is at present far more rapid in locomotive than in stationary or marine practice, but the widespread introduction of the turbine is accelerating its use extensively in marine work. Stationary practice cannot long continue so unresponsive to the influence of this development, for generally speaking, what is true of superheated steam in locomotive or marine practice is also true in stationary practice.

In the fifteen years since superheated steam was first used in locomotives in Europe, the practice has grown considerably there, especially in Germany. Happily America is awaking, as one Western road now has over 500 superheating locomotives in service. In marine practice, especially in warships, commendable progress is being made.

Fortunately, the troubles associated with superheated steam are being rapidly overcome by using better metals and design for the superheater and its moving parts in contact with the steam. The great differences in temperature between the outside and inside surfaces of the cylinder or turbine casing containing highly superheated steam, demand uniformity of thickness of the walls, and efficient insulation to avoid excessive contraction and expansion which is apt to stretch the metal beyond its elastic limit.

This particularly applies to all joints and flanges to reduce the likelihood of thermal losses by leakage. Superheated steam is ideal for turbines, as there are no evils incident to lack of thorough lubrication as exists with reciprocating engines, and the normally dry steam acting on the blades obviates the deleterious effects of saturated steam. But, as is pointed out in the next editorial, superheated steam in practice is only *normally* dry and provision should be made to separate the condensation that is bound to form through unavoidable temperature reductions in the steam lines.

The advisability of using superheated steam centers on the facility with which it may be introduced in nonsuperheating plants. The installation cost of the superheater is comparatively low, and there is no mechanism requiring frequent attention and repair. Moreover, the economy gained by reduced water, steam and coal consumption more than compensates for the cost of the installation and maintenance of the superheater.

Superheated Steam

Ever since the importance of superheated steam to economical engine performance was established, most steam users have been content to utilize it without running it through a separator before its admission to the engine. Indeed, most engineers believe it to be unnecessary to do so and are firm in the belief that "Water, no matter how formed, cannot exist in the presence of superheated steam"; that "it robs the steam of its heat and is itself evaporated into steam."

As a matter of fact, wet, saturated and superheated steam may exist simultaneously in the same section of pipe. Time is required to evaporate the water even though the temperature be considerably above that necessary to do so. The existence of ice in an atmosphere above 32 deg. is somewhat analogous.

Modern turbine practice, perhaps more than any other one thing, has shown the ill effects of water in supposedly superheated steam. The blades of turbines normally using superheated steam have been found to be badly eroded after comparatively short service, and it is possible that serious accidents may result from the turbine receiving a slug of water when being started. That the blades were worn considerably in a short time indicates that wet steam was being supplied. The friction set up by the presence of water in the steam also lowers the efficiency of the machine and both effects combine to make frequent and costly repairs necessary. The results of more investigation into the steam conditions of well regulated plants using superheated steam would be interesting.

Small Exhaust Steam Turbines

It is only within the last few years that the exhaust-steam turbine has been utilized in improving the steam consumption of power plants. The installations to which publicity has been given have been of large units and engineers of small power plants may have gained the impression that the adoption of small exhaust-steam turbines is not practicable.

When the load on a small power plant increases beyond the capacity of the engine there are several ways of securing more power. One is to install a larger or additional engine, which would demand additional boiler capacity, and in instances an enlarged boiler house. Another is to install a condenser, if the engine has been operating noncondensing, and obtain an additional output of between 20 and 25 per cent. Installing a low-pressure turbine between the engine and the condenser is the latest way. Small turbines can be operated with a jet condenser, maintaining a vacuum of from 24 to 26 inches, or a surface condenser can be used for higher vacuum. Much of the energy remaining in the steam after passing through the cylinder of a reciprocating engine with its limited range of expansion can be recovered in an exhaust-steam turbine, because the latter greatly extends the range of expansion. Often the turbine gets as much out of the steam as does the engine through which the steam first passes.

Engineers whose plants are being forced should look well into the merits of the exhaust-steam turbine from the standpoints of economy, small space occupied and low initial cost.

Why Isolated Plants Sometimes Lose

Two reputable consulting engineers after investigating the conditions of the plants in the county buildings of an Eastern state, reported that in no plant were records or other data kept from which the cost of power or light might be determined. The plant engineers "could not tell the boiler horsepower developed, the coal consumption per horsepower-hour, the average number of kilowatt-hours developed for lighting or power, the economy of the engines, or the proportion of steam used for heating, lighting and power."

And these plant engineers are civil-service men too. Is it to be wondered that central stations are making inroads into the isolated-plant field? Where central-station service can be supplied for less than the purchaser's own cost for producing power it should prevail, but there are many instances where over-zealous salesmen for that service secure its adoption where it does not mean higher economy, simply because a knowledge of true conditions is lacking.

That the Freeholders did not provide these plants with record forms is no justification of the engineers' ignorance of operating conditions. Had they fortified themselves with the information, the Freeholders would have been humiliated for their own laxity.

It is just as much an engineer's duty to know what his plant is doing as it is for him to keep the knocks from his engines. But so long as such deplorable conditions exist, those at fault must be content to eventually take their place in the "starter and stopper" class.

Precautions for Safety

Constant vigilance is the price of safety, and the plant or shop employee who is ever watchful has small need for liability enactments and workmen's compensation laws. But as familiarity with danger breeds contempt for its presence the employee must be safeguarded; hence it is necessary to compel employers to adopt safety measures and hold them liable for loss of life and limb.

By the installation of safety devices the Pennsylvania Railroad decreased the serious injuries to its 33,242 shop employees over 63 per cent. in ten months. This achievement is due to the systematic methods advised by an insurance company.

Unless plant owners and employees cooperate in reducing the causes of accidents, the best state-made laws will fail of their purpose. Almost any accident-insurance company is equipped to advise an owner how he can reduce his accident hazard, but "the problem of safety," says Vice-president Atterbury, of the Pennsylvania Railroad, "is not altogether a question of rules and safety appliances; it is the development of inherent self-restraint and control. If we can provide instruction in these principles, we will have taken a long step forward in conserving life and limb and increasing industrial safety."

Office Building Service

When the fifth convention of the National Association of Building Owners and Managers convenes in Seattle on July 11, it is to be hoped that in the discussions on construction, maintenance and service, the office-building power plant will have its share of consideration.

Striving after the "superficial attractions" tends to direct the owner's attention to high-speed elevators, expensively equipped toilet rooms, marble-walled corridors, etc., making him forgetful that most tenants are satisfied with prompt responses to elevator calls, sufficient heat and light—in a word, service.

This class of service is supplied by the power plant. The successful manager knows how much is dependent upon his chief engineer, and when the owner reduces his appropriation the manager is too wise to let the economy affect the power plant.

Good power-plant service has often been the means of attracting tenants dissatisfied with buildings having merely superficial attractions, and in keeping the plain office building on a paying basis.

Sunday Work in Power Plants

Now that the summer is here and the beach and field are alluring, Sunday work in hot boiler and engine rooms seems especially disagreeable. But Sunday work must be done. Unfortunately for them, some engineers have to work Sundays to furnish the power that means comfort and enjoyment to the public.

As long as Sunday work is necessary it should be made as easy and agreeable as possible. It is folly to leave until Sunday, work that could be done as well on week days. Those so favored as to have fresh air and sunshine at least once a week, if in charge of men less fortunate, should have consideration for their being confined long hours in underground rooms, especially boiler rooms, and endeavor to ameliorate uncomfortable conditions. The men will not only be pleased with the management but will cheerfully do better work.

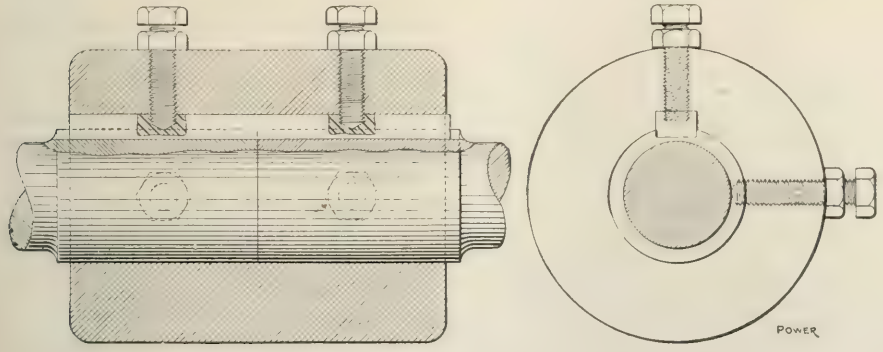
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Correcting a Troublesome Muff Coupling

A part of the shafting illustrated stopped running, and examination revealed that the key had come out of the coupling, and that both keyways in the shaft and coupling were badly worn.

The shaft that had worked out of the coupling was bent and had moved about in it until ridges nearly $\frac{3}{16}$ in. deep were worn in its surface. New keyways were cut by hand and the keys fitted. On



MUFF COUPLING, SHOWING KEYWAYS AND SETSCREWS

starting the engine the shaft wobbled considerably and after running for three-quarters of an hour the keys were noticed to be gradually coming out. They were then made secure by putting 1-in. steel setscrews, drilled and tapped through the coupling and half way through the keys as shown. Similar setscrews with locknuts were screwed into each section of the shaft at right angles to the keys.

This held good for about two months with an occasional tightening when new shafting was installed.

O. G. A. PETTERSSON.

Manitoba, Can.

Flue Gases and Combustion

It is often stated that insufficient air supply is the cause of CO in the flue gases. Except for a few minutes at a time when the steam pressure is falling, CO is formed by this cause in practical operation.

If CO is present, air enough to supply 24 lb. of oxygen could burn 12 lb. of carbon, one-half to CO_2 and the other half to CO, and generate 113,400 B.t.u. Air enough to supply 21 lb. of oxygen could burn 7.9 lb. of carbon to CO_2 and generate 114,000 B.t.u. Now, by so controlling the fire that CO_2 and not CO is made, 3 lb. less of oxygen is needed to give the same total heat. The 113,400 B.t.u. needed can be obtained with less air if the fire is so controlled as to make CO_2 instead of CO, unless the additional air and the air containing 3 lb. of oxygen does not enter into combustion at all, or, all of the heat generated by the added air and by the 3 lb. of oxygen is added as sensible heat to the loss up the chimney.

I do not believe that with more air supplied less will be used, and all the heat generated by the excess air will not go up the chimney. If these conclusions are true, then steam cannot be kept up with insufficient air, though this is conducive to the most economical fire. If CO is formed the remedy is not more air, but a different control of the fire, perhaps more air above the grates and less below, or a thinner fire. If a remedy is sought by simply supplying more air, more steam will be made instead of less coal burned. Of course, all these conditions are more or less modified in practice, but the margins are wide enough to justify the conclusion that the best practice is to use the least air and that if CO is formed more air than necessary is being used.

HARRY D. EVERETT.

Fort Apache, Ariz.

Automatically Emptying a Sump

In a plant of 5000 kw. capacity all the drainage from the engines, etc., empties into a sump having a capacity of about 400 cu.ft. when full. During the daytime the sump fills in about 25 min., but at night, when part of the plant is idle, it requires about twice as long. It is emptied by an ejector supplied with steam through a ½-in. pipe.

The intake pipe to the centrifugal circulating water pump passing directly alongside the sump suggested using that means of getting the water out. A 1¼-in. hole was drilled and tapped, and a pipe extended a little over half way to the bottom of the sump. A float from an old flush tank was fastened to the stem of a butterfly valve connected at the lower end of the pipe to prevent emptying the sump and breaking the pump's vacuum. The steam ejector with its individual controlling valve and float was set high and left for emergency service. This arrangement has worked very satisfactorily, and empties the sump at little cost.

CHARLES A. RICHARDS.

Mountaintop, Penn.

Improved Steam Piping

Fig. 1 illustrates a horizontal tubular and a vertical boiler in which steam is generated for an automatic engine. The vertical boiler has an angle stop valve *A*, and a vertical pipe connected to the reducing tee *B*. This tee has a drip pipe intended to discharge into both boilers, but cannot because it is below the water line of the higher boiler.

This boiler primes considerably, lower-

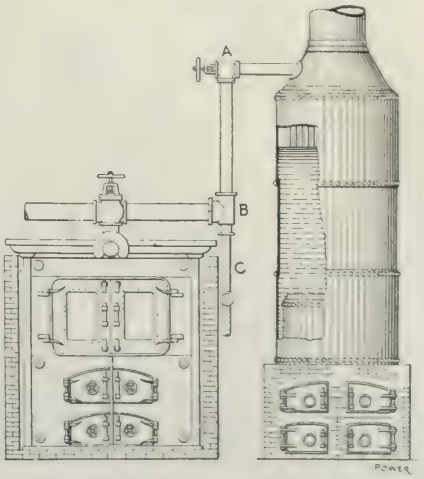


FIG. PIPING OF HORIZONTAL AND VERTICAL BOILERS

ing the water level, and much of the water carried over goes into the lower boiler. If wet steam is supplied by the horizontal boiler, it goes to the engine because it cannot travel against the flow of steam and thus reach the drip pipe *C*.

Fig. 2 shows the changes made in this piping for the vertical boiler. The original stop valve is retained, but a much larger horizontal pipe is substituted above the water line; hence water from priming

and condensation can be returned by gravity to the same boiler. Steam and water flowing through this pipe strike the tee at the left, connected "bull head" for this purpose, and are forced upward, the water being separated from the steam more effectively than with the previous arrangement.

At *A*, Fig. 2, is a continuation of this piping, the ell *B* of which is the same as that shown in the upper left corner. The short horizontal pipe is connected to this ell, followed by another ell *C* and

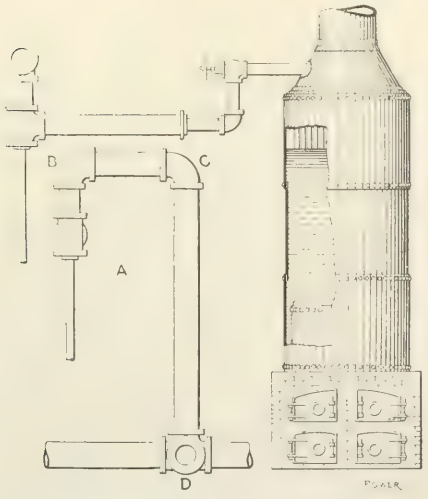


FIG. 2. SHOWING CHANGE OF PIPING

a long vertical pipe which ends in the reducing tee *D;* from this outlet steam goes to the engine. The right opening delivers steam to the heating system, while the left receives steam from the horizontal boiler.

Fig. 3, further illustrating these connections, is a plan of the horizontal boiler showing the angle stop valve *A* connected by a short pipe to the tee *B*. The

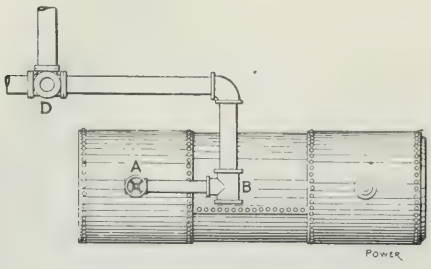


FIG. 3. SHOWING FURTHER CHANGES IN PIPING

horizontal pipe in Fig. 1, originally connected to this tee, was removed and a plug inserted as shown. The reducing tee with the side outlet *D* corresponds to *D* in Fig. 2.

Before these improvements were made, the horizontal boiler and all of the piping shown vibrated excessively because of a slight reduction of pressure in the pipes every time a throttle valve was opened. This loosened much of the brick setting and rendered the whole plant unsafe, but these changes practically eliminated the trouble.

W. H. WAKEMAN.

New Haven, Conn.

What Ruptured the Boiler?

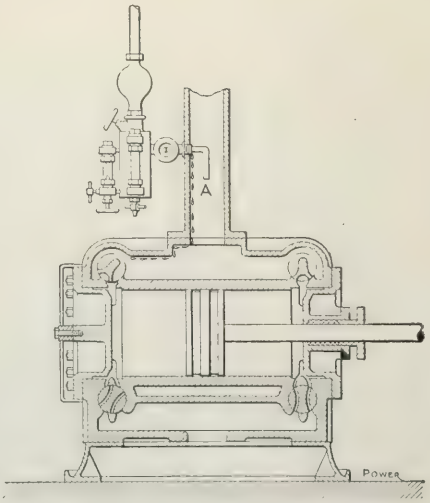
In our plant there are three boilers, each 24 ft. long and working under 80 lb. pressure. The blowoff valve burst on one of them, allowing the water to be blown out. After the boiler cooled off we found a large rupture at the first seam in the firebox and several rivet holes cracked through one to the other for about 3 ft. along the seam. The fire was immediately drawn as soon as we discovered the trouble. Will some interested reader inform me what caused the rupture?

J. W. RIDER.

Rimershing, Penn.

Faulty Lubrication

Unsatisfactory engine performance from faulty lubrication is often due to improperly placed lubricators. The illustration shows a common connection that causes the oil to follow along the side of the pipe and cylinder walls and giving one valve excess lubrication while the others are insufficiently supplied. To correct this a tube carrying the oil should extend to the center of the pipe, as shown at *A*.



LUBRICATOR CONNECTION, SHOWING MOVEMENT OF OIL

One end of the cylinder or valves of a Corliss engine are frequently found worn more than the other, undoubtedly due to improper oil distribution. If the condition obtains it is shown by the engine running poorly.

Some mechanically operated lubricators have no stop valve between the check valve and the cylinder. If the check valve fails to work and lubricator repairs are required, trouble and inconvenience follow from the absence of the stop valve.

C. R. MCGAHEY.

Baltimore, Md.

Slip of Eccentric

These indicator diagrams were taken from the 12x16-in. steam cylinder of an air compressor taking steam at 100 lb. and running at 100 r.p.m. The valve-

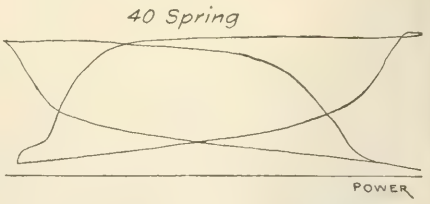


FIG. 1. EFFECT OF ECCENTRIC SLIP

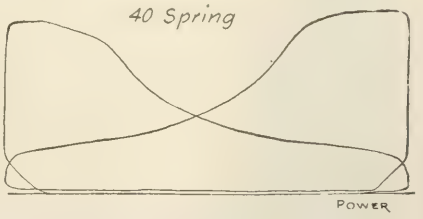


FIG. 2. ECCENTRIC READJUSTED

gear is of the adjustable piston-valve type. Fig. 1 was taken after the eccentric had slipped nearly 90 deg. from normal. Fig. 2 shows the result of properly resetting the eccentric.

WALTER L. HOUSTON.

Republic, Wash.

Securing a Loose Piston

While in charge of the power plant and engineering department of a mine in the mountains of British Columbia, I had an interesting experience on my initial inspection trip with the superintendent. The plant was in a deplorable condition and the superintendent told me to expect almost any kind of trouble.

We were told that the engineer was in the shaft at work on the sinking pump. A few minutes later he appeared, apparently drenched through. He informed the superintendent that the piston was loose again and the pump had stopped. It was a vertical sinking pump of the bucket and plunger type and was hung in the shaft 120 ft. below the surface. This pump drew the water from a sump 10 ft. deep and discharged it to the surface, thus keeping the mine from being flooded. The shaft was sunk about 40 ft. from the bank of the river and the subsoil was sand and gravel.

The leakage in the mine was excessive as the shaft at the time was only timbered. At the bottom the coal vein dipped for 400 yd. at an angle of 45 deg. and consequently if the pump at the bottom of the shaft filled and overflowed, the lower workings would soon be flooded.

It was also found that the steam piston of the pump had previously become loose, due to the stripping of the threads on the rod; attempts had been made to secure it by riveting the rod over the

top of the nut and fitting a taper pin through the nut and rod, but without success. No spare rod or parts were on hand and could not be obtained in less than 10 days, and with this pump out of service the mine would flood in 20 hours. The engineer was instructed to remove the piston and rod and bring them to the surface; meanwhile the writer collected such tools and material as the limited facilities would permit.

The piston was replaced on the tapered rod and a mark scribed on it level with the outer face of the piston, and the projecting end of the rod cut off $\frac{3}{32}$ in. below this mark. This left the small end of the taper 1¾ in. in diameter, into which a ⅞-in. tap hole 2 in. deep was drilled. No stud of suitable size being on hand, one was made from the end of a steel crowbar and screwed in the tapped hole, after which the piston was replaced and a large steel washer ¼ in. thick put on the stud. The piston was forced up hard against the shoulder of the rod, with a nut on the end of the stud to prevent it from backing off. A ⅛-in. hole was drilled in the stud back of the nut and a split-pin inserted. The pump was again in operation three hours after it was dismantled, which is a fair record considering the meager facilities and the difficulties under which the repair was made.

JOHN CREEN.

Seattle, Wash.

Accident to Air Compressor

Recently I was impressed with the fact that considerable care should be taken in removing the head from the air cylinder of an air compressor.

New piston rings were to be put in the piston of the air cylinder. The compressor was not shut down until everything was ready for doing the job quickly, when the discharge valve was closed and the nuts removed from the cylinder head. A chisel was used to force the head from the cylinder and a lighted candle placed on the engine bed to see to work. As soon as the head was loose from the cylinder, gases within, caused by lubricating oil, ignited, blowing the head from the studs and breaking two adjacent pipes. Luckily these were only a 1-in. water pipe and a 1½-in. steam pipe under very low pressure.

J. G. KOPPEL.

Montreal, Canada.

Reference Blanks

While many engineers visit power plants through interest or curiosity, without intending to gather data, they often wish later that they had some particular information relative to these plants. A visitor usually remembers a few facts, such as types, horsepowers or engines, steam-pressure and vacuum-gage readings.

An engineer could profitably equip himself with a few blank sheets arranged in a form similar to that indicated here-

Boilers
- Steam pressure

Feed pumps
- Type
- Size
- Capacity
- Water pressure

Injection water
- Source
- Discharge
- Temp. injection
- Temp. discharge

Prime movers
- Type
- Size
- Hp. or kw

Condensing pump
- Size
- Type

Dry condenser
- Size
- Type

Gage readings
- Steam
- Receiver
- Step bearing
- Vacuum

with, and jot down what he can remember when he leaves. He will feel amply repaid for his trouble some later time when this information may help solve an engineering problem in his plant.

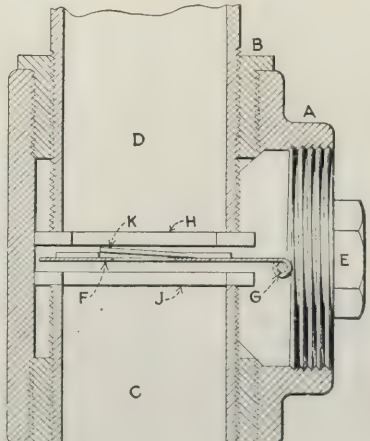
DAVID FLIEGELMAN.

Holyoke, Mass.

Home Made Foot Valve

The foot valve illustrated is one I made for emergency use.

A is a 2-in. tee into the ends of which



DETAILS OF FOOT VALVE

are fitted the 2-in. to 1¾-in. bushings *B*. Into the bushings are screwed the nipples *C* and *D*, 1¾x4 in. and 1¾x6 in. respectively. *E* is a 2-in. plug. The valve *F* is fitted into the tee and swings on the pin *G*. The washer *H* covers the nipple *D* ⅛ in. on its inside edge.

The valve *F* is practically the same as that found in an ordinary hand pump, containing a flapper which is shown at *K*. The action of the valve is obvious.

A. ENGELHARDT.

Newburgh, N. Y.

Steam Trap Experience

We have six bucket-type traps which take care of the condensation in a 14-in. loop and the receiver separators on the engines. The traps discharge into a 4-in. line to a receiver pump, which forces the water back to the boiler. The condensation from the engine jackets runs to a tail trap on the first gallery of the engines and then to a 4-in. drain line in the basement. The tail trap has a vent so the water will drain to the basement. The pressure carried on the low-pressure jacket is 2 lb. The needle valves on the bucket traps gave trouble, due to cutting and grooving, thus putting a pressure on the drain line and forcing the water back up the jacket pipe and out the vent.

Putting check valves on the vent pipes prevented the water from overflowing, but still was not satisfactory. The traps were on a bracket between the steam main and the drain line, so we lowered them 2 ft. below the drain line, thinking that by making the traps discharge against a head the valves would last longer, as they would not hold tight more than two weeks. This new position was very convenient for repairs, but did not help the valves much.

A repairman suggested using a metal other than bronze for the valves, and tool steel was tried. The first or experimental valve is still doing duty and giving excellent results after six months' service. All our traps are now equipped with steel valves.

HENRY R. BLESSING.

Philadelphia, Penn.

Is the Packing Defective?

We are having trouble with the metallic sectional ring packing in the piston-rod stuffing-box of our compound engine. Our efforts to prevent leakage here avail nothing.

The packing consists of six rings, each divided into three pieces, forming scarf joints. A spring fits over each ring which causes it to bear on the rod. A case covering two rings forms part of the packing, and in this instance three cases, each containing two rings, are used. This packing has been in service a little over a year and leaks considerably.

Can some reader suggest a remedy for the trouble?

JAMES PIERPONT.

Oakville, Vt.

An accumulation of one-eighth inch of scale will make a difference of 15 to 18 per cent. in evaporation.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Calculation of Heat Value of Coal

On page 558 of the Apr. 16 issue; Mr. Kowalke gave a method of calculating the heat value of coal from the proximate analysis. The following modification of his formula enables it to be used without the aid of an auxiliary curve.

Mr. Kowalke's formula is:

$$\textit{B.t.u. per pound of coal as received} = 14{,}760\ C + aV \qquad (1)$$

Where C is the decimal percentage of fixed carbon in the coal as received, V is the decimal percentage of volatile matter in the coal as received and A is a constant which varies for different coals.

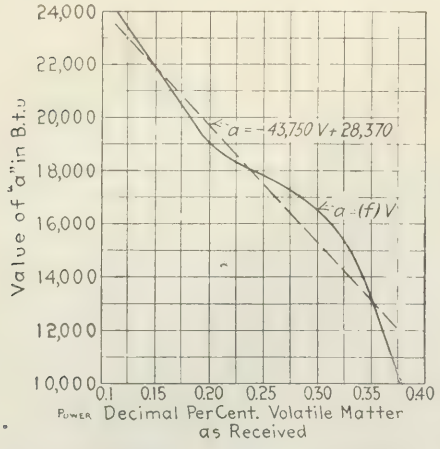


CHART SHOWING HEAT VALUE FACTOR FOR VOLATILE CONTENTS OF COAL

An auxiliary curve giving values for a is necessary for substitution in the formula.

It appears that a is a function of V, but apparently the relation of a to V cannot be expressed mathematically. If it could, there would be no need for the curve. From an observation of the aV function it is seen that a straight line can be superimposed, so that it approximates the relation. The exact location of the straight line is a matter of individual judgment. But having once decided upon it, and assumed that it represents the relation of a to V, it can be stated in mathematical terms, thus eliminating the curve.

I have done this, and found the relation between the variables a and V (as shown by the straight line) to be

$$a = -43{,}750\ V + 28{,}370 \qquad (2)$$

Substituting (2) in (1)

$$\textit{B.t.u. per pound} = 14{,}760\ C + 28{,}370\ V - 43{,}750\ V^2$$

which is the desired formula, in which may be substituted the values of C and V as found from the proximate analysis, without reference to any curve.

It must be noted that this formula gives results which differ somewhat from those found by means of Mr. Kowalke's curve and formula. The maximum difference comes from

$V = 30$ *per cent. to* $V = 33$ *per cent.*

where the curve "humps up," and beyond $V = 37$ per cent. The difference at these points may reach 5 per cent.

Solving the example, given in the issue mentioned, of coal containing 40.63 per cent. of fixed carbon, and 34 per cent. of volatile matter

$$\textit{B.t.u. per pound} = 14{,}760 \times 0.4063 + 28{,}370 \times 0.34 - 43{,}750 \times 0.34^2 = 10{,}585.5$$

Mr. Kowalke's formula and curve gave 10,773 B.t.u. per pound. The difference is 10,773 — 10,585.5 = 188 B.t.u. per pound, or about 1.7 per cent.

G. H. CRAWFORD, JR.

Washington, D. C.

New Valve Effects Economy

In the May 14 issue, J. C. Hawkins told of putting in a new valve and securing a quiet running engine with less steam consumption, but he did not state whether he figured out how much outside lap was needed to cause the valve to cut off at half stroke, and if so how, or just guessed it correctly.

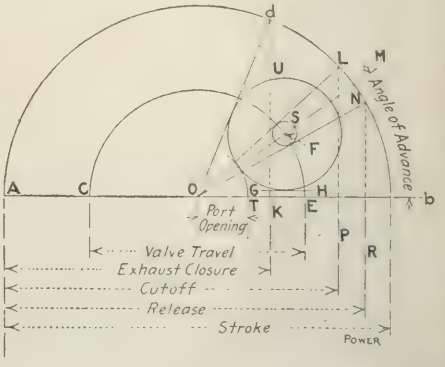
The Bilgram valve diagram would have been very useful to Mr. Hawkins. In a case of this kind the outside and inside lap are known, and if the lead is not known it may be assumed.

To any convenient scale draw on the line AB with O as a center the semicircle ADB having a radius equal to that of the crank (half of the stroke). About O as a center and with a radius equal to one-half the valve travel, describe the semicircle COE. Draw a line GH parallel to AB and at a distance from it equal to the lead. With a radius equal to the outside lap of the valve and with a center F on the semicircle COE describe a circle U that it tangent to GH. About F as a center and with a radius equal to the inside lap of the valve describe a circle S.

Then draw the lines OD, OL, OM, etc., and from their points of intersection with the semicircle, as DLN, etc., drop vertical lines as DK, LP and NR.

With O as a center describe an arc tangent to the outside lap circle and intersecting AB at T. The rest of the diagram is self-explanatory. The angle OMB is the angle of advance. If there is no inside lap no inside lap can be drawn, but perpendiculars dropped from M will indicate exhaust closure and release.

To cause the valve to cut off at half stroke, the lead remaining as before, it is evident that the line OD will be vertical. Describe a circle tangent to OD and GH with its center on the semicircle COE. Now to know the extra outside lap needed to cut off at half stroke, subtract the radius of the first lap circle from the radius of the second circle drawn. This will be the outside lap necessary to add to the old valve to cause it to cut off at half stroke (to the same



BILGRAM VALVE DIAGRAM

scale, of course), or, the radius of the new lap circle, will represent the outside lap for the new valve to cut off at half stroke.

The effects on the other events of the stroke by increasing the outside lap, the lead remaining as before, will cause the center F to be moved farther to the left, and by examination it is seen that exhaust closure will be earlier, release will be earlier and the angle of advance will be increased and, unless the travel of the valve is increased, the port opening will be decreased, because the circle U will be larger and come nearer to O; therefore the distance OF will be less.

A careful study of this diagram will be found very useful. For instance, if a person wished to increase the lead, it is easily seen that all other events of the stroke will take place earlier and to do so the angle of advance must be increased.

B. COESRIB.

Detroit, Mich.

Automatic vs. Fixed Cutoff for Compound Engines

C. J. Mason's letter in the May 14 issue does not give the real reasons for using a fixed cutoff on the low-pressure cylinder of a compound engine. It is because this cutoff is invariable and the reciprocal of the ratio of the two cylinders.

By drawing theoretical diagrams for different loads it will be found that the correct point of cutoff for the low-pressure cylinder remains the same for any high-pressure cylinder terminal pressure. At the end of the stroke in the high-pressure cylinder is a unit volume of steam at the terminal pressure. As shown in Fig. 1, this can be expressed by the product of the volume and pressure. If we call the high-pressure cylinder volume 1, then the energy in the steam doing useful work is $1 \times 30 = 30$. Assuming that the steam follows the simple and convenient law, $pv = constant$, this 30 will be the constant.

Now the steam exhausts and mingles with the steam in the receiver, which is assumed to be large enough to equalize all irregularities. The receiver pressure will then be constant and the same amount will remain in the receiver all the time, the steam doing useful work simply passing through the receiver from the high-pressure to the low-pressure cylinder. The effect of the receiver can therefore be neglected. Then the same amount of steam exhausted into the receiver, $pv = 30$, must be drawn from it to keep its pressure uniform. But the cylinder which draws the steam from the receiver is four times as large; hence the steam available will only fill ¼ of its volume and we have to cut off at ¼. If cutoff is earlier, more steam is put into the receiver than is taken out. This raises the receiver pressure until the equilibrium is again restored, as shown in Fig. 2, where the receiver pressure is higher than the terminal pressure, resulting in a loop.

If cutoff is later, more is withdrawn from the receiver than was put in, resulting in a drop in the receiver pressure, as shown in Fig. 3.

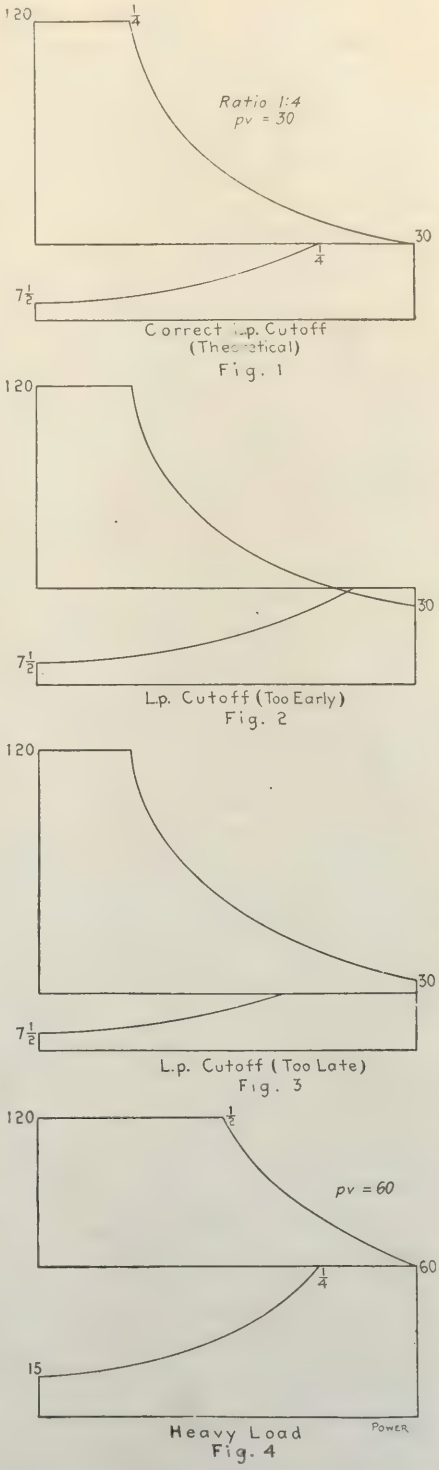
The low-pressure cylinder terminal pressure is fixed by the high-pressure cylinder terminal pressure; it is

$$\frac{30}{4} = 7\tfrac{1}{2}\ lb.$$

irrespective of the point of cutoff in the low-pressure cylinder. This is in accordance with the above mentioned law. At the end of the low-pressure stroke we exhaust four high-pressure volumes; hence the pressure must be equal to

$$\frac{30}{4} = 7\tfrac{1}{2}\ lb.$$

Fig. 4 shows the diagram for a high-pressure cylinder cutoff at one-half stroke. There the amount of steam going through the engine is $\frac{1}{2} \times 120 = 60$ lb.; the high-pressure cylinder terminal pressure is $60 \div 1 = 60$ lb.; the low-pressure cylinder terminal pressure $60 \div 4 = 15$ lb.; the low-pressure cylinder cutoff $15 \div 60 =$ one-quarter stroke.



COMBINATION HIGH- AND LOW-PRESSURE CYLINDER DIAGRAMS

The diagrams are purely theoretical, neglecting clearance, back pressure, etc., and also the fact that in practice it is desirable to have a slight drop at the end of the high-pressure stroke.

F. F. NICKEL.

East Orange, N. J.

Benefits of Mutual Aid

The comment by A. Rauch in the May 28 issue on the Foreword in the Apr. 2 issue is a step in the right direction. If most chief engineers were of the same mind there would be more engineers of higher capabilities. But the man that has to find out things for himself, and has the will to do so, gets there in the end. I have seen the time when I would have given a great deal to have had the chief explain a few things to me, but as he would not I had to find out the best way I could; it took me longer, but I learned it thoroughly. Those calculating to be successful in engineering should spare no reasonable expense or time to develop themselves.

There is a great difference in men. I worked three years in one plant, and had eighteen different fellow engineers with me during that time. I took two engineering journals at that time, and I do not remember of but three men out of the eighteen that took any interest in them aside from looking at the pictures. Each of those has a good position at present; the others are probably just "staying" somewhere.

I know of one instance where a chief engineer lost his position by not helping his assistant. He would not cooperate with his help, refusing to show them or answer questions. The assistant was studious and became a better engineer than his chief, and eventually got the place. If the chief tries to show his help, and answers the questions they ask, they will respect him, and try to live up to the standard he sets, which will more than repay him for the time spent.

V. C. WOOD.

Copenhagen, N. Y.

Instruments and the Engineer

Mr. Martin should receive different advice from that given in John Strong's recent letter. The following incident well illustrates the value of instruments to an engineer.

The plant was under the direct supervision of a master mechanic with two watch engineers working twelve hours each at $2.70 per day. Neither engineer was in charge.

The company was prosperous and able to purchase anything they needed, but could not be interested in an indicator. One of the engineers, more for his own education than the firm's benefit, bought one on monthly payments that he saw advertised in POWER, but with a family of four to support on $2.70 per day it meant a great sacrifice for him. The in-

dicator was put to practical use in setting valves, and bettering conditions in the plant. Later the fuel consumption increased unusually when No. 2 compressor was run; the indicator showed that the high-pressure cylinder was doing no work, as the piston was broken in very small pieces and lying in the bottom of the cylinder; the low-pressure cylinder was receiving the high-pressure steam and doing all the work.

When the master mechanic examined the trouble he noticed the indicator and inquired about it. He found he had a progressive engineer with more intelligence than he had realized. The engineer was made chief with a substantial increase in wages.

This plant has since grown to large size and the engineer who considered the indicator indispensable to his education is a salaried man with one of the best jobs in the city, gets a ten-day vacation every summer, and is in line for higher promotion. Most assuredly it pays to show your employers you are interested in their business welfare.

I have employed many engineers but find most of them as adverse to the use of indicators as they are to mathematics.

W. D. Borrie.

Richmond, Va.

Cotton Rope Drives

The editorial on the above subject in the Apr. 9 issue calls to mind my experience with cotton ropes when I once tried cotton rope on the strength of what I had heard of its lasting qualities. I installed it on an engine drive, the driving-wheel of which was 22 ft. in diameter, driven wheel 6 ft. diameter, with 48 ft. center to center of the wheels; the rope speed was 5000 ft. per minute and thirty 1½-in. ropes were used.

The grooves were standard, with inclined bottom sides at 45 deg., with curved sides joining the angle. The drive was installed by a cotton-rope expert sent out from the rope mill.

The drive started out in fine shape and the white ropes gliding by made a splendid appearance, but six hours after it was necessary to stop the engine and take up four of the ropes on account of the stretch; this delayed us two hours. We started up again and five hours later we stopped to again take up the ropes.

This continued for a whole week, starting and stopping the plant every six hours to take up ropes, until the entire 30 ropes had all been taken up. A high-priced lubricant was used on the ropes as recommended by the rope expert; he was game, however, and stayed with us.

The second week started in with more delays and more taking up of ropes. One of the ropes broke during the third week and we replaced it with a manila rope. During the past six months that these cotton ropes have been in use, as fast as they give out we have been replacing them with manila ropes, the use of which has never given trouble. From my experience a manila rope is by far the superior of the cotton rope.

A. Rauch.

Pittsburg, Penn.

Cable Cuts Shaft

John T. Chambers' letter in the Apr. 2 issue brings to my mind a similar case of shaft cutting in which a ¾-in. wire cable was used to drive a machine which was about 200 ft. from the line shaft.

An electric motor was installed to run the machine and the cable discarded; the cable was just lifted off from the grooved pulley and allowed to hang on the shaft. No attention was given it for a couple of hours and then a man went to remove it. He found that during the time the line shaft had been running, the cable, which was stationary, of course, had cut a groove in the line shaft over half way through, and, strange to relate, the cable was not injured or worn in the least. The shaft was not renewed as it is still strong enough to carry the load.

John Thorn.

London Junction, Ont.

Pump Would Not Force Water

J. W. Dickson in the Apr. 23 issue asks why raising the suction pipe of his centrifugal pump made it work where it would not work before.

Although he says $100 was spent in trying to prime the pump, he does not say whether the suction pipe was clear at the bottom. I believe his trouble was due to the end of the suction pipe being buried in the sand or mud in the bottom of the well, and raising the pipe 4 in. lifted the strainer out of the mud so that it could get water.

About three years ago I had a similar trouble with a sump pump. The motor was direct-connected to the pump in the bottom of the pit, and the pump had a lift of about 5 ft. and a discharge head of about 30 ft. It supplied water to the skid pipes in the furnaces and the waste from these pipes and the rolling mill ran into this sump and overflowed to the sewer. The sump was of concrete and there was a distance of over 2 ft. from the end of the suction pipe to the bottom of the sump to allow for the collection of sediment.

The pump worked fine for a few days, but the water supply began to decrease and finally stopped entirely. The pump was repacked and primed, but it did no good. Examination of the suction pipe showed that the strainer was buried in sand and scale. After the sump was cleaned out, no further trouble was experienced.

J. C. Hawkins.

Hyattsville, Md.

When Mr. Dickson raised the suction pipe a pocket was formed which held the horizontal length and part of the riser length full of water; in fact, the water in the riser will be at a higher level than that in the raised end of the horizontal length on account of a partial vacuum being formed between the check valve on the discharge side of the pump and the water level in the riser. This makes a short lift when starting.

In such an installation a foot valve placed in the suction line would prevent the formation of a pocket in which sediment collects, thereby reducing the area of the suction pipe. A pocket also makes it difficult to drain the pipe in winter.

Henry C. Briggs.

Jersey City, N. J.

Brown Engine Diagrams

In the June 4 issue, Mr. Chase asks advice about his Brown engine diagram. It would seem at first that the valves were improperly set, but I think if more care had been taken in tracing the cards, things would not look so bad. The crank-end card shows that the steam cam is too far advanced, causing too much lead, which makes the steam line incline downward to the point of cutoff. I would also advance the exhaust cam to give an earlier compression to increase the economy. The head-end card is not so bad as it looks to be. It shows that it was traced over three times, and at no time with the same governor position, and that the paper on the indicator drum had moved upward, causing the card to be taken lower than the crank end. The steam cam on the head end should be advanced because the round corner at the top of the card shows the cylinder did not receive full pressure at the start.

The exhaust cam should also be advanced for more compression to equal that of the crank end. The cutoff will need adjusting, but the crank-end knock-off cam should not be disturbed because in most old Brown engines it is permanently fixed, and the adjusting is done by the head-end taper pin. The head-end cutoff should be made to equal the cutoff on the crank end. If there is no three-way cock on the indicator piping one should not depend much on the cards being equal, because the time taken to manipulate two globe valves is such that the governor will change its position, and one might think that the cutoff was not correct even when it was. After making the above changes, raise the governor balls as far as they will go, and then let them down ½ in. Put a stick between the balls to hold them in this position. Now turn the engine over and see that the valves do not pick up enough to uncover the lap on the valves. They should not or the engine would race if the main belt broke.

A. C. Waldron.

Revere, Mass.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Varying Speed of Engine

Can a 14x30-in. Corliss engine, which makes 100 r.p.m., be converted into a good variable-speed engine by installing a variable-speed belt drive for the governor?

H. F. O.

From the difficulty of obtaining a positive speed from almost any kind of belted variable-speed mechanism, the ordinary Corliss-engine governor will not work satisfactorily with any such form of drive. A better way to vary speed within moderate limits, would be to vary the governor loading by changing the position the governor balance weight. Before placing any dependence on speeding up of engine, a test should be made to determine the highest speed at which the cutoff will work properly.

Size of Boiler Steam Pipe

How is the proper size of main steam connection and main steam pipe for a boiler found?

J. McG.

The sizes of main steam connection and steam pipe are determined according to the amount of reduction from boiler pressure which is permissible in delivering the required flow of steam to the point where the steam is used.

In ordinary power-plant practice, more than 5 lb. reduction of pressure is undesirable; and with the usual number of fittings and length of steam pipe employed, greater reduction of pressure is likely to result unless the pipe is made large enough for the steam velocity in the pipe to be as low as 6000 ft. per minute, or 100 ft. per second.

The rule ordinarily employed for finding the area of steam-pipe connection for a power boiler is: multiply the area of the engine piston by the piston speed and divide by the allowed mean steam velocity in the pipe.

Point for Cutting Off

How can the point of cutoff be found which, with a given initial pressure, will produce a given mean effective pressure?

P. E. M.

Assuming that the absolute pressure varies inversely as the volume, which is approximately true with unjacketed cylinders using dry saturated steam, the mean forward pressure per pound of initial will be given by

$$P_m = (1 + \log_e R) \frac{f + c}{R + c} - c$$

where

P_m = Mean forward pressure per pound of initial pressure (absolute);

$\log_e R$ = Hyperbolic logarithm of the ratio of expansion;

R = Ratio of expansion $= \frac{1 + c}{f + c}$;

f = Fraction of stroke completed at cutoff;

c = Clearance in percentage of piston displacement.

Assume an engine with 5 per cent. clearance, cutting off at quarter stroke, 120 lb. absolute initial pressure, exhaust pressure 15 lb. absolute.

The ratio of expansion would be

$$\frac{1 + c}{f + c} = \frac{1.05}{0.25 + 0.05} = 3.5$$

The hyperbolic logarithm of 3.5 is 1.2528. Then the mean forward pressure with an initial pressure of 120 lb. absolute would be

$$120 \times [(1 + 1.2528)\ (0.25 + 0.05) - 0.05] = 120 \times [(2.2528 \times 0.30) - 0.05] = 75.1008\ lb.$$

say 75 lb. Subtracting from this the back pressure, 15 lb., gives

$$75 - 15 = 60\ lb.$$

as the ideal m.e.p. This would have to be multiplied by a diagram factor to allow for falling steam line, rounding corners, compression, etc., the percentage of the ideal diagram which the actual diagram is likely to cover.

This formula cannot be worked backward to find the point of cutoff directly, and the only way is to try successive points until the right one is found. A table, such as will be found on page 115 of Low's "Steam Engine Indicator," enables one to locate it closely.

Size of Steam Exhaust Connections

A 44x80-in. engine is run at 75 r.p.m. What actual diameter of steam and exhaust-pipe connections should be used?

J. D.

The area of the 44-in. diameter piston is 1520.53 sq.in. and the piston speed is

$$\frac{80}{12} \times 75 \times 2 = 1000 \text{ ft. per minute.}$$

Assuming that an engine of this size would cut off at one-half stroke, or earlier, it would be safe to assume an average velocity for steam in the pipe at the engine as high as 9000 ft. per minute; hence the size of steam pipe at the engine should be

$$\frac{1520.53 \times 1000}{9000} = 168.95\ sq.in.$$

in area or 14.6 in. in diameter. Not to creat an undue amount of back pressure the size of the exhaust pipe should be proportioned for a steam velocity of 6000 ft. per minute; hence the exhaust pipe at the engine should have an area of

$$\frac{1520.53 \times 1000}{6000} = 253.42\ sq.in.$$

or a diameter of 17.96 in.

The nearest larger standard sizes are a 15-in. diameter steam pipe and an 18-in. diameter exhaust pipe.

Safety of Flywheel Rim

If the centrifugal force which tends to disrupt a flywheel rim is directly proportional to its weight, why is it that increasing the sectional area of the rim would not increase the danger of rupture?

C. H.

Because the cross-sectional area, and consequently the strength of the rim, would be increased in direct proportion with the increase of weight.

Atmospheric Pump

What is an atmospheric pump?

W. H. K.

An atmospheric pump is one in which water is raised by the pressure of the atmosphere. The term, atmospheric pump, though rarely used in present-day practice, is a more scientific name for "suction pump."

Weight of Boiler Plates

What is the need of limiting the variation in weight of a unit area of boiler plates? What is the objection to having one long longitudinal seam the entire length of a boiler?

H. C.

Uniformity in weight of a boiler plate per unit of area, is one of the best checks on its uniformity in thickness. A longitudinal seam the entire length of a boiler is objectionable because it results in greater inequality of stiffness and expansion between the opposite sides of a boiler shell than when the length is broken into several girth courses with the horizontal seams staggered.

Study Questions

This Week's Questions
Last Week's Answers

36. What horsepower is required to drive a 9x13-in. ammonia compressor working against 180 lb. condenser pressure and 15 lb. suction pressure at a speed of 90 r.p.m., assuming a volumetric efficiency of 65 per cent.?

37. A single-acting pump has leaky discharge valves so that for each forward stroke when it discharges 3 gal. it draws back 2 gal. on the return stroke. It makes a forward stroke in ⅗ of a second and a return stroke in ⅖ sec. Starting at the end of a return stroke with water at the bottom of a standpipe holding just 22 gal., how long will it be before water is forced to the top?

38. A building one mile from the source of electrical supply has two thousand 60-watt lamps and a motor load of 50 hp. The voltage at the building is 550 and the drop is 5 per cent. of the initial voltage. What is the ohmic resistance per mile of the line from the power house?

39. A fireman had been having trouble with the injector on one of his boilers for a number of days but finally succeeded in fixing it. When asked how long it was out of order and how long since it was fixed he replied: "When it is the day after tomorrow I fixed it a week before yesterday and my trouble with it began the day after five weeks before a fortnight and three days after the day before a couple of days after I fixed it." How long had it been repaired and how long was it out of order?

40. What is the tensile stress per square inch in a 24-in. pipe, ⅜ in. thick, under a pressure of 80 lb. gage?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

31. Let x = the total number of lamps and a, b, c and d the number of lamps in each row when there are three, four, five and six rows respectively.

Then

$x - 1 = 3a$ $\qquad$ $x = 3a + 1$
$x - 2 = 4b$ $\qquad$ $x = 4b + 2$
$x - 3 = 5c$ $\qquad$ $x = 5c + 3$
$x - 4 = 6d$ $\qquad$ $x = 6d + 4$

A direct solution cannot be obtained as there are only four equations and five unknown quantities, so the solution must be obtained by trial. It is best to begin with d, as that must be the smallest of the unknown quantities, and assume different values for it beginning with 1, until a value for x is obtained which satisfies the other equations, remembering that all of the unknown quantities are whole numbers and x is less than 100. When $d = 4$, $x = 28$ and the third equation is satisfied, but neither of the other two. The next value for d that will bring c out a whole number is 9, x then being 58, and this also satisfies the first two equations; hence is the right answer. The only number of rows into which the lamps can be divided evenly is two.

32. Imagine the stairway unwound and stretched out as a straight line. It becomes the hypothenuse (h) of a right triangle having an altitude (a) of 200 ft. and a base (b) of 5 times the circumference of the tower.

From

$$h^2 = a^2 + b^2$$
$$(282.8333)^2 = (200)^2 + b^2$$
$$b^2 = 39{,}994$$
$$b = 199.98 \text{ ft.}$$

The circumference of the tower is

$$\frac{b}{5} = 39.996 \text{ ft.}$$

and the diameter

$$\frac{39.996}{\pi} = \frac{39.996}{3.1416} = 12.731 \text{ ft.}$$
$$= 12 \text{ ft. } 8\tfrac{3}{4} \text{ in. nearly}$$

33. Absolute zero measures an imaginary condition where no heat is present. It has been determined by calculation to be —460 deg. F. or —273 deg. C. Therefore, 60, 100 and 247 deg. F. would be 520, 560 and 707 deg., respectively, above absolute zero, and 0 and 100 deg. C. would be 273 and 373 deg. above.

34. At 62 deg. F., 2 gal. of water weighs

$$8.3356 \times 2 = 16.6712 \text{ lb.}$$

Then

16.6712 *lb.* × 9 *deg.* = 150.0408 *B.t.u.*

is the amount of heat lost by the iron to the water and corresponds to a reduction in temperature of

$$\frac{150.0408}{0.13} = 1154.16 \text{ deg. F.}$$

The original temperature of the iron therefore must have been

$$71 + 1154.16 = 1225.16 \text{ deg. F.}$$

35. Using the prismoidal formula

$$V = \frac{h}{6}(A_1 + 4A_m + A_2)$$

where

V = Volume;
h = Height;
A_1, A_m and A_2 = Top, middle and bottom areas, respectively;

the volume of the chimney, if solid, would be

$$V_1 = \frac{32}{6}\left[\frac{(40)^2 + 4\left(\frac{40+56}{2}\right)^2 + (56)^2}{144}\right]$$
$$= 516.74 \text{ cu.ft.}$$

The volume of the space inside the stack is

$$V_2 = \frac{32}{6}\left[\frac{(24)^2 + 4\left(\frac{24+40}{2}\right)^2 + (40)^2}{144}\right]$$
$$= 232.29 \text{ cu.ft.}$$

The actual volume therefore is

$$V_1 - V_2 = 516.74 - 232.29 = 284.45 \text{ cu.ft.}$$

and the weight, at 120 lb. per cubic foot, is

$$284.45 \times 120 = 34{,}134 \text{ lb.}$$

Stole Current Worth $100,000?

Two men were arrested in New York City on June 21, charged with defrauding the Brooklyn and New York Edison companies, causing a loss of $100,000.

According to the press reports, the men, an ex-policeman and an Edison electrician, had the assistance of dishonest users of electricity. Much of their recent work has been done on Coney Island, where they are said to have been in collusion with 50 users of electricity.

The officials are not sure whether these dishonest users will be proceeded against, but they threatened to turn off the lights in the 50 places where they know they were cheated. A police inspector says that one of the accused men would go to a dishonest consumer and offer to cut the bills down. When the financial agreement had been reached he would take a key that opened meters, and with a certain piece of mechanism, turn the meter back. He would visit a place just before the company's reader came around, and the amount lost would be almost the whole sum due. A $500 bill has been reduced to $100. It was the inspector who estimated the losses of the companies at $100,000.

The Edison companies have known they were being defrauded for the last 11 months, and four months ago they asked the detective bureau to find the men responsible. The detectives who made the arrest, it is said, saw the men tampering with the meter of a big Coney Island dance hall.

Similar contrivances have been used before. In 1907 the police arrested four persons who had tried them. They were simple, but required an expert knowledge of meters to use them.

A man arrested in 1907 showed at his trial just how his contrivance was used. He had discovered that by disconnecting a switch and attaching a "ground wire" the meters would not register most of the electricity that passed through. With this device a customer might use $20 worth of electricity in a month and the meter would show only $2 worth. The man pleaded guilty and was sent to jail for a year.

An Intentional Boiler Explosion

A demonstration, which promises to revolutionize the construction of locomotive boilers, was made at Coatesville, Penn., on June 20, under the direction of Dr. W. F. M. Goss, Dean of the College of Engineering, University of Illinois. Two full size locomotive boilers, designed for high-speed heavy passenger service, were each subjected to severe low-water tests. Both boilers were identical in size and design, except that one had a Jacobs-Shupert sectional firebox, built by the Jacobs-Shupert U. S. Firebox Co., while the other had an ordinary radial stay firebox.

FIG. 1. "BOMB-PROOF" FROM WHICH BOILERS WERE OPERATED DURING TESTS

For the purposes of these tests, both boilers were mounted in a field and 50 ft. apart, and were operated from a "bomb proof" located 200 ft. away from the nearest boiler. Oil was used for fuel because of the danger to which a fireman shoveling coal would have been exposed. Fig. 1 shows the "bomb proof" and on top may be seen the telescope for reading the graduated water gages and pressure gages on the boilers, as shown in Fig. 2.

In the process of testing, each boiler was brought to a condition of operation closely approaching maximum power estimated to be 1400 hp., which is equivalent to that required to haul a heavy passenger train at 60 miles per hour. The supply of feed water was then shut off, but all other conditions were continued unchanged.

The water level gradually fell under control, exposing the crown-sheet and other portions of the heating surface to the full effect of the fire. The boiler having the Jacobs-Shupert sectional firebox was continuously tested under these severe conditions for 55 min. without developing any failure, notwithstanding the fact that the level of the water fell to a point more than 25 in. below the crown-sheet. The water gage-glass did not read below 25 in. The test was then discontinued because the small amount of water remaining did not evaporate sufficiently fast to supply the draft necessary to maintain the fire. At the conclusion of this test, the Jacobs-Shupert firebox was apparently in good condition, and ready for further service, as may be seen in Fig. 4.

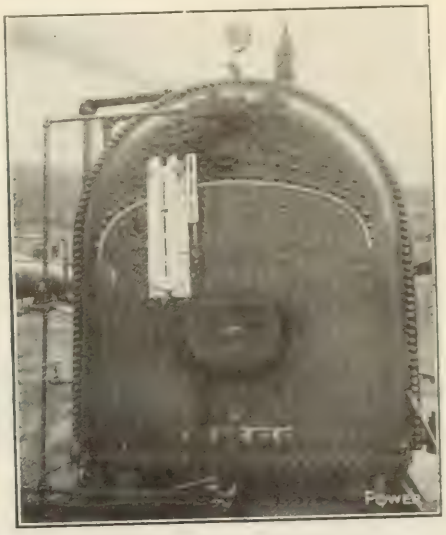


FIG. 2. GRADUATED WATER LEVEL GAGES READ THROUGH TELESCOPES AND PRESSURE

The ordinary radial stay boiler was then tested under conditions identical to those above described. After the test of the ordinary boiler had been in progress for 23 min., and the water level had fallen to 14½ in. below the crown-sheet, an explosion occurred. The crown-sheet and the stays, which hold it in place, having become highly heated, pulled away from each other and released the pressure in the boiler. The discharge of steam was through the firebox and the force of the explosion amidst clouds of steam and smoke, was sufficient to throw parts from the furnace in all directions

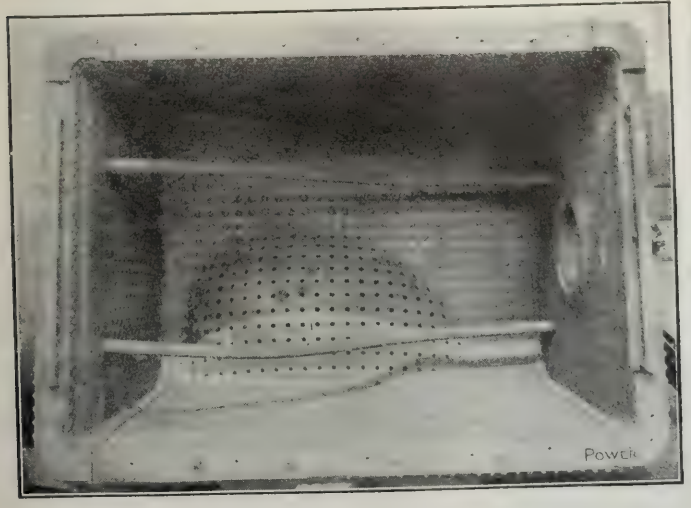


FIG. 3. INTERIOR OF RADIAL STAY FIREBOX AFTER LOW-WATER TEST

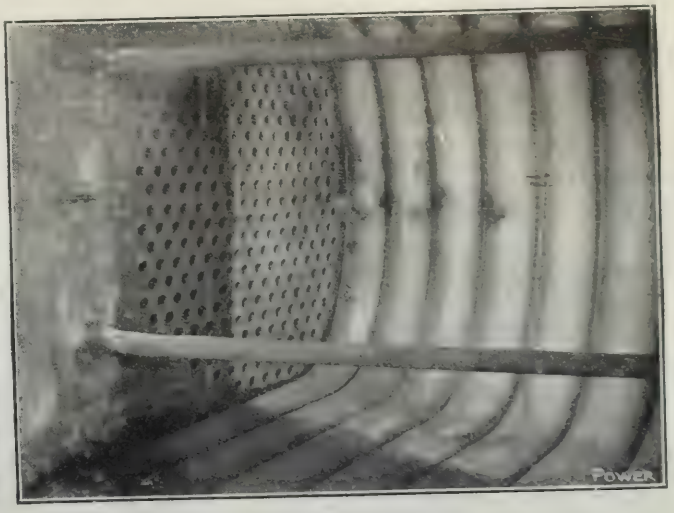


FIG. 4. INTERIOR OF JACOB-SHUPERT FIREBOX AFTER LOW-WATER TEST

for a considerable distance, and to lift the entire boiler, weighing 40 tons, several feet above its foundation. The damage to the boiler was such as to make necessary its reconstruction. Fig. 3 shows the failure of the crown-sheet and Fig. 5 is a view taken when the boiler exploded.

Some of his tests showed an evaporation as high as 50 lb. per square foot of heating surface. The efficiency of the Jacobs-Shupert boiler was a little better than that of the ordinary radial-stayed boiler against which it was tested.

The Lukens Iron & Steel Co., of Coatesville, has financed the Jacobs-Shupert

FIG. 5. EXPLOSION OF THE RADIAL STAY FIREBOX BOILER

The steam pressure during the Jacobs-Shupert test varied between 215 and 225 lb. for the first 27 min. and gradually dropped thereafter until it reached 50 lb. at the end of the test. The steam pressure on the radial boiler varied between 220 and 230 lb., being 228 lb. when failure occurred.

The Jacobs-Shupert firebox is constructed of U-shaped sections between which are riveted stay sheets as shown in Fig. 6. Previous tests by Prof. Goss show that from 45 to 50 per cent. of the evaporation occurs on the firebox surfaces with moderate capacity and some 35 per cent. when forcing.

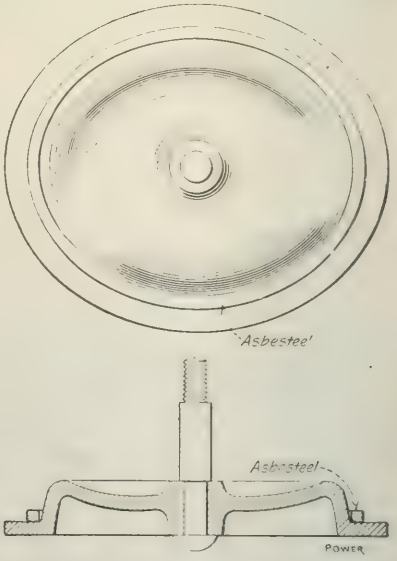
United States Firebox Co. and provided the means for carrying on these expensive and important tests.

Asbesteel Gasket

This gasket is made up of asbestos inclosed in a steel casing and in suitable sizes and shapes to fit the handhole and tube plates of water-tube boilers.

It is suitable to use as a union gasket and also for Van Stone joints. The illustration shows its application to a water-tube boiler tube plate.

As the gasket is compressible it is squeezed flat when brought under pressure between a handhole or tube plate and the boiler shell or header. The steel casing forms a metal-to-metal contact between the header and the tube plate; consequently there is nothing to adhere to either, and when the joint is broken, the gasket falls off, leaving a clean surface. This eliminates scraping



ASBESTEEL GASKET ON TUBE PLATE

and cleaning the plate or header. The gasket is manufactured by Faul & Timmons, 286 Ellicott Sq., Buffalo, N. Y.

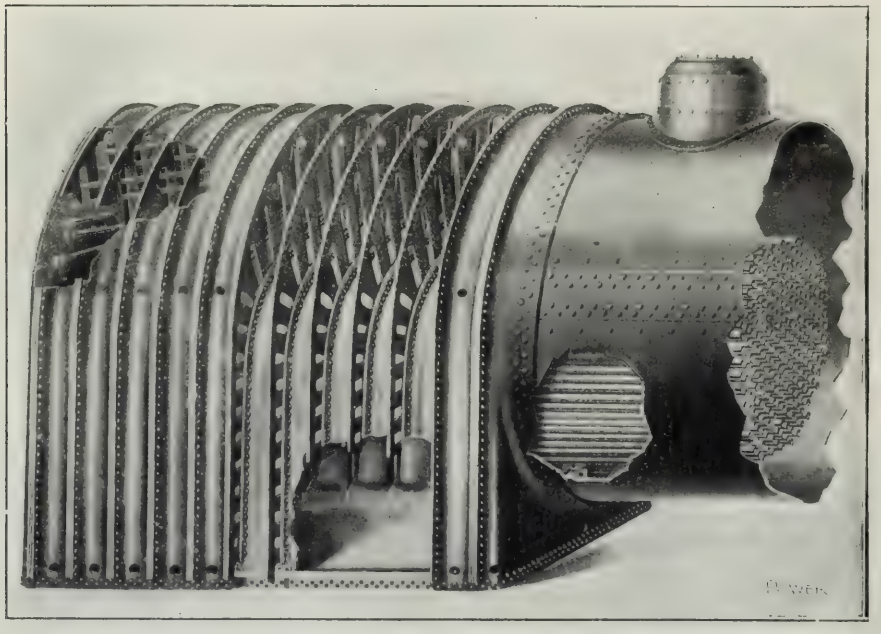


FIG. 6. BROKEN VIEW OF THE JACOBS-SHUPERT FIREBOX SHOWING CONSTRUCTION

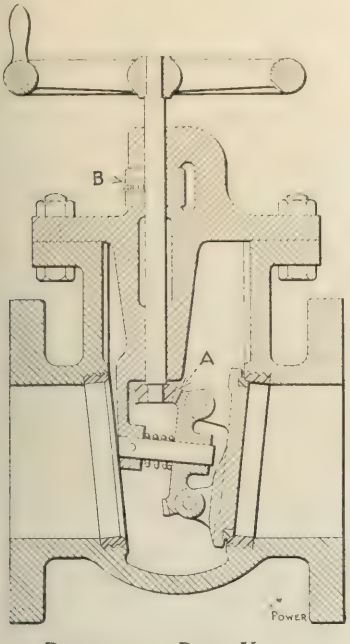
Regrinding Double Disk Gate Valve

This valve is a combination of the globe-valve seat and straightway body which can be reground without removing the valve from the line.

The double-disk feature, although not new, is so constructed that the valve can be used in any position without the faces of the disk and valve body constantly rubbing while being operated.

When regrinding, the bonnet and working parts of the valve are removed from

the body. The valve disks are detached from the head and, one at a time, are attached to the bevel gear *A*. The grinding fixture is attached to the body, as shown by two bolts, allowing the adjusting nut *B* to slip on the shaft. The disk will readily find its seat and the adjusting



REGRINDING DISK VALVE

nut is then tightened to allow the bevel gears *A* to run freely.

As the opening in the upper part of the body is but partially closed by the grinding fixture, the disk can be seen and plenty of room left for inserting the grinding mixture on the seat. This valve is the invention of F. K. North, Pottstown, Penn.

"Ideal" Corliss Valve Engine

One of the first of a new line of Corliss-valve engines built by A. L. Ide & Sons, Springfield, Ill., is shown on the testing floor in Fig. 1. The valves are placed in the cylinder heads, to reduce the port clearances and minimize steam losses.

The valve-gear is nondetachable, and the steam valves are operated by a positive valve motion, Fig. 2, which permits the valves to remain in a closed position for one-half of the revolution, while the eccentric is crossing the dead center and reversing its direction of travel. When the eccentric crosses the other dead center the valves which are balanced are open. The valves are moving at their highest velocity when the eccentric is approximately on the quarter and is imparting the maximum speed to the eccentric rod and connecting gear. They thus call for the greatest driving power when the most is available.

The multiported valve-gear links are inclosed and submerged in an oil-tight case, on the frame of the engine, dispensing with the connecting rocker shafts between the eccentric rods and the valve rods.

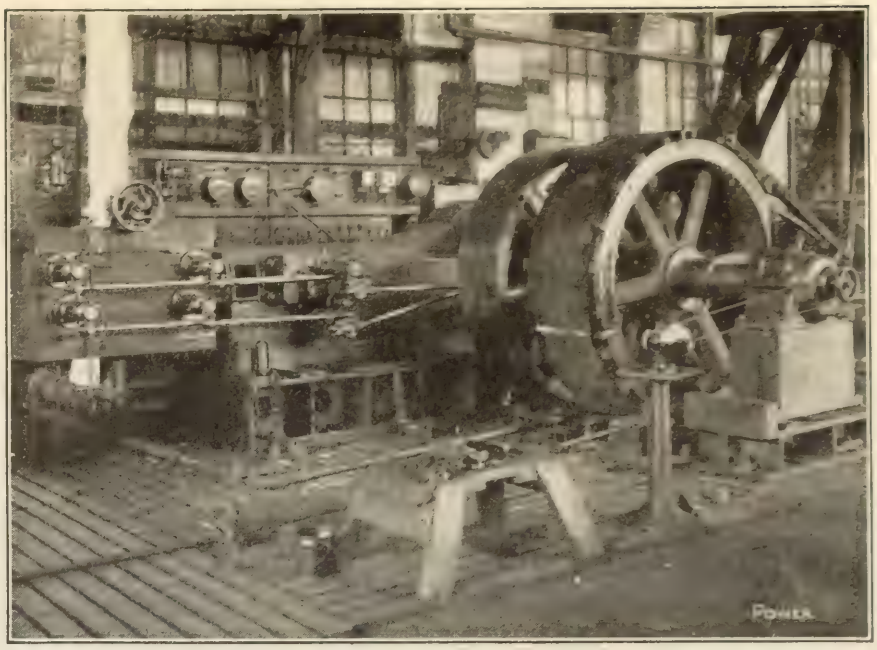


FIG. 1. "IDEAL" CORLISS VALVE ENGINE ON TESTING STAGE

The exhaust valves are driven by the ordinary wristplate motion because they have to be open during nearly one-half revolution and cannot remain motionless as long as the steam valves. The latter are controlled by a shaft governor and the exhaust valves by a fixed eccentric. The steam eccentric operates the lever *A*, Fig. 2, within the case by an external arm, shown in Fig. 1. The lever *A* actuates the pins *B* and *C* through links as shown. The links *D* slide through the guide pins *E* which are fitted in bearings in the valve-gear case and cover. Fig. 6 shows the details of the links *D* and guide pins *E*. The rotation of the pins *B* and *C* is transmitted to the valves through the external lever arms and rods, shown in Fig. 1.

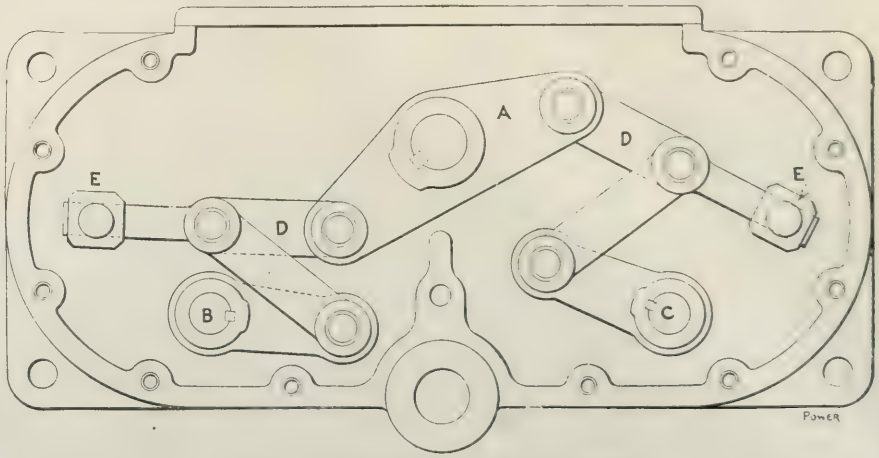


FIG. 2. VALVE GEAR LINK IN CASE

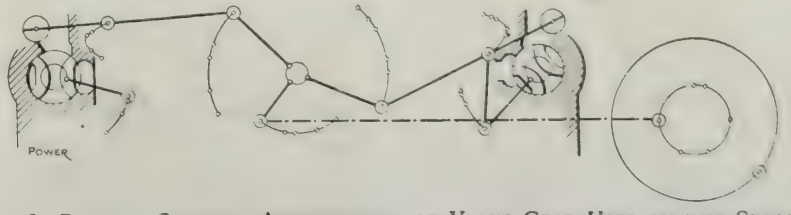


FIG. 3. DIAGRAM SHOWING ARRANGEMENT OF VALVE GEAR USED ON THE STEAM VALVE

In the valve-motion diagrams, Fig. 4, the outer circles at the right represent the crankpin path and the inner, the eccentric travel. In the top diagram the

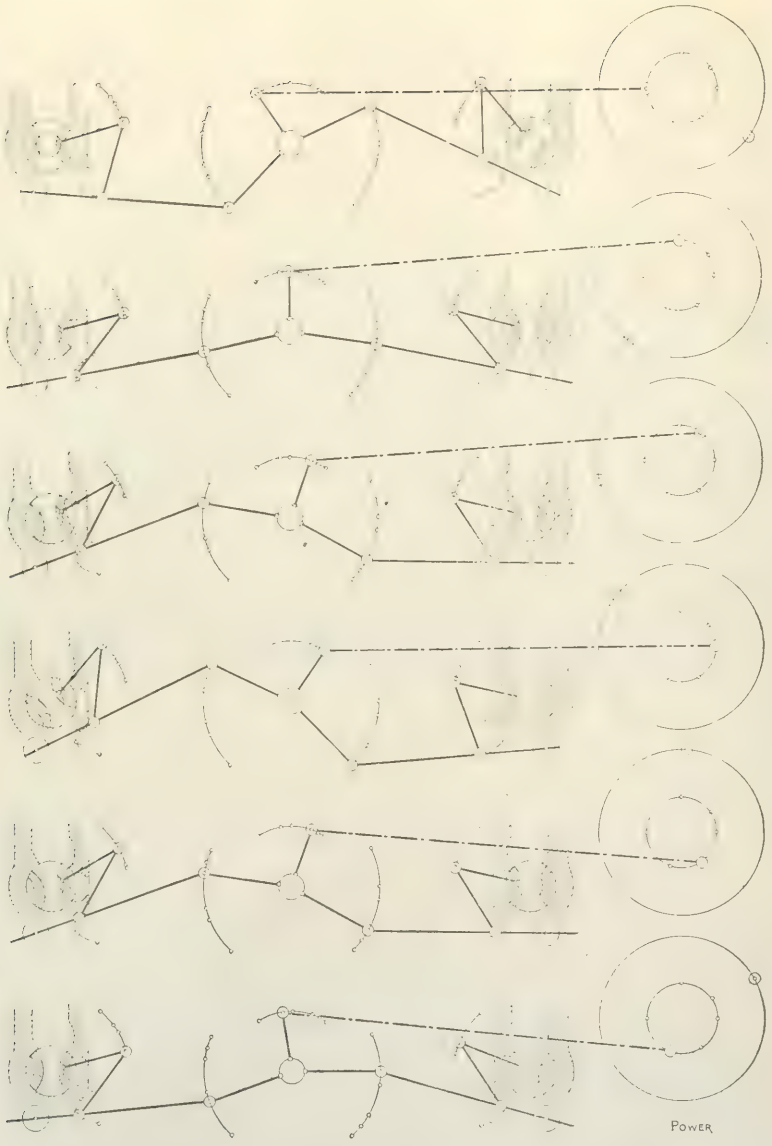


FIG. 4. VALVE GEAR MOTION DIAGRAMS

eccentric is on dead center, the head-end valve is closed and at rest and the crank-end valve is at maximum opening. The position of the valves and the gear at the various other points in the eccentric travel is shown by the other diagrams. The gear in actual use is inverted from that shown, as the gear will work in both positions, as illustrated in Fig. 3.

Typical maximum-load indicator diagrams from the 16x22-in. engine under test, are given in Fig. 7. The steam pressure was 147 lb. gage, speed, 200 r.p.m., and the mean effective pressures, head end, 109.75 lb., crank end, 97.29 lb. per square inch.

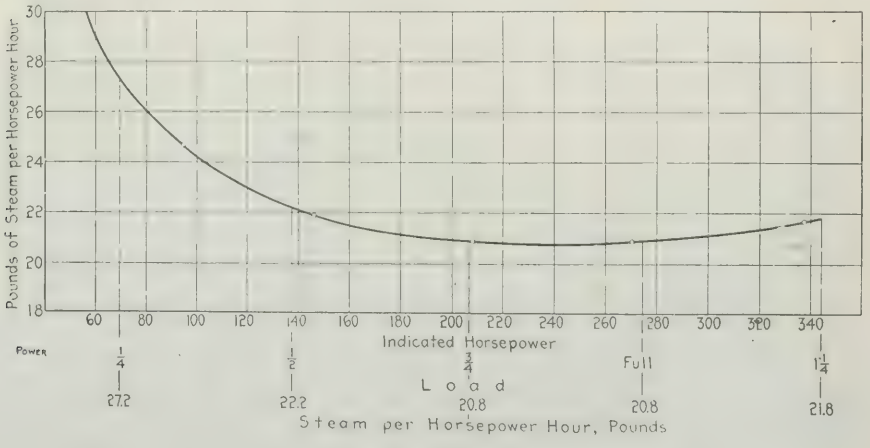


FIG. 5. STEAM CONSUMPTION CURVE OF TEST ENGINE

In Fig. 5 is shown the steam consumption of the test engine at various loads. The pressure was held as close as possible to 150 lb. gage throughout the test, not varying over 4 lb. above or below.

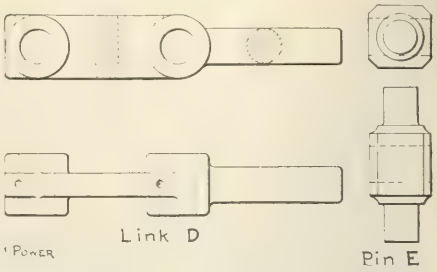


FIG. 6. DETAILS OF LINK AND GUIDE PINS

No correction for moisture was made in determining the steam consumption, as the steam was supplied by a water-tube boiler having no heating surface above the water line, and the superheater was flooded, to prevent superheating the steam. A separator was placed just above the engine throttle. As a further check against the steam carrying superheat a thermometer was inserted in the cylinder steam chest and the temperature and gage readings taken simultaneously every five minutes. In each instance the observed temperature was about 6 deg. below the corresponding temperature of saturated steam of the observed pressure. This difference is chargeable to radiation from the thermometer cup and the exposed length of the thermometer. The indicator diagrams and water records proved that the load was held practically constant, as the planimeter readings did not vary over 2 per cent. in any case and the water records checked equally close.

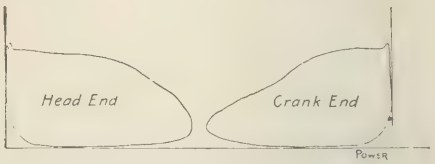


FIG. 7. MAXIMUM LOAD DIAGRAMS

Platt Iron Works Reorganization

In July, 1911, the Platt Iron Works Co., Dayton, Ohio, went into the hands of receivers and its affairs have since been administered by them. Under the plan of reorganization, recently indorsed by a large majority of its creditors, ample working capital for development and improvement will be provided. The necessary amount has been subscribed and is now available.

It is estimated that the net manufacturing profit, from July 24, 1911, to Mar. 1, 1912, without allowing for depreciation above ordinary repairs and maintenance

nance or for interest on investment, was $47,672.96, or about $79,000 per annum. It is believed that upon reorganization and under competent and aggressive management the volume of business can be largely increased and certain economies of operation effected which will materially increase the annual earnings of the property.

The failure was brought about through borrowing large sums, wholly or partially on the faith and credit of the company, no substantial part of which was used for its purpose or benefit. The present organization is believed to be a valuable asset which should be preserved, and in the judgment of the bondholders and creditors, their interests will be promoted through a reorganization rather than by the sale and liquidation of the properties, the dismemberment of the plant and the practical abandonment of organization and good will. The plan is designed to give the new company a working capital of approximately $300,000.

Lima N. A. S. E. Convention

On June 20, Lima Subordinate Association No. 21 of the National Association of Stationary Engineers began its state convention in Lima. It was intended to have a three-day session, but the convention adjourned at 12:15 o'clock on June 21, as many of the delegates were anxious to return to their homes.

The convention was formally opened at 10 o'clock on June 20 in Memorial Hall by Chairman J. F. Reed, of the local executive committee. The Rev. G. W. Watson delivered the invocation. Mayor C. N. Shook in welcoming the visitors spoke briefly of the great progress made by the stationary engineers and of the moral influence of the organization.

State President John J. Coughlin responded to the mayor. The association was not, in the usual meaning of the term, a labor organization, said Mr. Coughlin, but it was accomplishing its purpose of raising the stationary engineer to the highest standard, making him more valuable to his employer and promoting the safety of the public.

C. H. Wirmel, chief examiner of steam engineers, who represented Governor Judson Harmon, spoke of the advancement in stationary engineering through the efforts of the N. A. S. E., which had been instrumental in having license and inspection laws enacted. Other states, said Mr. Wirmel, are copying the Ohio laws. Under the influence, education and requirements of the organization, stationary engineering had been raised from a drudgery to a profession.

Prof. D. D. Ewing, of the Ohio Northern University, responded briefly. On behalf of the Lima Progressive Association, E. J. Maire, its president, also welcomed the visitors, expressing the appreciation of the business men of Lima for the honor of entertaining the engineers. Judson Pratt, a delegate, outlined the objects and high attainments of the association. He was formerly an organizer in Ohio and the N. A. S. E. representative when Lima Local No. 21 was first instituted. He referred to the heroism of the "Titanic" engineers. "These men," said Mr. Pratt, "did not seek an excuse to go on board the rescue ship 'Carpathia' —they went down with the 'Titanic' without leaving their posts of duty."

John Lawlor, a pioneer stationary engineer of Lima and one of the oldest members of Lima Local, welcomed the visitors on behalf of the Lima organization. Anthony Dietz, of Cincinnati, gave an original toast to the ladies of the auxiliary.

John Roberts, of Cleveland, a member of the state organization and the father of the stationary engineers' license law, dwelt at some length on the high objects and attainments of the order.

The business of the final session on June 21 was concluded by electing the following officers and selecting Toledo as the convention city for 1913:

T. T. Synett, Dayton, Ohio, president; John E. Radigan, Cleveland, vice-president; W. D. Spangler, Lima, secretary; John Wirmel, Cincinnati, treasurer; Charles Flenner, Hamilton, Ohio, conductor; B. A. Held, Toledo, doorkeeper; T. E. McFadden, Columbus, state deputy.

Exhibitors

White Star Refining Co., Crandall Packing Co., Strong, Carlisle & Hammond Co., Keystone Lubricating Co., Standard Oil Co., Diamond Power Specialty Co., Cling-Surface Co., Westinghouse Electric & Manufacturing Co., Vulcan Soot Cleaner Co., Knopf & Johnson, *National Engineer*, Ohio Grease Lubricant Co., C. E. Squires Co., Fairbanks Co., Quaker City Rubber Co., Automatic Steam Trap Specialty Co., Jenkins Brothers, Hooven-Owens-Rentschler Co., William Powell Co., American Steam Gauge & Valve Manufacturing Co., Lagonda Manufacturing Co., Greene, Tweed & Co., Garlock Packing Co., V. D. Anderson Co., Lunkenheimer Co., Clement Restein Co., Wickes Boiler Co., H. W. Johns-Manville Co., Dearborn Drug & Chemical Works, Hawkeye Boiler Compound Co., Anchor Packing Co., McLeod & Henry Co., Home Rubber Co.

OBITUARY

William Northrop

William Northrop, president of the Virginia Ry. & Power Co., Richmond, Va., died suddenly of heart failure on June 11, 1912, at his home near Forest Hill. Mr. Northrop was born in Pennsylvania 43 years ago. He was educated at Philadelphia and at the University of New York. Shortly after graduating, Mr. Northrop accepted a position with the Philadelphia & Reading R.R.

The Richmond Passenger & Power Co. and the Richmond Traction Co. were consolidated in 1902 and the control passed in 1902 to the Gould interests. Mr Northrop was assistant secretary-treasurer until 1904, when he and Henry T. Wickham were appointed the company's receivers. On July 1, 1909, the Virginia Ry. & Power Co. succeeded the Richmond Passenger & Power Co., Mr. Northrop being elected president. He was also an officer in several other companies. He is survived by a widow.

SOCIETY NOTES

The headquarters of the American Society of Heating and Ventilating Engineers during the Detroit convention on July 11 and 12 will be at the Hotel Tuller.

The Massachusetts State Convention of the N. A. S. E., which takes place this year at Northampton, July 11 and 12, promises to be larger than any yet held in the state. Earnest work by the various committees has perfected all of the arrangements.

PERSONAL

Frank Salomon, president of the Otto Gas Engine Works, Philadelphia, Penn., sails on July 4 for a six weeks' trip in Germany.

O. D. Dillman, formerly in the main office of the Hooven, Owens, Rentschler Co., Hamilton, Ohio, has been appointed assistant manager of the Chicago branch.

Edgar D. Sibley recently resigned as assistant to the superintendent of power of the Brooklyn Rapid Transit Co. and is now associated with W. G. Clark, consulting engineer, New York City.

D. L. Fagnan, for some time an erecting engineer with the De La Vergne Machine Co., has been assigned to the Cincinnati office of that company in charge of sales for the De La Vergne oil engine.

Paul M. Warburg, of Kuhn, Loeb & Co., New York City, was elected a director of the Westinghouse Electric & Manufacturing Co., at the annual meeting in East Pittsburgh. He succeeds Charles A. Moore, resigned.

S. G. Meek has been appointed assistant general manager of the electrical department of the H. W. Johns-Manville Co., New York City. He has been with the company as special representative in this department over 15 years.

Samuel Insull, president of the Commonwealth Edison Co. of Chicago, recently elected a director of the Pacific Gas & Electric Co., of San Francisco, was given a banquet at the Fairmont Hotel, San Francisco, on the evening of June 14.

Moments with the Ad. Editor

Charles H. Parson, treasurer of the Parson Manufacturing Company, sent us a letter recently which we are reproducing here because it is another definite example of some of the things we have been trying to point out on this page.

Mr. Parson wants a "prize."

Not because he needs it but because he is the kind of a man that when he goes into something in the shape of a contest he likes to win for the sport's sake.

He writes us as follows, with his letter headed—

"Does 'Power' advertising pay?"

"In response to your original suggestion of advertisements that attract attention, we at that time wrote you advising that we thought our advertisements were reasonably strong and attractive and gave our reason. You didn't send us any prize but it may interest you to know that the following are facts:

"In February we received an inquiry regarding the Parson [illegible] Ejector, to which we replied by mail as the point was nearly 2,000 miles distant. On receipt of our letter and booklet we were favored with a night telegram asking price and giving details. On March 14 we closed the order and on March 30 the material was shipped with working blueprints. May 5 the material had arrived and on June 4 in response to our statement, and a request for their advice as to whether they had any difficulties in erection, we received the following reply:

" 'Everything in, working satisfactorily. Will send check to cover in a few days.'

"Your Mr. Robbins was shown this letter which happened to arrive on my desk at the time of his visit, and can bear witness to the authenticity of the statement herein mentioned, although names have been purposely omitted because we have neither asked for nor received the consent of our customer to use the letters above referred.

"On reconsideration, if you find that we have won a prize, send it along."

We didn't offer any prizes, Mr. Parson, but if we had you would have undoubtedly won one with this letter.

We did offer to pay for those letters which were printed and this one will be paid for.

You readers of "Power" may not know it, but one of the hardest things a publisher has to do is to extract definite information from an advertiser as to results received.

Mr. Parson didn't send us this letter with an idea of getting a prize, or being paid for it. He sent it because he was broad enough between the eyes to know that this testimonial to "Power's" pulling power would be appreciated and would be encouraging to the publishers of the paper.

There are two strong points brought out here, one for readers and one for advertisers.

In the first place, a concern 2,000 miles distant from the manufacturer found exactly what they were evidently looking for in the way of an ash ejector. Their night telegram and the rapid closing of the sale is evidence that they were in *need* of such a device. Their need was supplied through "Power" advertising. And they were just as well satisfied to find the advertisement as the concern was to sell the goods.

The goods are pronounced by them "satisfactory." Both parties to the transaction benefited. Way off from this part of the world some subscriber of "Power" has seen the wisdom of reading the advertising pages and has confidence in advertised goods. His confidence was not misplaced.

The other point of view—that of the advertiser—is this:

A large piece of power plant apparatus has been sold through the mails without the aid of personal salesmanship. As a general rule this is a thing which cannot be done. But, in this instance it was successfully accomplished because the live wire who is a subscriber to "Power" knew that he could gamble on anything which appeared in the advertising columns of this paper.

Remember that it pays to read advertising and buy advertised goods, because only reliable goods can be continuously advertised.

An advertising bluff is called quicker than any other.

POWER

Vol. 36 NEW YORK, JULY 9, 1912 No. 2

For Example

On taking charge of a saw-mill plant having one 60 in.x12 ft. boiler and a 12x18 in. slide-valve engine, the owner assured me that everything was in good condition. I soon found, however, that the steam had been down for a half hour out of every two for a month. On Sunday I began work by "punching" soot from the flues and cleaning the combustion chamber, and at 5 a. m. Monday I put in my first fire. I used the mill refuse for fuel, and in an hour had 95 lb. of steam.

I made up my mind that the fault was not all in the boiler, but I had been too busy the day before to give attention to the engine. When I started it I wished that I had, as the sound of the exhaust was exceedingly loud and every joint seemed loose. When the sawyers stalled the engine by making a large cut I promptly shut down and refused to start again until I had an opportunity to overhaul it. This angered the foreman and, to my surprise, the owner also. He finally said: "You can do as you like, about it, but you can't spend a cent for new material."

This letter, recently received, recounts an experience that's regrettably common. It is one of the many instances where an ignorant owner had an intelligent engineer—several of them, in fact—but rather than take his engineer's advice he purchased central-station current, thereby increasing his power costs and wasting fuel which cost him practically nothing.

Unfortunately, the reasons for changing over are not always made known to the public and the engineer of the isolated plant is blamed. This letter should convince the hesitant owner that the engineer usually knows best how to operate the plant and should be consulted.

With a bar I could raise the crosshead $\frac{3}{8}$ in. in the guides; the pillow block had been hot and melted part of the babbitt; the flywheel was loose on the shaft, and the brasses had been severely burned. I scraped and babbitted the brasses and pillow block, tightened the guides and set the valve, but on starting, the engine was almost as noisy as before.

Then I took off the cylinder head and found the piston loose on the rod, and after much arguing I was allowed to get a new rod. I thought my troubles were over, but on starting again the pounding was still very severe. I calipered the pins and found them flat, and told the owner I must have new ones, but he refused to buy them.

The 4-in. exhaust pipe ran out through the side of the mill, but instead of using an elbow to deflect the steam to the ground, a bend was made, the inside of which buckled, so that there was only a $\frac{3}{4}$x2 in. opening.

As the management would not support me in my efforts to put the plant in running order, I left. I had several successors, but eventually central-station service was used. They still have to keep steam on the boiler seven months in the year for steaming cheesebox hoops and two or three days a week they have to carry 100 lb. All this requires the service of a fireman. The surplus sawdust is allowed to run into the river and is an absolute waste of good fuel.

Thirty dollars would have put that engine in good condition, and with fuel costing practically nothing, it is evident how much the owner could have saved by running his own plant. Under the circumstances it would seem as though the engineer's advice should have been given a little more consideration.

Coal and Ash Handling at Sioux City

A complete system for the power plant with facilities for saving the small coal which drops through the grate and returning it with the raw coal to the furnace.

The plant of the Sioux City Service Co., at Sioux City, Iowa, has been partially rebuilt and equipped with an unusually complete system for handling coal and ashes. The boiler and engine rooms are side by side under the same roof. The former is about 77 ft. wide and 131 ft. long. The boilers are in two rows along the side walls and face toward the center of the room. A large serving space between the boilers allows withdrawing the chain-grate stokers. The boilers, of Stirling type in 400-hp. units, total 3300 hp. The stack is at one side of the room and is central to the boilers. On the other side, directly opposite the stack, is the water tower.

The railroad track, on which coal is received and to which ashes are delivered, runs past the ends of the engine and boiler rooms, thus coming in at right angles to the conveyor system, which is an ideal arrangement. Provision has been made for storing a certain amount of coal on the ground outside of the building which may be used in an emergency, but the main storage of 850 tons is in steel bins above and midway between the two lines of boilers. The coal from the bins is fed through valves and chutes directly to the stoker hoppers and no hand labor is required.

To provide storage for ashes, the conveyor system is continued over the coal track and an ash storage of 140 tons is afforded by a steel bin just outside the boiler room. The ashes from this bin pass through valves and chutes to the same car in which the coal is received.

One of the features of the system is the saving of small particles of coal from the chain grates. In some installations this fine coal has been reclaimed by shoveling it back into the stokers from a small receiving pit in the floor, but more often it has gone into the ashes. In the present case hoppers have been so arranged underneath the stokers that they receive practically all of the fine coal which is of value and yet allow space at the end of the grates for the ashes to fall into the pit beneath. When a quantity of coal has accumulated in the hoppers, it is fed slowly into the conveyor with the raw coal from the receiving hopper. This mixes the finer particles of coal dust and partially burned coal with the raw coal and enables the former to be used advantageously in the stokers.

The coal- and ash-handling system, shown in outline in Fig. 1, consists of a pivoted bucket carrier which runs underneath the coal-receiving hopper and through the basement, thence ascending to the upper runway, discharging the coal to the coal bins and the ashes to the ash bin and descending outside of the building to the receiving hopper.

The coal from the cars is dumped into the receiving hopper and then fed by a reciprocating plate, or feeder, to a crusher underneath the feeder, or is bypassed to a chute beyond the crusher and delivered directly to the conveyor. This avoids running the crusher when slack coal is available. The power for operating the feeder is furnished by the conveyor itself. The arrangement of the feeder and crusher is shown in Fig. 4. The opposite

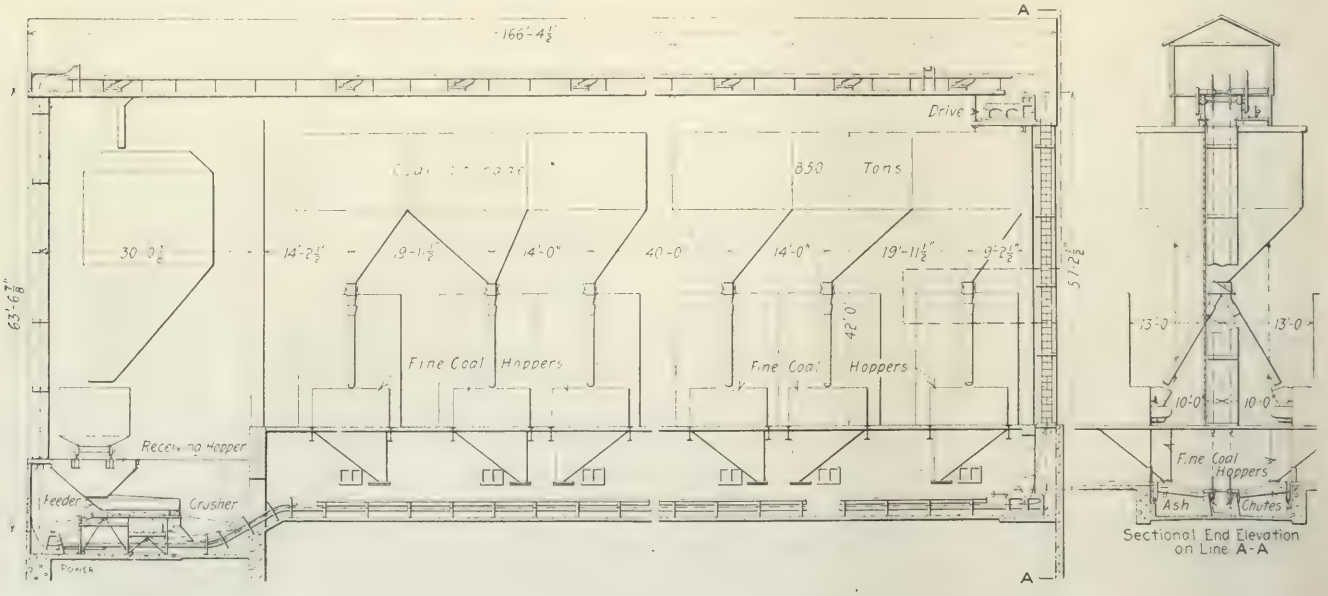


FIG. 1. SECTIONAL ELEVATIONS OF BOILER ROOM, SHOWING CONVEYING SYSTEM

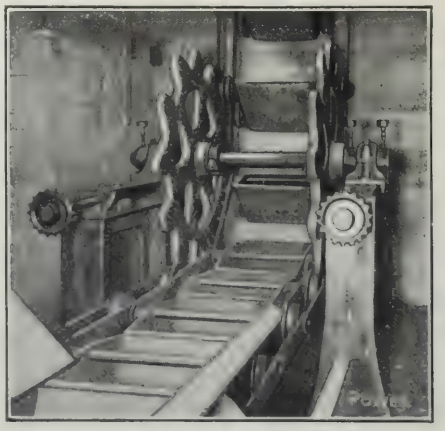


FIG. 2. END OF LOWER RUN OPPOSITE CRUSHER

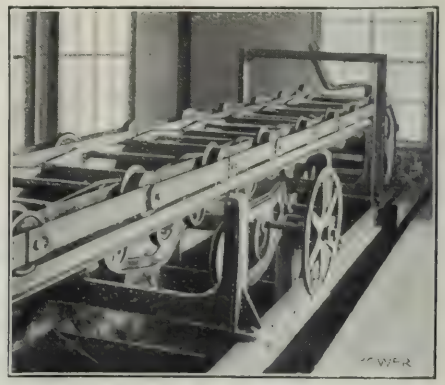


FIG. 3. UPPER RUN, SHOWING TRIPPER OVER ASH CHUTE

FIG. 4. FEEDER AND CRUSHER

end of the lower run of the conveyor, together with the take-ups and general type of construction, is shown in Fig. 2.

The equipment in the basement deserves special mention. This is seen in section *AA*, Fig. 1. The ash chutes traverse the full length of the space in front of the ashpits and fine-coal hoppers and may be placed as desired by the operator. The ashes are raked from the pits into these chutes and thence into the conveyor, no shoveling being required. These same chutes are also used to transfer to the conveyor coal dust from the fine-coal hoppers.

The capacity of the conveyor is 50 tons of coal per hour and 10 hp. is required for its operation. Considering the length of runs and the lift, this is low, but is due in great measure to the large self-oiling chain rollers which turn on extra-large case-hardened bushings, and because there is no scraping or dragging action, the material being carried throughout its entire course. The drive, which is placed at the upper end of the ascending strand, is shown in Fig. 5, and the opposite end of the upper run, with the tripper or dumper over the ash chute, is shown in Fig. 3.

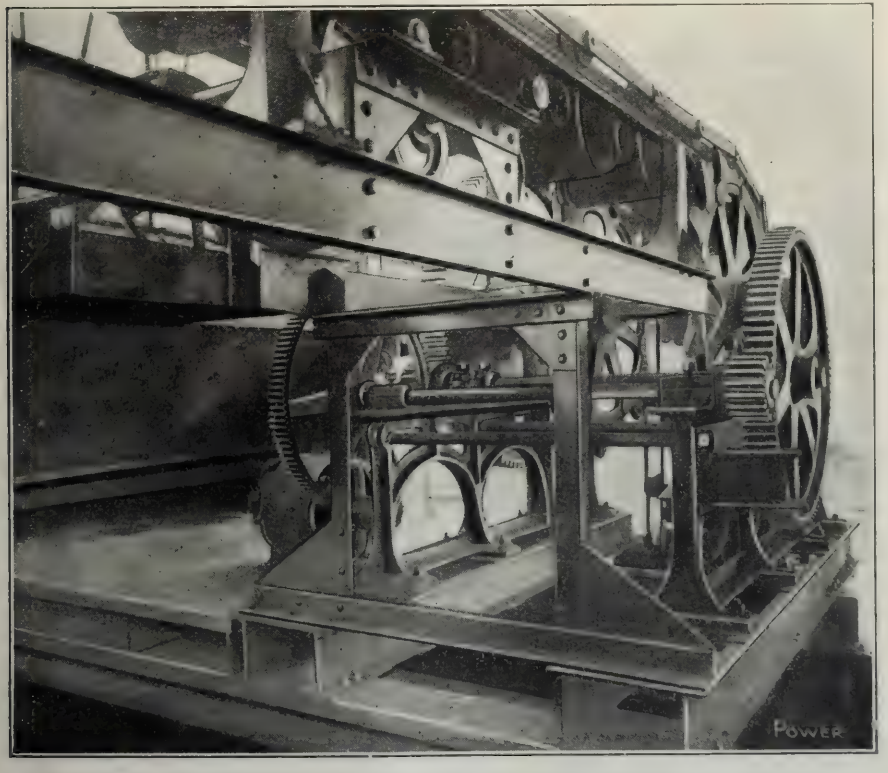


FIG. 5. DRIVE AT UPPER END OF ASCENDING STRAND

Bracing Flat Surfaces in Steam Boilers

There is considerable variation as to the load allowed per square inch of net section on diagonal braces, rod braces and staybolts by the authorities who have laid down rules on this subject. The United States Government rules allow 6000 lb. on welded iron stays below 1¼ in., 7500 on 1¼ in. and above, and from 7000 to 9000 lb. on weldless steel stays. Chicago has a flat scale of 6000 lb. on all stays or braces, Philadelphia has a limit of 7500 lb.; the Massachusetts rules allow from 6500 to 9000 lb. per square inch net section, varying as the braces are welded or weldless and with the size, the latter for the reason that with a given waste of material the percentage of reduction is greater with the smaller sizes.

The above applies to flat surfaces and refers to flat heads, such as dome heads, segments of heads, etc. The United States Government has a rule to find the pressure on flat heads not exceeding 20 in. in diameter as follows:

$$P = \frac{C \times T^2}{A}$$

where

P = Pressure;
C = 112 ($\frac{7}{16}$ or under) and 120 over $\frac{7}{16}$);
A = ½ the area;
T = Thickness in sixteenths.

With a ¾-in. head 20 in. in diameter, 110 lb. would be allowed by these rules.

A short time ago the Board of Boiler Rules for the State of Ohio issued instructions to the inspectors holding certificates of competency that the following formula could be used in flat surface of heads:

$$P = \frac{T \times T.S. \times 10}{\quad \times 8}$$

where

T = Thickness;
$T.S.$ = Tensile strength;
A = Area;

In addition, a limitation allowance of 8000 lb. per square inch irrespective of size of brace is granted. This applies to boilers now in use, but not to boilers to be installed after July 1, 1912. This ruling is far more liberal than any other authority has heretofore allowed as a comparison will show. Assume a 72-in. boiler, height of segment 24 in., thickness of head ½ in., tensile strength 60,000 lb. The total area of the segment = 1186.4

sq.in., while the area requiring bracing = 814 sq.in. Hence

$$\frac{0.5 \times 60{,}000 \times 10}{814 \times 8} = 46 \text{ lb.}$$

allowed without braces.

Let the pressure required be equal to 100, then 100 — 46 = 54 lb. to be braced, and 54 × 814 = 43,956 lb. Assuming the proposed brace to be of 0.79 in. area, then 0.79 × 8000 = 6320 lb. per brace, and 43,956 ÷ 6320 = 7 braces of practically 1-in. diameter.

It may be said that flat surfaces subjected to internal pressure will spring, and proportionally to the unsupported area. Samuel Nichols, in his tests of circular flat heads, showed the springing began with very low pressures even at 20 lb. on 28-in. heads and increased as the pressure was raised. Applying this fact then to the Ohio ruling, it seems the head would so spring that at 100 lb. pressure the total load on the braces would be 0.79 × 7 = 5.53 into the total load, 81,400 lb., or 14,718 per square inch net section instead of 8000 lb. Applying this ruling to a flat dome head 36 in. diameter, ½ in. thick, 60,000 lb. tensile strength, area to be braced 707 in., gives 53 lb. without bracing. The results of allowing a flat head unbraced to spring and return times without number would be final failure due to such action.

Reverting to the segment as in a horizontal tubular boiler, it may be said other authorities have been careful to avoid allowing excessive stresses on the chord of the segment which is supported by the tubes inasmuch as the latter are not a constant in strength as is the flange of the head in the arc of the segment, and this view has been approved by most students as the tubes are subjected to more or less rapid wear and reduction in thickness.

Further, such calculations apply to boilers now in use irrespective of age. Indeed, Ohio has no limitation as to age as respects pressure to be determined with a factor of safety of 4 together with this exceedingly liberal allowance on braces. Comparing this with the Massachusetts, Chicago, Philadelphia and Detroit rules, what results may be expected?

Preparing for License Examination

By W. Nelson

This man worked out problems based on conditions existing in his power plant. He learned to "do things" by doing as well as by reading and finally won out.

Many engineers have criticised, justly and otherwise, license examiners and examinations, and occasionally a list of examination questions is submitted with the purpose of aiding others, but no one, to my knowledge, has proposed a plan of preparation for meeting the examiner.

My method was no short cut; it did not eliminate hard work, but it was so effective that every examination yielded the license sought.

Learn By Doing

I soon realized that one best learns by doing; that there is nothing like actually performing an operation to impress it upon the mind. Reading directions for babbitting crankpin brasses gives some ideas, but to make a lasting impression, start right in and clean, babbitt, bore and scrape them. When the job is completed, one knows how to do the work.

When I began studying boiler computations, I measured the shells and tubes of the boilers I was firing, calculated the heating surface, computed the efficiency of the joints and figured the allowable pressure. The coal slips were then added up and the coal burned per square foot of grate per hour calculated, and then I determined the amount of air necessary to burn that amount of coal.

The horsepower of every engine in the plant was computed and some in other plants. One engineer allowed me to take diagrams from his engine, loaning me his planimeter to measure the area.

I determined the cylinder ratios of the engine, the total expansion at several cut-offs, the size of condensers required and the amount of cooling water, the gain from heating the feed water, and the size of safety valves required; many other things I learned to do—by doing. When I learned how to work a problem, the idea was fixed in my mind by applying it to the apparatus about me.

Experimented on Old Engine

One day in my early training after reading how to put an engine on its centers, I was allowed to practice on an idle auxiliary engine. I made a tram, and after my watch was over I started my work. A month's reading could not have fixed the scheme in my mind as firmly as it was fixed in the next hour. The chief was interested, and had me put the engine on each center and mark each position on the guide. Then he took down the rod and showed me how to find the striking points and the clearance and how to assemble and key up the rod. In learning one thing well I also learned several others, and for keeps.

An acquaintance who had an old engine let me use it for practising valve setting, my watch engineer furnishing the instruction. After setting the valve several times, we took out the brasses and scraped them, keyed up and made a slight change in the clearance and several other adjustments. The engine was put in shape and I got some very practical instructions out of this day's work.

Formulas

To formulas I applied the principle of learning by doing. I selected some of the simplest and applied values from concrete examples about me. After mastering these, I gradually worked up to the more complex ones and soon found that my former fears were foolish, that the formula was a very useful servant. All that is necessary is a little patient work.

Many gain little from their reading because they shun the hard spots and the mathematics; they swallow things whole and accept statements without question. The man who challenges a statement until he proves it, conquers the tough passages and solves problems, gets something of value for the time spent.

Data Sheets

One chief gave me the idea of gathering data sheets of the plant apparatus. Upon his desk is a data-sheet folder. Each boiler, engine and device has a sheet. If he wants to order a repair part its dimensions are found on the card.

When I took charge of a plant I remembered these things and started a data book, taking what measurements I could with things running; the rest I got while the plant was shut down. The pumps and boilers received the same attention; in fact, everything of importance about the plant that I could think of was jotted down for the data sheets.

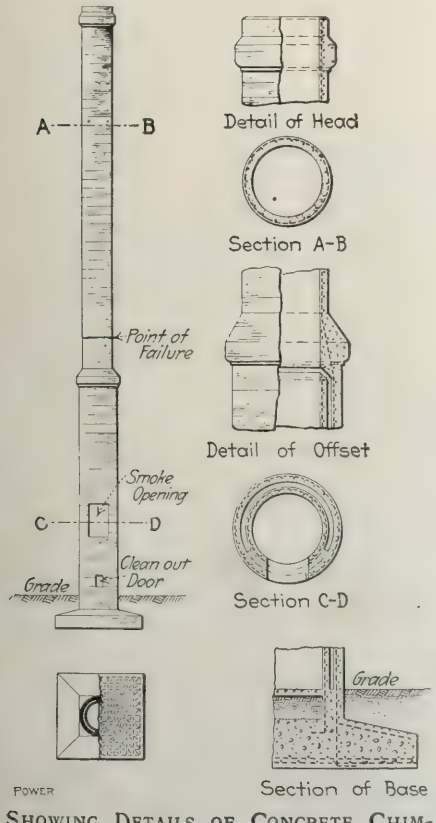
It meant work, but I was learning a lot that I otherwise would never have known. When I had opportunity I studied the data sheets. Working from the cylinder, I computed the required size of wristpin, crankpin, connecting-rod, etc., by several formulas. Next, using the shaft, pins and connecting-rod as a basis, I figured out how large a cylinder they would carry. The radiator surface required for the several rooms was computed by different methods and the results compared. The stress upon the boiler head above the tubes, the allowable pressure, and various other things about the boiler and other apparatus were computed. These sheets furnished a fine opportunity to learn by doing.

Another chief said: "Don't hurry or worry about a license. Try to fit yourself to be a real engineer. If you do that, the license will come." I took a course in a correspondence school which proved profitable and enjoyable. Engineering journals have been constant helps. One day I took up the questions in the questions and answers department, and before reading the answers I worked out ones of my own; then I compared them with those published.

To get a varied experience one should not work too long in one plant. Changing gives the engineer a broader view from contact with machinery and men. In a small plant, six months might suffice, but a year would be better. In a large plant, two years can be spent to good advantage. Visit plants in your vicinity and study them carefully.

Wind Wrecks Concrete Chimney

A cyclonic wind storm wrecked the 175-ft. reinforced-concrete chimney of the Pennsylvania Cement Co., Bath, Penn., at about 10 o'clock Sunday night, June 2, and caused the death of three men.



SHOWING DETAILS OF CONCRETE CHIMNEY CONSTRUCTION AND POINT OF FAILURE

The chimney was 8 ft. outside diameter from the lower double-shelf section to the top. The point of failure was 65 ft. from the base and the chimney was apparently sheared off at that height by the excessive wind pressure. The illustration shows the general design of the chimney and the point of failure.

From information obtainable at this writing it seems that the wind was exceptionally strong, sweeping a path approximately ¼ mile wide, uprooting trees, unroofing houses and also freight cars which were standing on a siding. From the appearance of the remaining portion of the stack and the fallen parts, the upper section, approximately over 100 tons, after breaking away, was blown sideways a distance of 30 ft. before it landed on the roof of the concrete boiler house, the striking point being indicated by a semicircular hole in the roof, showing that the fractured stack struck the roof an end blow, when the stack crumbled into small pieces. The concrete appeared to be porous and weak.

In breaking through the roof, a T-fitting on the main steam header was cracked and the 5-in. flanged saddle on one of the boilers was broken. The firemen remained at their post and immediately pulled the fires from under the nine 250-hp. boilers which the stack served.

They did not know of the death of their fellow workmen until after this work had been accomplished. It was then found that one fireman had been killed by some flying object and that two ash handlers had died from the effects of inhaled steam. None of the other firemen were injured and the property loss, outside of the chimney, will be small.

This chimney was built by a chimney construction company, now out of existence, and was finished in May, 1907. According to officials of the concrete company, there was trouble from the start regarding the methods of construction. Two rings of the stack above a height of 125 ft. were built during freezing weather. The work was stopped until spring and then, instead of removing the frozen rings, the remainder of the stack was built upon them. Because of this or some other cause the chimney was out of plumb 6 in. within 30 days after its completion. The chimney was plumb, however, for 125 ft. above the base.

This type of stack construction has been discarded by all concrete chimney builders both as to design and method of mixing the concrete material.

Recording Liquid Level Gage

There are several points of difference between this gage and that described in our issue of May 14. The pressure tube is so compact, being made with few diaphragms, that it can be located in the ordinary circular case, as shown in Fig. 1, and requires very little volume change and air movement to produce the indication. The movement of the diaphragm is restrained upon one side by one or more helical springs, two of which are shown at *SS* in the illustration, instead of by a flat metal strip, and the pen-arm is carried upon a shaft supported at both ends, the motion caused by the action of the pressure in the tube being transmitted to the shaft through a series of simple levers.

In the helical form of tube used for higher ranges the pen-arm is also fast-

FIG. 1. RECORDING LIQUID LEVEL GAGE

ened to the shaft, the movement being transmitted by a connection which is flexible radially. A friction joint is provided in the pen-arm to enable any necessary adjustment and this joint is in the line of the axis so that any change affects the adjustment of the pen on its proper arc without shortening the pen-arm or disturbing any other adjustments.

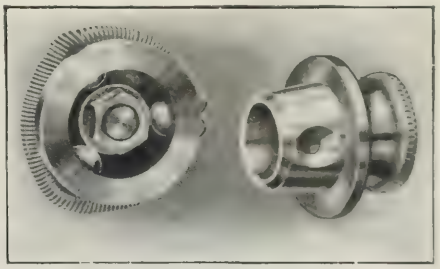


FIG. 2. SELF-CENTERING CHART HOLDER

The tension head of the chart holder, shown by Fig. 2, is tapered so as to automatically center the chart, and when removed leaves a flat surface on the dial, facilitating the putting on and taking off of the charts without disturbing the pen-arm or any other working part. The chart is held friction-tight between the upper and the lower surfaces of the hub and can be readily adjusted to the time position.

Skeleton and bold figures distinguish the day and night hours, as shown in Fig. 1. The clock movement has a 3-point support eliminating the possibility of strain from imperfect seating upon the case. All the cases are made dust and moisture proof. The Industrial Instrument Co., Foxboro, Mass., is the manufacturer.

Superheated Steam in Locomotive Service

In bulletin No. 57 issued by the University of Illinois Engineering Experiment Station, Prof. W. F. M. Goss reviews his previous work on the above subject carried on in the laboratory of Purdue University. The results of the study are given in the following paragraphs, and the bulletin giving the details of the tests may be obtained by addressing the author.

Foreign practice has proved that superheated steam may be successfully used in locomotive service without involving mechanism which is unduly complicated, or difficult to maintain.

There is ample evidence to prove that the various details in contact with the highly heated steam, such as the superheater, piping, valves, pistons and rod packing, as employed in German practice, give practically no trouble in maintenance; they are ordinarily not the things most in need of attention when a locomotive is held for repairs.

The results of tests confirm, in general terms, the statements of German engineers to the effect that superheating materially reduces the consumption of water and fuel and increases the power capacity of the locomotive.

The combined boiler and superheater tested contains 943 sq.ft. of water-heating surface and 193 sq.ft. of superheating surface; it delivers steam which is superheated approximately 150 deg. The amount of superheat diminishes when the boiler pressure is increased, and increases when the rate of evaporation is increased, the precise relation being

$$T = 123 - 0.265\,P + 7.28\,H$$

where

T = Superheat in degrees F.,
P = Boiler pressure by gage,
H = Equivalent evaporation per foot of water-heating surface per hour.

The evaporative efficiency of the combined boiler and superheater tested is

$$E = 11.706 - 0.214\,H$$

where E is the equivalent evaporation per pound of fuel and H is the equivalent evaporation per hour per foot of water-heating and superheating surface.

The addition of the superheater to a boiler originally designed for saturated steam involved some reduction in the total area of heat-transmitting surface, but the efficiency of the combination, when developing a given amount of power, was not lower than that of the original boiler.

The ratio of the heat absorbed per foot of superheating surface to that absorbed per foot of water-heating surface ranges from 0.34 to 0.53, the value increasing as the rate of evaporation is increased.

When the boiler and superheater are operated at normal maximum power, and when they are served with Pennsylvania or West Virginia coal of good quality, the available heat supplied is accounted for approximately in Table 1.

TABLE 1. AVAILABLE AND LOST HEAT

	Per Cent.
Absorbed by water	52
Absorbed by steam in superheater	5
Utilized	57
Lost in vaporizing moisture in coal	5
Lost in CO	1
Lost through high temperature of escaping gases	14
Lost in form of sparks and cinders	12
Lost through grate	4
Lost through radiation, leakage and unaccounted for	7

Water consumption under normal conditions of running has been established as in Table 2.

TABLE 2. WATER CONSUMPTION

Boiler Pressure, Lb.	Corresponding Steam per I.hp.-hr., Lb.
120	23.8
160	22.3
200	21.6
240	22.6

The minimum steam consumption for the several pressures is materially below the values given. The least for any test was 20.29 lb.

Table 3 gives the established coal consumption under normal conditions of running.

TABLE 3. COAL CONSUMPTION

Boiler Pressure, Lb.	Coal Consumed per I.hp.-hr., Lb.
120	3.31
160	3.08
200	2.97
240	3.12

Neither the steam nor the coal consumption is materially affected by considerable changes in boiler pressure, a fact which justifies the use of comparatively low pressures in connection with superheating.

TABLE 4. COMPARATIVE ECONOMY OF LOCOMOTIVES USING SATURATED AND SUPERHEATED STEAM

Saving in Water Consumption				Saving in Coal Consumption			
	Locomotive				Locomotive		
Boiler Pressure, Lb.	Saturated Steam, Lb.	Superheating, Lb.	Gain, Per Cent.	Boiler Pressure, Lb.	Saturated Steam, Lb.	Superheating, Lb.	Gain, Per Cent.
120	29.1	23.8	18	120	4.00	3.31	17
160	26.6	22.3	16	160	3.59	3.08	14
200	25.5	21.6	15	200	3.43	2.97	13
240	24.7	22.6	9	240	3.31	3.12	6

Contrary to the usual conception, the conditions of cutoff attending maximum cylinder efficiency are substantially the same for steam superheated 150 deg. as for saturated steam. With superheated steam, when the boiler pressure is 120 lb., the best cutoff is approximately 50 per cent. stroke, but this value should be diminished as the pressure is raised, until at 240 lb., it becomes 20 per cent.

Tests under low steam pressures, for which the cutoff is later than half stroke, give evidence of superheat in the exhaust. The saving in water consumption and in coal consumption per unit of power developed, which was effected by the superheating locomotive, Schenectady No. 3, in comparison with the saturated-steam locomotive, Schenectady No. 2, is given in Table 4.

The power capacity of the superheating locomotive is greater than that of the saturated-steam locomotive.

Condenser Performance and Economical Vacuum

The report of the Prime Movers Committee of the National Electric Light Association treats the above subject as follows:

Radical changes have been made during the past two years in condenser design, the effort being to so arrange the tubes and baffle the flow of steam that the steam side of the tubes will be readily freed of condensation and that the entrained air will be swept toward the dry vacuum pump suction and over a nest of special tubes designed to condense any remaining steam in the air, thus reducing the volume of the vapor handled by the pump; but probably the greatest improvement has come about through a realization that additional cooling surface in a condenser is of little value unless ample provision is made for the steam to reach it without undue loss of pressure.

The accompanying curve shows how the theoretical thermal efficiency of a perfect steam engine varies with terminal pressures and indicates that the most promising field of endeavor for increased efficiency in steam-turbine work lies in the production and maintenance of the highest vacuums obtainable. The reduction of terminal pressure from 0.4 lb. to 0.2 lb. indicates an addition of about 8 per cent. in the theoretical amount of energy available, and this saving is double the amount which would be saved by a reduction in pressure from 0.6 lb. to 0.4 lb.

It has often been demonstrated that the vacuum may be improved by removing many of the tubes to provide liberal passages for the steam to distribute among the remaining tubes without excessive velocities or loss of pressure.

Another advantage to be obtained by the opening of steam passages and a reduction of steam velocities is the elimination of the whirling and creeping of tubes, which result in their becoming loose in the packing and even in being worn through in the tube plate.

The importance of this is apparent when it is considered that a 12,000-kw. turbine, when running at full load and with a condenser pressure of ½ lb. absolute pressure, is discharging into the condenser each second 6000 cu.ft. of steam; and if in the first pass of the condenser there is between all of the tubes of the first row only 10 sq.ft. of passage, the steam must have a velocity of 600 ft. per second, which can only be attained at the expense of a considerable amount of back pressure on the turbine, with a corresponding loss of efficiency.

Where condensing water is plentiful, but so dirty that the amount flowing through the condenser is restricted, due to the tubes being choked, relief may be obtained by the installation of grid-iron valves so located that the path of the circulating water may be readily changed from 3-pass to single-pass, thus reducing the friction and increasing the amount of cooling water.

Life of Condenser Tubes

There has been very little development during the past year on this subject. One large company reports that it is experimenting with tubes of different compositions, the two principal types being Admiralty (70 per cent. copper, 29 per cent. zinc and 1 per cent. tin) and Benedict bronze (92 per cent. copper and 8 per cent. aluminum). Sufficient time, however, has not elapsed to warrant the drawing of any conclusions.

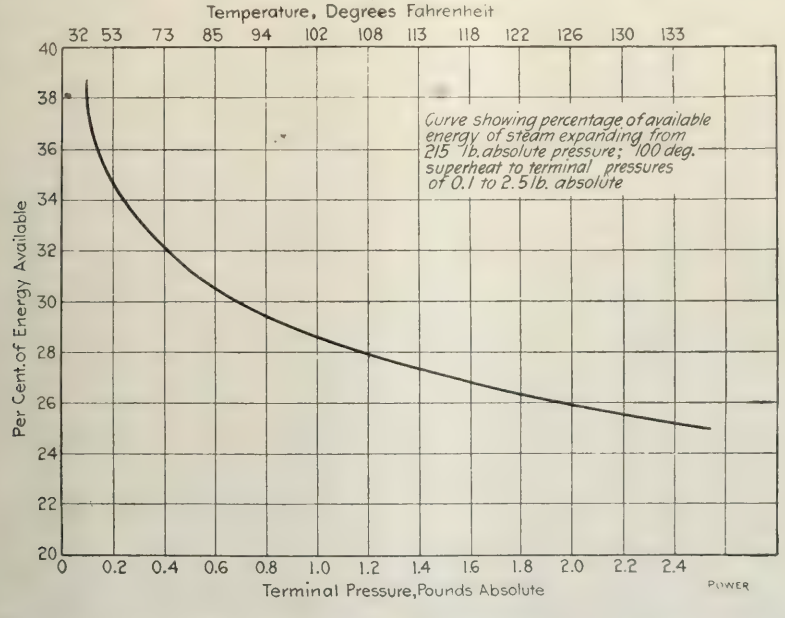
Another company that has had very serious tube troubles reports an installation of Benedict bronze tubes in a 32,000-ft. condenser for nearly three years with no indication of deterioration of any kind, while a similar installation a few months old is quite disappointing.

The so called Cumberland protective process is being tried out in several stations. The method employed consists of passing a small current of from 10 to 20 amperes through the tubes of the condenser to the ground in such a way that the tubes are made electro-negative to the water surrounding them. This system has not been in service a sufficient time to warrant any statements in regard to its value.

Combustion of Coal*

By E. A. Uehling

Every pound of coal of a given composition contains a definite amount of potential energy. Combustion liberates this energy in the form of heat energy. To produce combustion two factors are necessary: oxygen and high temperature.


Temperature, Degrees Fahrenheit
32 53 73 85 94 102 108 113 118 122 126 130 133
Curve showing percentage of available energy of steam expanding from 215 lb. absolute pressure; 100 deg. superheat to terminal pressures of 0.1 to 2.5 lb. absolute
Per Cent. of Energy Available
40 38 36 34 32 30 28 26 24 22 20
0 0.2 0.4 0.6 0.8 1.0 1.2 1.4 1.6 1.8 2.0 2.2 2.4
Terminal Pressure, Pounds Absolute
POWER

Combustion in the sense under consideration is the chemical reaction between oxygen (O) and carbon (C) forming carbon dioxide (CO_2), and between hydrogen (H) and oxygen forming water vapor; that is, steam (H_2O).

This chemical reaction, called combustion, liberates the potential energy in coal in the form of heat energy. For every unit of carbon and every unit of hydrogen, a fixed quantity of heat energy is liberated. This heat energy becomes effective at once by heating the solid carbon and ash of the coal to incandescence and raising the gaseous products of combustion to a high or flame temperature. The highest flame temperature results from complete combustion with the theoretically correct quantity of oxygen. In practice theoretically perfect combustion is impossible, but complete combustion is attainable, and, to produce the greatest useful heat effect, should be accomplished with a minimum excess of oxygen.

The rapidity of combustion depends on the temperature of the fire, and the quantity of oxygen supplied.

The oxygen required for the combustion is derived from the atmosphere, which is composed of a mixture of 23 parts oxygen and 77 parts nitrogen by weight. Nitrogen retards combustion; hence too great an excess of oxygen with its accompanying fourfold quantity of nitrogen will tend to bring the flame temperature below the point of ignition.

In the burning of coal, decomposition always precedes combustion; hence the greater the amount of volatile combustible contained in a coal the greater the flame volume, and the greater the length of flame the greater the loss of heat from radiation and the probability of incomplete combustion because of low-flame temperature.

Smoke is the usual manifestation of incomplete combustion. It results from three causes: low-flame temperature, stratification of air and the volatile combustible gases, and deficiency of oxygen. Stratification is caused by incompetent or careless firing, leaving an uneven fuel bed. The air following the course of least resistance rushes through the light covering in excess of the requirement and in inadequate quantity through the heavier portions, resulting in strata with oxygen in great excess and strata with deficiency of oxygen, all below the temperature which would result from proper intermingling of the air with the combustible gases, the one because of a deficiency in combustible and the other because of a lack of oxygen, all quickly becoming cooled below the point of ignition by giving up their heat to the boiler before opportunity for combustion was offered by proper intermingling.

That decomposition precedes combustion is not only true of coal, but also for the hydrocarbons. Heat separates the heavy hydrocarbons into free carbon and light hydrocarbons, and before combustion can take place the latter are further separated into carbon and hydrogen. In the act of combustion two atoms of hydrogen combine with one atom of oxygen, forming H_2O, water vapor, and one atom of carbon combines with two atoms of oxygen, forming CO_2, carbon dioxide.

The combustion of the hydrogen precedes that of the carbon and liberates an intense heat which raises the temperature of the carbon to incandescence, producing a luminous flame, which, if brought in contact with a cold body, produces smoke and soot. The same takes place if the flame temperature is reduced from any other cause, such as rapid radiation or admixture of an excess of

*Presented at the twenty-fifth meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers, May 16, 17 and 18, Pittsburgh, Penn.

cold air. Too great an excess of air is a prolific source of incomplete combustion.

To secure complete combustion there must be had a hot furnace, a moderate excess of air and a thorough intermingling of the air with the volatile combustibles. The first means a furnace so constructed that combustion will be complete before the temperature of the flame is materially reduced; the second, that the air supply must be controlled so as to avoid too great an excess; and the third, that the firing must be judiciously and skillfully done and the air admitted in the proper manner.

The object of burning coal under a steam boiler is to convert the chemical potential energy contained in the coal into mechanical potential energy, in the form of steam under pressure. Heat is the medium of transfer and the problem before the combustion engineer is, first, how to burn the coal completely, and second, how to get the maximum amount of heat liberated by combustion into the water and steam contained in the boiler under which the coal is burned.

The greatest single loss is the heat carried away by the flue gases. This increases as the difference in temperature between the flame and the flue gases decreases, and increases directly as the weight and temperature of these gases increase; now since the weight of the flue gas per unit of combustible increases directly with the excess of air supplied, and since the temperature of the flame is decreased thereby and that of the flue gas is increased, the importance of regulating the air supply becomes evident.

It is practically impossible to attain complete combustion of coal with the theoretical amount of air required to oxidize fully all its combustible elements. The amount of air required for maximum economy will vary with the chemical nature and the physical condition of the coal, with the type and construction of furnace and boiler, the availability and regulation of the draft and the manner of stoking; but for any one of these or combination of these conditions, there is a minimum excess of air that will produce maximum economy. Every pound of air in excess of that required carries heat to waste and a very great excess may even be the cause of incomplete combustion, thus proving wasteful at both ends.

Since every pound of combustible contained in a given coal carries a definite per cent. of carbon, it follows that in complete combustion with the theoretically required amount of air, the products of combustion must contain a definite per cent. of CO_2. It further follows that the per cent. of CO_2 must decrease as the excess of air increases and that CO_2 is, therefore, a correct index of the loss of sensible heat up the chimney.

Sporadic determinations of CO_2 are of small value, and unless judiciously taken, may even be quite misleading. The CO_2 from an average sample collected uniformly over a day's run is more valuable, since it shows the average efficiency of the fire, and in connection with the factor of evaporation will be of great aid in determining the conditions for improving economy; but here again the results may be misleading, especially in connection with hand-fired boilers. A proper per cent. of CO_2 in an average sample spread over a day's run is no guarantee of regular and efficient stoking. At some intervals the CO_2 may have been too high and at others too low for efficient work, causing a loss of combustibles due to deficiency of oxygen during the former, and a loss of sensible heat due to too great an excess of air during the latter intervals. These uncertainties are eliminated by employing an apparatus which will continuously indicate and autograhpically record the per cent. of CO_2 contained in the flue gases. Such a record is not only of the greatest assistance in determining the excess of air required for maximum efficiency, but it is necessary also to be able to maintain the high efficiency which it has been possible to attain by intelligent application of theory and practice to the existing conditions.

Sawmill Boiler Explosion

By W. J. Smith

A disastrous boiler explosion occurred at the Orillia Lumber Co. mill, about two miles west of Orillia, Wash., on the morning of Apr. 11. Three men were injured, one of whom died later.

The boiler and engine house and a considerable portion of the mill structure were destroyed, and the loss was several thousand dollars.

The boiler was about five years old, of the horizontal tubular type, 66 in. in diameter by 16 ft. long, made up in three courses; it had double-riveted lap seams. The boiler was designed to operate at about 110 lb. and this seems to have been the pressure carried. The failure occurred at the longitudinal seam of the rear course and was caused by a lap seam crack underneath the edge of the inner plate which gave no evidence of its presence until complete rupture occurred.

Subsequent examination revealed that the fracture, which was blackened and dull, must have existed for some time. It extended about 4 in. from the head seam to the same distance from the girth seam, and nearly severed the plate, some $\frac{3}{32}$ to $\frac{1}{16}$ in. of metal failing when the explosion occurred. What is left of the boiler is shown in the illustration.

The rear course apparently whipped off, snapping the rivets from both head and girth seams. When near the bottom a rending or tearing of the second course occurred and extended longitudinally into and through the first course. This metal showed a reduction of thickness due to the terrific strain imposed before these parts failed.

The braces supporting the heads above the tubes were of the round, upset or weldless type. One of the interesting features is the demonstrated high efficiency or holding power of this form of brace; though bent nearly at right angles to the shank or palm, only one was broken. The feet were pulled from a T- to a Y-form. In many instances the rivets fastening these ends and being in tension were reduced in area nearly 50 per cent. before giving way.

Close examination of the plate which failed first revealed that it was defective, due possibly to some constituent impurity. The openings for the safety valve and blowoff were both in this sheet and radiating from the punched holes on each opening were several small cracks. As these parts had never been under a strain, they indicate a brittle, unreliable material.

The rear course, opening on the sides and near the top, was driven downward and remained near the point of the accident, but the first and second courses, opening at or near the bottom, were lifted from their position and at the same time driven forward, bending the front head at right angles, stripping it from the tubes and then crashing through the roof of the building.

Remains of Exploded Sawmill Boiler

Horsepower and Kilowatts

By F. R. Low

A horsepower is equal to 745.65 watts or 0.74565 of a kilowatt.

A kilowatt is equal to 1000 watts or

$$1000 \div 0.74565 = 1.3411 \text{ hp.}$$

These factors will do for the abstract or mathematical conversion of horsepower and kilowatts, but in their physical conversion there is always a loss. If the efficiency of a unit is 90 per cent.—that is, if the electric energy at the terminals of the generator is only 90 per cent. of the mechanical energy applied to its production—one horsepower so applied will produce only $0.74565 \times 0.90 = 0.67108$ of a kilowatt and it will take

$$\frac{1}{0.74565 \times 0.90} = 1.49012 \text{ hp}$$

to produce a kilowatt.

For each pound of steam required per kilowatt at 90 per cent. efficiency there would be required 0.67108 lb. per horsepower; or for each pound required per horsepower 1.49012 lb. per kilowatt.

The accompanying table gives these factors for efficiencies from 67 per cent., where it takes practically 2 hp. to make a kilowatt, to 100 per cent., advancing by tenths of a per cent. The factors for the even percentages are given in the second column under zero, and those for the successive tenths under the columns so respectively denominated.

For example, with an efficiency of 88.5 per cent. it will be found in the line opposite 88 and under 0.5 that 1 hp. will produce 0.6599 of a kilowatt and that it will take 1.51538 hp. to produce a whole kilowatt.

A few examples will make the application and use of the table apparent.

EQUIVALENT HORSEPOWER AND KILOWATTS AT VARIOUS EFFICIENCIES

Efficiency, Per Cent.	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	Efficiency, Per Cent.
99	0.73819 1.35466	0.73894 1.35329	0.73968 1.35193	0.74043 1.35057	0.74118 1.34921	0.74192 1.34785	0.74267 1.34650	0.74341 1.34515	0.74416 1.34380	0.74490 1.34245	99
98	0.73074 1.36848	0.73148 1.36709	0.73223 1.36569	0.73297 1.36430	0.73372 1.36292	0.73447 1.36153	0.73521 1.36015	0.73596 1.35878	0.73670 1.35740	0.73745 1.35602	98
97	0.72328 1.38259	0.72403 1.38117	0.72477 1.37974	0.72552 1.37833	0.72626 1.37691	0.72701 1.37550	0.72775 1.37409	0.72850 1.37268	0.72925 1.37128	0.72999 1.36988	97
96	0.71582 1.39699	0.71657 1.39554	0.71732 1.39409	0.71806 1.39264	0.71881 1.39119	0.71955 1.38981	0.72030 1.38831	0.72104 1.38688	0.72179 1.38545	0.72253 1.38402	96
95	0.70837 1.41170	0.70911 1.41021	0.70986 1.40873	0.71060 1.40725	0.71135 1.40578	0.71210 1.40431	0.71284 1.40284	0.71359 1.40137	0.71433 1.39991	0.71508 1.39845	95
94	0.70091 1.42671	0.70166 1.42520	0.70240 1.42369	0.70315 1.42218	0.70389 1.42067	0.70464 1.41917	0.70538 1.41767	0.70613 1.41617	0.70688 1.41467	0.70762 1.41318	94
93	0.69345 1.44206	0.69420 1.44051	0.69495 1.43896	0.69569 1.43742	0.69641 1.43588	0.69718 1.43434	0.69793 1.43281	0.69867 1.43128	0.69942 1.42976	0.70017 1.42823	93
92	0.68600 1.45773	0.68671 1.45615	0.68749 1.45457	0.68823 1.45299	0.68898 1.45142	0.68973 1.44985	0.69047 1.44828	0.69122 1.44660	0.69196 1.44516	0.69271 1.44361	92
91	0.67854 1.47375	0.67929 1.47213	0.68003 1.47052	0.68078 1.46891	0.68152 1.46730	0.68227 1.46570	0.68301 1.46410	0.68376 1.46250	0.68451 1.46091	0.68525 1.45932	91
90	0.67109 1.49012	0.67183 1.48847	0.67258 1.48682	0.67332 1.48517	0.67407 1.48353	0.67481 1.48189	0.67556 1.48026	0.67630 1.47862	0.67705 1.47700	0.67780 1.47537	90
89	0.66363 1.50687	0.66437 1.50518	0.66512 1.50349	0.66587 1.50180	0.66661 1.50012	0.66736 1.49845	0.66810 1.49678	0.66885 1.49512	0.66959 1.49344	0.67034 1.49178	89
88	0.65617 1.52400	0.65692 1.52226	0.65766 1.52053	0.65841 1.51881	0.65915 1.51709	0.65990 1.51538	0.66065 1.51367	0.66139 1.51196	0.66214 1.51026	0.66288 1.50856	88
87	0.64872 1.54151	0.64946 1.53974	0.65021 1.53798	0.65095 1.53621	0.65170 1.53445	0.65244 1.53270	0.65319 1.53095	0.65394 1.52920	0.65468 1.52746	0.65543 1.52572	87
86	0.64126 1.55943	0.64020 1.55714	0.64275 1.55581	0.64350 1.55401	0.64424 1.55221	0.64499 1.55042	0.64573 1.54863	0.64648 1.54684	0.64722 1.54506	0.64797 1.54328	86
85	0.63380 1.57778	0.63455 1.57592	0.63529 1.57407	0.63604 1.57223	0.63679 1.57039	0.63753 1.56855	0.63828 1.56672	0.63902 1.56489	0.63977 1.56307	0.64051 1.56125	85
84	0.62635 1.59656	0.62709 1.59466	0.62784 1.59277	0.62858 1.59088	0.62933 1.58900	0.63007 1.58711	0.63082 1.58524	0.63157 1.58337	0.63231 1.58150	0.63306 1.57964	84
83	0.61889 1.61580	0.61964 1.61385	0.62038 1.61191	0.62113 1.60998	0.62187 1.60805	0.62262 1.60612	0.62336 1.60420	0.62411 1.60228	0.62485 1.60037	0.62560 1.59846	83
82	0.61143 1.63550	0.61218 1.63351	0.61292 1.63152	0.61367 1.62954	0.61442 1.62756	0.61516 1.62559	0.61591 1.62362	0.61665 1.62166	0.61740 1.61970	0.61814 1.61775	82
81	0.60398 1.65569	0.60472 1.65365	0.60547 1.65162	0.60621 1.64958	0.60696 1.64756	0.60770 1.64554	0.60845 1.64352	0.60920 1.64151	0.60994 1.63950	0.61069 1.63750	81
80	0.59652 1.67639	0.59727 1.67430	0.59801 1.67221	0.59876 1.67013	0.59950 1.66805	0.60025 1.66598	0.60099 1.66391	0.60174 1.66185	0.60249 1.65979	0.60323 1.65774	80
79	0.58906 1.69761	0.58981 1.69546	0.59055 1.69332	0.59130 1.69119	0.59205 1.68906	0.59279 1.68693	0.59354 1.68481	0.59428 1.68270	0.59503 1.68059	0.59577 1.67849	79
78	0.58161 1.71937	0.58235 1.71717	0.58310 1.71498	0.58384 1.71279	0.58459 1.71060	0.58534 1.70842	0.58608 1.70625	0.58683 1.70408	0.58757 1.70192	0.58832 1.69976	78
77	0.57415 1.74170	0.57490 1.73944	0.57564 1.73719	0.57639 1.73494	0.57713 1.73270	0.57788 1.73047	0.57862 1.72824	0.57937 1.72601	0.58012 1.72379	0.58086 1.72158	77
76	0.56669 1.76462	0.56744 1.76230	0.56819 1.75999	0.56893 1.75768	0.56968 1.75538	0.57042 1.75309	0.57117 1.75080	0.57191 1.74852	0.57266 1.74624	0.57340 1.74397	76
75	0.55924 1.78815	0.55998 1.78577	0.56073 1.78339	0.56147 1.78102	0.56222 1.77866	0.56297 1.77631	0.56371 1.77396	0.56446 1.77161	0.56520 1.76928	0.56595 1.76695	75
74	0.55178 1.81231	0.55253 1.80987	0.55327 1.80743	0.55402 1.80500	0.55476 1.80257	0.55551 1.80015	0.55625 1.79774	0.55700 1.79533	0.55775 1.79293	0.55849 1.79054	74
73	0.54432 1.83714	0.54507 1.83463	0.54582 1.83212	0.54656 1.82962	0.54731 1.82713	0.54805 1.82464	0.54880 1.82216	0.54954 1.81969	0.55029 1.81722	0.55104 1.81477	73
72	0.53687 1.86266	0.53761 1.86007	0.53836 1.85750	0.53910 1.85493	0.53985 1.85236	0.54060 1.84981	0.54134 1.84726	0.54209 1.84472	0.54283 1.84219	0.54358 1.83966	72
71	0.52941 1.88889	0.53016 1.88623	0.53090 1.88358	0.53165 1.88094	0.53239 1.87831	0.53314 1.87568	0.53389 1.87306	0.53463 1.87045	0.53538 1.86784	0.53612 1.86525	71
70	0.52196 1.91587	0.52270 1.91314	0.52345 1.91042	0.52419 1.90770	0.52494 1.90499	0.52568 1.90229	0.52643 1.89959	0.52717 1.89690	0.52792 1.89423	0.52867 1.89155	70
69	0.51450 1.94364	0.51524 1.94083	0.51599 1.93802	0.51674 1.93523	0.51748 1.93244	0.51823 1.92966	0.51897 1.92688	0.51972 1.92412	0.52046 1.92136	0.52121 1.91861	69
68	0.50704 1.97222	0.50779 1.96933	0.50853 1.96644	0.50928 1.96356	0.51002 1.96069	0.51077 1.95783	0.51152 1.95497	0.51226 1.95213	0.51301 1.94929	0.51375 1.94646	68
67	0.49959 2.00166	0.50033 1.99868	0.50108 1.99570	0.50182 1.99274	0.50257 1.98978	0.50331 1.98683	0.50406 1.98389	0.50481 1.98096	0.50555 1.97804	0.50630 1.97513	67

How many kilowatts would a brake load of 896.45 hp. produce when applied to a generator having an efficiency of 91.6 per cent.?

In the line opposite 91 per cent. and in the column under 0.6 it is shown that with an efficiency of 91.6 per cent. 1 hp. will develop 0.68301 kw. The given horsepower will therefore produce

$$896.45 \times 0.68301 = 612.28 \text{ kw.}$$

What indicated horsepower would have to be developed in an engine which was part of a unit having an overall efficiency of 87.3 per cent. to produce 1200 kw.?

With an efficiency of 87.3 the table shows that it takes 1.53621 hp. to produce a kilowatt. To produce 1200 kw. would therefore require

$$1200 \times 1.53621 = 1843.45 \text{ hp.}$$

An engine has a mechanical efficiency of 95 per cent. and the generator to which it is attached an efficiency of 94 per cent. What would be the indicated horsepower when the unit was turning out 2500 kw.?

The combined or overall efficiency of the engine and generator would be

$$95 \times 94 = 89.3 \text{ per cent.}$$

The table shows that at this efficiency it takes 1.5018 hp. to produce a kilowatt. To produce 2500 kw. would therefore require

$$2500 \times 1.5018 = 3754.5 \text{ hp.}$$

What would be the brake or shaft horsepower for the same output?

The generator converts the shaft energy with an efficiency of 94 per cent., at which the table says it takes 1.42671 hp. to make a kilowatt. To make 2500 kw. would therefore require

$$2500 \times 1.42671 = 3566.77 \text{ b.hp.}$$

This is, as it should be, 95 per cent. of the indicated horsepower

$$3754.5 \times 0.95 = 3566.77$$

A turbine-driven unit with an overall efficiency of 90.6 per cent. requires 16.5 lb. of steam per kilowatt-hour; what is the water rate per indicated horsepower?

At 90.6 per cent. the table shows that the kilowatts are 0.67556 of the horsepower. The steam consumption per horsepower is therefore 0.67556 times that per kilowatt or for the present case

$$0.67556 \times 16.5 = 11.1467 \text{ lb. per i.hp.}$$

There need be no uncertainty which of the two numbers in the table to use. It is evident that the steam per horsepower will be less than that per kilowatt, and that the smaller number is to be used as a multiplier in this case, while if the water rate is given in pounds per horsepower-hour it should be multiplied by the larger value in the table to obtain the water rate per kilowatt-hour.

The Quality of Steam in Boilers

By O. Z. Howard

How to make a calorimeter from pieces of pipe and standard fittings; also the method of figuring the quality of the steam and making corresponding corrections in the total evaporation.

An essential feature of a boiler test that is frequently omitted is the determination of moisture in the steam. While this may be so small as to affect the result but little in some cases, in others it may affect the equivalent evaporation as much as 2 or 3 per cent.

If a calorimeter is not available, one can be made very easily from pipe and standard figures like the illustration. The perforated nipple, with the long thread, should be screwed into a vertical portion of the steam pipe, as near the boiler as possible. The nipple should be of such length that the ⅛-in. holes near the ends will be about ½ in. from the sides of the steam pipe when screwed in place. The calorimeter should be thoroughly lagged with hair felt, from the point where it leaves the steam pipe to the elbow below the lower thermometer well.

The thermometer wells should be filled with mercury or a heavy oil, and a second thermometer should be suspended, in the open air, near each of those in the wells, to obtain the temperature of their stems, for making corrections as shown later. A manometer gage should be mounted on the side of the lower tee for measuring the back pressure in the exhaust pipe. An ordinary U-tube, as commonly used for taking boiler draft, will answer all requirements if filled with mercury in place of water.

If the amount of evaporation is determined by weighing the water before entering the boiler, no account of the loss of steam from the calorimeter will be necessary; but if the steam from the boiler is condensed and weighed, a correction for this loss will have to be made. This can be accomplished by running the exhaust into a barrel partly filled with water, thus condensing the steam and then weighing; by calibrating the orifice for the pressure used during the test, or by calculating the discharge. This latter method is correct enough for all practical purposes, and the discharge in pounds per minute, for an orifice ⅛ in. in diameter, will be $0.0105P$, where P is the absolute boiler pressure.

Before readings are taken on the calorimeter, the ½-in. valve should be opened wide for 10 or 15 min., then frequent readings of the four thermometers and the gage should be noted. The reading of the thermometer in the upper well, if properly corrected, should be a check on the gage reading. Assuming that there is no loss of heat by radiation in passing through the instrument, the total heat in the mixture of steam and water before throttling and in the superheated steam after throttling, will be the same, and can be expressed by the equation

$$xr + q = H_c + 0.476\,(T_1 - T_c)$$

or

$$x = \frac{H_c - q + 0.476\,(T_1 - T_c)}{r} \qquad (1)$$

where

x = Quality of the steam;
r = Latent heat of the steam at the pressure in the main;
q = Heat of liquid due to the pressure in the main;
H_c = Total heat in 1 lb. of dry steam at the calorimeter pressure (shown by manometer gage);
T_1 = Temperature in calorimeter (shown by lower thermometer);
T_c = Normal temperature of steam in calorimeter due to calorimeter pressure.

Thermometer Corrections

In general, all thermometers are calibrated for total immersion; that is, for the condition where both the stem and the bulb are at the same temperature. But in taking the temperature of the steam in a calorimeter some of the stem projects into space colder than the bulb. That being the case, a correction must be applied to the observed reading in addition to the correction found by the previous calibration of the thermometer.

The Bureau of Standards gives the following formula:

$$\text{Stem correction} = 0.000092 \times n\,(T - t) \qquad (2)$$

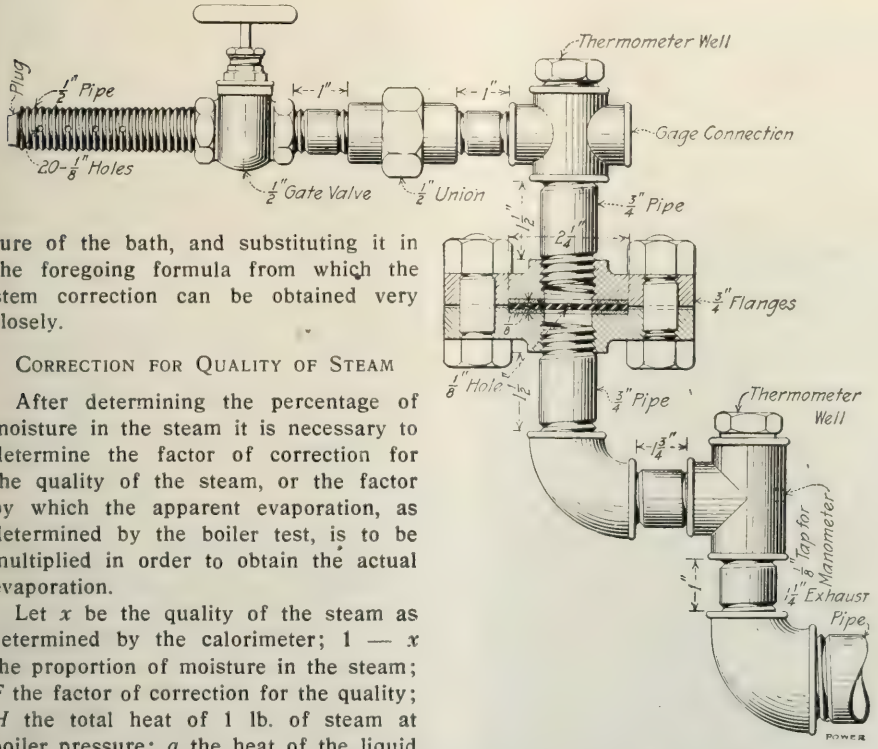
where

n = Number of degrees projecting from bath;
T = Temperature of the bath;
t = Mean temperature of the projecting stem.

The mean temperature t may be measused approximately by means of an auxiliary thermometer suspended near-by.

If the thermometer is used under conditions such that it has only a small stem correction, the reading of the ther-

mometer corrected by the mean of the table (from previous calibration) may be taken as the temperature of the bath, since a difference in the bath temperature makes a much smaller difference in the value of the stem correction. If, however, the number of graduations projecting from the well and the difference of temperature between the bath and the space above it are large, it will be necessary to determine the stem correction approximately and apply this to the corrected reading of the thermometer in order to get the approximate temperature of the bath. This may be done by using the reading of the thermometer, corrected from the table, as the temperature of the bath, and substituting it in the foregoing formula from which the stem correction can be obtained very closely.



Sampling Tube and Homemade Calorimeter

Correction for Quality of Steam

After determining the percentage of moisture in the steam it is necessary to determine the factor of correction for the quality of the steam, or the factor by which the apparent evaporation, as determined by the boiler test, is to be multiplied in order to obtain the actual evaporation.

Let x be the quality of the steam as determined by the calorimeter; $1 - x$ the proportion of moisture in the steam; F the factor of correction for the quality; H the total heat of 1 lb. of steam at boiler pressure; q the heat of the liquid at the temperature due to boiler pressure, and h the heat of the liquid at the feed-water temperature.

The proportional part x receives from the boiler the sensible and the latent heat, or, the total heat in this portion above the temperature of the feed water is $x\ (H - h)$ B.t.u.; and the part $(1 - x)$ representing the water in 1 lb. of the mixture is $(1 - x)\ (q - h)$ B.t.u. But if all the water had been evaporated, each pound would have received the total heat in the steam above the temperature of the feed water, or $(H - h)$ B.t.u.

Hence,

$$F = \frac{x\,(H - h) + (1 - x)\,(q - h)}{(H - h)}$$

or

$$F = x + \frac{(1 - x)(q - h)}{(H - h)} \qquad (3)$$

Example: From a boiler test the following data are obtained with a throttling calorimeter: Steam gage, 144 lb. per square inch; temperature in the upper well, when immersed to the 200-deg. mark, 361 deg. F.; auxiliary thermometer at the upper well, 102 deg. F.; temperature in the lower well, 260 deg., when immersed to 240 deg.; auxiliary thermometer at the lower well, 102 deg.; manometer gage, 0.42 in. of mercury (corresponding to a pressure of 0.21 lb. per square inch), and the barometer (corrected) stood at 29.77 in.

Assume that from the calibration of the thermometers before the test, the reading of the thermometer in the upper well at 361 deg. was 2 deg. too high, and the reading of the thermometer in the lower well at 260 deg. was 1 deg. too low. Then from equation (2) the stem correction for the upper thermometer is

$$0.000092 \times 161\ (359 - 102) = 3.8\ deg.$$

and the true reading will be

$$361 + 3.8 - 2 = 362.8\ deg.\ F.$$

As so little of the stem of the lower thermometer projects, the stem correction need not be taken into account, and the true reading will be

$$260 + 1 = 261\ deg.\ F.$$

The reading of the barometer, 29.77 in., corresponds to a pressure of 14.62 lb. per square inch; hence the absolute steam pressure is

$$144 + 14.62 = 158.62\ lb.$$

per square inch, which from the steam table corresponds to a temperature of 362.9 deg., agreeing within 0.1 deg. of the temperature as obtained by the thermometer.

The quality of the steam may now be obtained. From the steam tables: r for saturated steam at 362.8 deg. F. equals 859.6 B.t.u.; q for saturated steam at 362.8 deg. F. equals 334.8 B.t.u.; H_c for 1 lb. of dry steam at 14.83 lb. (14.62 + 0.21) pressure absolute equals 1150.5 B.t.u.; T_1 from the corrected temperature in the lower well equals 261 deg. F., and T_c for the steam at 14.83 lb. pressure equals 212.41 deg. F.

Substituting in formula (1),

$$x = [1150.5 - 334.8 + 0.476\ (261 - 212.41)] \div 859.6 = 0.9758$$

or, the steam is 97.58 per cent. dry.

If the temperature of the feed water entering the boiler is 100 deg. in equation (3): $x = 0.9758$ and $(1 - x) = 0.0242$; q for saturated steam at 362.8 deg. = 334.8; h for water at 100 deg. = 67.97; H for saturated steam at 362.8 deg. = 1194.4.

Then, the factor of correction is

$$F = 0.9758 + \frac{0.0242\ (334.8 - 67.97)}{1194.4}$$

or $F = 0.9812$.

Wear in Steam Turbines

It is frequently claimed for the steam turbine that its initial efficiency will be maintained whereas the steam engine, due to wear producing leakage of the piston and valve and derangement of the steam distribution, will rapidly deteriorate. S. Schulte, in *Glückauf*, says that the erosion of the blades of the turbine will produce changes in the direction and conditions of the entry of the jet which have a serious effect upon the efficiency. Temperatures over 250 deg. C. (482 deg. F.) may totally destroy bronze blades in from two to four years. Nickel steel with 25 per cent. nickel has been found to be unusable. The best metal for blades is nickel-steel with 5 per cent. of nickel. The lateral wear of the bearings has also a detrimental effect upon the efficiency. The duration of the blade is considerably greater with lower steam velocity.

With a turbine of 500 kw. capacity, running at 3000 r.p.m., after 29,700 hours of operation with a production of 8,350,000 kw.-hr. the consumption had increased about 27 per cent.

A Curtis turbine of 1000 kw. capacity, running at 3000 r.p.m., with bronze blades, working at only one-half or three-quarters capacity, after 31,000 hours of running and a production of 17,800,000 kw.-hr. had an increased water rate of about 18 per cent.

In the case of a Zoelly turbine of 1000 kw. capacity, 19,500 hours of operation and a production of 13,160,000 kw.-hr. resulted in an increased consumption of about 3 per cent.

It would appear that in order to maintain the initial efficiency of a turbine, it may require occasional, if not frequent, reblading.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Catechism of Electricity

MANAGEMENT OF ELECTRICAL STATIONS

1304. *Is it not important to have system in the operation of an electrical station?*

Yes; because with a proper system each attendant has his special duties to perform, someone is responsible for whatever is to be done and the chance of overlooking matters is reduced to a minimum.

1305. *What provision should be made to safeguard attendants?*

Belts and moving parts of machinery should be inclosed with railings or guards, also dark pits and dark passages, such as back of the switchboard, must be well lighted, as shown in Fig. 428. "Stumbling blocks" should be removed or made as few as possible. Switchboards on which high-voltage currents are handled, should have an insulated space in the floor or an insulated platform in front of the board, as is also shown in Fig. 428. The same precaution should be taken with the generators, especially if the frames are not grounded.

FIG. 428. SHOWING WELL LIGHTED PASSAGES BACK OF SWITCHBOARD AND TRANSFORMERS

1306. *Should the generator frames be grounded?*

Generators operating at a potential over 550 volts should have their base frames permanently grounded.

1307. *What advantage results from grounding a generator frame?*

Grounding a frame removes the possibility of dangerous shocks when coming in contact with it. The insulation of the entire system then depends upon the insulation of the generator conductors from the frame; if this breaks down, the system is grounded and the condition is indicated at once by the switchboard instruments.

1308. *Is there any argument against grounding a generator frame?*

Yes; the insulation resistance of the entire installation is reduced and there is an increased danger of shock from the line wires. This objection, however, can be overcome by employing a thorough system of insulation—a precaution which should obtain in any case, and which in consequence lessens the force of the argument against the practice.

1309. *How may a generator frame be properly grounded?*

By securely fastening one end of a wire to the frame and the other to a main water pipe inside the building. With a direct-connected generator an excellent ground may be had through the engine coupling and piping.

1310. *What precaution should be observed in connection with generators having grounded frames?*

High-potential generators with their frames grounded, should be surrounded by a small platform raised above the floor on porcelain or glass insulators so that those working about the machine may be protected from shock when adjusting the brushes.

1311. *How should generators be protected when not in use?*

Waterproof covers should be placed over them. Otherwise water may get into the working parts and cause an armature or field coil to burn out when the machine is started.

1312. *What provision should be made in the station wiring and apparatus to afford safety from fire?*

The station wiring and apparatus should conform to the rules and requirements of the National Board of Fire Underwriters. These are embodied in the "National Electric Code," a booklet which may be obtained free upon request from the electrical department of the National Board of Fire Underwriters, 135 William St., New York City.

1313. *What kind of fire-extinguishing apparatus should be placed in the station?*

Every station should be supplied with a fire pump and an adequate line of pipe and hose. The pump should be run at least once a week and kept in good working order. Care should be taken in locating the sprays and pipes so that water will not come in contact with the generators or the conductors. Also fire buckets filled with water should be placed in convenient places. Furthermore, it is important to have at hand a number of buckets filled with dry sand, which may be used in extinguishing fire between electrical conductors where water would do more harm than good. The buckets should be kept covered to prevent the sand from being blown around in the station.

1314. *What should be done to insure cleanliness in the station?*

The machines must be kept free from grease and dirt. Greasy rags and waste should not be thrown carelessly around but should be placed in cans provided for that purpose. Filing or repairing of any kind that produces dust or small particles of metal should, if possible, be performed elsewhere than in the generator room. Another aid to cleanliness and order is to keep all supplies under lock and key in a store-room.

1315. *How should the stock in the store-room be handled?*

If the station is large there should

be a person whose only duty consists in keeping charge of the supplies; and accurate accounts should be made by him of all receipts and disbursements. Applications for material should be made on regular printed forms which may be placed on file and used in keeping the stock books and accounts. In smaller stations, the store-room keeper may perform other duties as well.

1316. *What is the best method of issuing orders to the station attendants?*

Having them plainly written and fastened by means of thumb tacks upon a bulletin board in a conspicuous part of the station. All operating orders, such as the dynamo schedule, circuit schedule and orders for changes, should be handled in this manner. Each dynamo and each circuit should have its number, and the orders should make it unmistakably plain as to when each dynamo shall start and stop; also during what time and from which machine, each circuit shall be operated. There should always be room left on the board for a notice to be posted when a high-voltage line is being worked on, and when a man is in a flywheel pit or in any other position where he is liable to be injured by the starting of some particular machine.

1317. *How are reports of value in the operation of a plant?*

Without accurate reports an engineer cannot keep a check on what is being done.

1318. *What is the best way of collecting material for the reports?*

By means of recording instruments and keeping all the paper dials or cards obtained from them on file. Most of the necessary calculations may be made from these records. When the station is not provided with recording instruments, the different meters in service should be read at least every half hour and the readings entered on regular report blanks. In large stations there should be separate blanks for the different departments, but in a small station, one blank may be used for all.

1319. *Give an example of a common form of power-house report.*

Fig. 429, represents a power-house report on four machines for one day, from midnight to midnight, the generators being driven by two waterwheels. Two of the generators are 550-volt direct-current machines, the third is a 2000-volt alternator having a capacity of 75 amperes, and the fourth an arc-light generator carrying a load of forty-five 2000- cp. arc lamps. There are but three circuits, the two direct-current generators being run in multiple when the load exceeds 100 amperes.

1320. *What information regarding the operation during the 24 hr. may be obtained from this report?*

In the vertical columns are recorded the readings on the instruments of each machine, at the end of every half hour. At the end of a day's run, the columns are added and averaged; the values are then recorded in the upper right-hand corner, as shown. The voltmeter for the alternator is connected across the secondary wires of a transformer that reduces the pressure to 100 volts when the current is generated at 2000 volts; in other words, it transforms the voltage in a ratio of 20 to 1. In making out the report, the voltage across the secondary terminals of the transformer is recorded. The arc-light voltage across the terminals at the switchboard is recorded and indicates, approximately, the number of lamps burning when the readings are being taken. This voltage is usually entered in round figures, that is, to the nearest multiple of 50—the voltage per lamp. The average voltage is treated in the same manner.

1321. *How are the calculations relating to the individual generators arrived at?*

At the end of the day the average current from each generator is multiplied by its average voltage, giving the average watts developed by the machine. This multiplied by the number of hours it has been in operation and divided by 1000 gives the output of the dynamo for the day, in kilowatt-hours. By adding together the outputs of the machines, the total work done by the plant will be found, which, in this case, is 1846 kw.-hr. This may be reduced to horsepower by multiplying by 1.34.

THE POWERVILLE ELECTRIC CO.

POWER HOUSE REPORT

TUESDAY, JAN. 16, 1911

TIME.	DYNAMO NOS. 1 A	1 V	2 A	2 V	3 A	3 V	4 A	4 V
12.30	40	500			20	103	10.2	1250
1.00	30	506			18	103	10.2	1100
1.30	30	500			15	102	10.5	1000
2.00	20	500			12	102	10.5	1000
2.30	40	500			10	101	10.5	1000
3.00	20	500			14	102	10.4	1050
3.30	15	500			28	104	10.3	1150
4.00	50	510			32	104	10.2	1500
4.30	60	525			37	105	10.2	1850
5.00	65	540			37	105	10.3	1850
5.30	80	540			25	103	10.1	1950
6.00	110	525			33	103	10.5	1000
6.30	80	530	45	530	20	103	10.5	1000
7.00	70	530	70	530	16	102		
7.30	100	550	100	550	14	102		
8.00	100	550	90	550	9	101		
8.30	90	550	90	550	10	101		
9.00	95	550	85	550	10	101		
9.30	80	550	75	550	15	102		
10.00	60	540	65	540	14	102		
10.30	65	540	65	540	16	102		
11.00	80	545	65	545	16	102		
11.30	80	545	70	545	15	102		
12.00	90	550	85	550				
12.30	90	550	85	550				
1.00	90	550	80	550	12	102		
1.30	85	550	85	550	14	102		
2.00	80	550	75	550	14	102		
2.30	75	550	75	550	14	102		
3.00	75	540	60	540	15	102		
3.30	70	540	60	540	16	102		
4.00	80	550	75	550	20	103		
4.30	80	555	80	555	28	104		
5.00	100	555	90	555	39	105	10.5	1050
5.30	105	560	105	560	58	108	10.3	1200
6.00	100	560	110	560	70	109	10.1	1950
6.30	90	550	110	550	75	110	9.8	2250
7.00	90	550	95	550	78	110	9.8	2250
7.30	80	550	90	550	76	110	9.8	2250
8.00	80	540	85	540	75	110	9.8	2250
8.30	60	530	80	530	73	110	10.2	2200
9.00	50	540	70	540	73	110	10.2	2200
9.30	40	530	65	530	65	109	10.2	1850
10.00	40	530	50	530	55	107	10.2	1700
10.30			85	555	40	105	10.2	1400
11.00			45	535	38	105	10.2	1400
11.30			45	525	29	104	10.2	1350
12.00			40	500	25	103	10.2	1250

Dynamo Nos.	Started	Stopped.	Hrs. Run	Averg. Amps.	Averg. Volts	Kilowt Hours.
1	12.00 p.m.	10.00 p.m.	22.0	70.7	536	834
2	6.30 a.m.	12.00 p.m.	17.5	76.3	543	726
3	12.00 p.m.	11.30 a.m.				
	1.00 p.m.	12.00 p.m.	22.5	31.0	104.2	73
4	12.00 p.m.	6.30 a.m.				
	5.00 p.m.	12.00 p.m.	13.5	10.2	1550	213
Wheel No. 1	12.00 p.m.	11.00 p.m.	23.0			
" 2	4.00 a.m.	12.00 p.m.	20.0			

Average No. Arc Lamps..... 31
Average No. Incandescent Lamps..... 1,240
Total No. Incandescent Lamp-hours..... 27,900
" " Kilowatt-hours..... 1,846
Head of Water, 6 a.m., 27 ft. 4 in.; 6 p.m., 27 ft. 6 in.

SUPPLIES RECEIVED.	SUPPLIES USED.
31 gal. cylinder oil.	1 gal. cylinder oil.
31 " Renown engine oil.	2 " Renown engine oil.
4 doz. dynamo towels.	1 " O.K. dynamo oil.
6 " 10-in. files.	2 cans potash.
	1 10-in. file.

REMARKS.

Short circuit on dynamo No. 1, from 6.10 a.m. to 6.20 a.m.

Continued heavy ground on dynamo No. 3 Occasional poor contact in arc circuit.

Dynamo No. 2 running exceptionally warm

Signed,

JOHN BROWN

FIG. 429. FORM OF POWER HOUSE REPORT

1322. *Why is it necessary to record the average number of arc and incandescent lamps that are burning during the run?*

Because the revenue of a lighting company is normally figured on this basis.

1323. *How is the average number of arc lamps calculated for the report?*

The number of arc lamps is directly proportional to the voltage of the arc dynamos; therefore, dividing the average

pressure, 1550, by 50, which may be taken as the voltage of each lamp, the result will be approximately the average number of lamps burning during the run of 13½ hr., which in this case is 31.

1324. *How is the average number of incandescent lamps calculated for the report?*

These may be calculated from the current of the alternator. The 16-cp. lamp is taken as the standard, and it is customary to allow 50 watts for such a lamp. If the voltage is 100, each lamp will consume ½ ampere, and, since the ratio of the transformer is 20: 1, 20 amperes in the secondary winding will result from a current of 1 ampere in the primary; consequently there will be 40 lamps (20 ÷ ½) burning in the secondary circuit for each ampere of current generated by the alternator. The machine was delivering

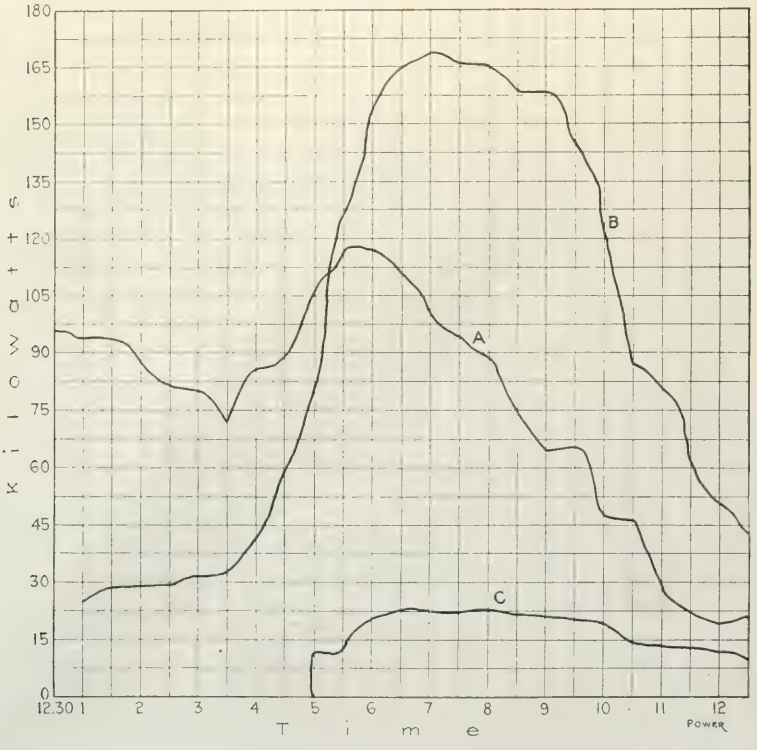


FIG. 430. TYPICAL LOAD CURVE

on an average 31 amperes, making the average number of incandescent lamps

$$31 \times 40 = 1240$$

Multiplying this number by 22½ hours during which the alternator was running, shows the number of lamp-hours to be 27,900.

1325. *Is it not sometimes more desirable to know the total kilowatt-hours supplied to the different parts of a system, than to know the average number of lamps burning?*

It may be; in which case this part of the report should be changed to suit conditions.

1326. *What information belongs under the heading "Remarks"?*

Notes regarding the general condition of the plant and of the individual machines, especially when anything out of the regular order occurs. It is especially important to give an accurate and minute description of all accidents, together with their causes. Carefulness in this matter often enables the engineer to shake off responsibilities that are liable to be heaped upon him by other parties.

1327. *What conditions besides those previously mentioned will affect the general makeup of the report?*

When there are many different circuits, the time when they are opened and when they are closed should be noted, together with the conditions of the circuits. When this is done, it is best also to note the state of the weather, since that greatly affects the condition of the lines.

The report shown is for a water-power plant. When steam is used, a report of the boiler room, etc., is usually included. An extra column is also provided for a record of the steam pressure, and space must be left to show when the fires are started, when the boilers are cleaned, the amount of fuel received and consumed, and further data necessary to make a complete report. Here, again, a variation may take place. A separate report may be kept of the steam plant, which may be done by an engineer who has nothing whatever to do with the dynamos, etc. This method is generally adopted in large plants, but in small plants it is not necessary and will only cause extra labor.

1328. *How often should power-house reports be turned in to the office?*

In most companies it is required that a report be sent to the office each day; but sometimes only the general information regarding load and accidents is wanted, in which case it is advisable to always make out a full report as explained, and then make a synopsis of this for the office.

1329. *What is the best method of furnishing the office with information regarding the loads carried by the different machines?*

Plot the readings on a sheet of coordinate paper, as shown in Fig. 430.

1330. *Explain how the graphical report in Fig. 430 furnishes information regarding the loads on the generators.*

Curve *A* represents the output of dynamos Nos. 1 and 2, curve *B* the output of the alternator and curve *C* the output of the arc-lamp dynamo. The height of the curve *B*, for instance, at 8 p.m. shows the output of the alternator at that moment to be 165 kw. When desired a curve may be similarly drawn showing the combined output of all the generators.

1331. *Does the management of a substation differ from that of a central station?*

Only in minor details. Rotary converters and synchronous motors, as has previously been explained, are started somewhat differently from ordinary generators and they are not handled in exactly the same way when they are brought to rest. They are, however, taken care of and repaired in the same manner as generators. The principal thing to observe is that these machines must not be thrown onto the line unless they are in step with the apparatus supplying the current. They must also be protected with reliable safety devices that will open the circuit if they should be pulled out of step.

Electrically Heated Office Building

Cheap hydro-electric power, generated on the Snake River, Idaho, has helped develop the city of Twin Falls very rapidly and has even rendered practicable the electric heating of buildings, says the *Scientific American*. In place of steam radiators, electric heating units have been installed in the vicinity of Twin Falls to the extent of nearly 1000 kw. connected load. A six-room house having eight outlets uses $70 to $100 worth of electrical energy during the season of eight months. A building having stores and offices on the first floor and a ballroom on the second floor uses $625 worth of electrical energy for the season.

Three-phase current is brought into this building through an underground conduit at 2300 volts and transformed to 220 volts for the 40 electric radiators and a water-tank heater, which keeps water continually at the boiling point through thermostatic control. The size of the building is 50x120 ft., and the electrical equipment cost about $1000, exclusive of the cost of the transformers, which were furnished by the power company.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Oil Engines

The current report of the "prime movers" committee of the National Electric Light Association contains interesting data upon the fuel-oil production and the status of the oil engine in this country. The following statistics for 1909, covering the petroleum production and the amount of oil available for power purposes, are given:

	Barrels	Per Cent.
Total production (crude)	182,000,000	100
Exported (crude)	5,184,700	2.8
Domestic consumption	176,815,300	97.2
Total amount available for gas and fuel oil	68,264,000	100
Exported (gas and fuel oil)	2,901,000	4.2
Domestic consumption (gas and fuel oil)	65,363,000	95.8
Oil burned in locomotives	20,000,000	29.3
Used in gas making	13,184,000	19.3
Available for power	32,179,000	47.2

The report states further that, although figures on the oil distribution for 1911 are not yet available, it is estimated that the total production in the United States last year was 209,500,000 bbl., of which 77,800,000 bbl. were produced west of and 132,200,000 east of the Rocky Mountains. While the total production showed an increase over 1910, the Eastern oil fields showed a slight decrease. This was offset, however, by an over-production in excess of the consumption in California of 10,500,000 bbl. This had a tendency to stop further drilling.

Fuel oil as supplied for power purposes consists ordinarily of a distillate that comes from the distillation of crude petroleum after the gasoline, kerosene and lubricating oils have been abstracted. It distills at temperatures from 300 to 700 deg. F. and varies in specific gravity from 40 to 20 deg. Baumé. The modern oil engines have been able to handle fuel oils successfully that are as heavy as 17 deg. Baumé and as light as kerosene, or 40 to 48 deg.

The heat value of oil varies from about 18,000 to 20,000 B.t.u. per pound and this seems to bear an almost direct relation to the density. In Germany as well as in the United States gas-house tar has been used as fuel in engines of the Diesel type, but its use has not proved entirely satisfactory as it is impossible to completely free it from water, which is apt to cause the engine to misfire and give poor speed regulation.

The character of the fuel oil is not always dependent upon the nature and quality of the original crude oil, as the methods of distillation are such that a uniform grade of residual oil is not always produced. However, the various grades of fuel oil, as now supplied for power purposes, give satisfactory results if free from objectional ingredients, such as sulphur, moisture and acids, the acid being an ingredient mixed with the oil during its purification after distillation. Sulphur, in the form of SO_3 or CS_2, ordinarily in a dry gas produces no harmful results, but in the presence of moisture sulphuric acid is formed and this attacks the pistons, cylinders and valves.

In the language of the committee, the oil engine differs from the more familiar gas engine only in that the fuel must be prepared for combustion at the engine, or injected at high pressures, and therefore either a carburetor, vaporizer or an injection valve must be provided for vaporizing the liquid-oil fuel, depending upon whether low, high or modified compression types of engines are used. The simple automobile engine comes within the former class (low pressure). For power purposes the low-pressure engine frequently employs a hot-bulb or "preheating chamber," to enable the engine to burn comparatively low-grade oils. The Diesel engine employs high-combustion pressures (500 lb., approximately) and requires an air-injection pressure of about 1000 lb. It has a cycle peculiar to itself, and the distinguishing feature is that the fuel is injected at constant pressure for about 10 per cent. of the working stroke, while the air is hot enough by reason of the high compression to cause combustion on contact.

With the fuel admitted only after the working stroke begins, there is no opportunity of a premature explosion to take place. Moreover, as only a small clearance of about 7 per cent. is necessary, more complete scavenging may be effected, which improves the efficiency accordingly.

The long-sustained high pressure and temperature of the regular Diesel engine obviously places some severity upon the operating parts, thereby requiring sturdy construction. On the other hand, the suddenly applied pressure in the low-pressure type, Beau de Rochas or Otto cycle, introduces high stresses which must also be provided for.

In point of fuel economy the superiority of the Diesel type has been very well established. Regarding maintenance and attendance, the low-pressure type has much in its favor. This would lead to the opinion that for the larger sizes, exceeding 150 hp., the Diesel motor should be chosen, while in case of small power requirements (under 100 hp.) the low-pressure type would generally be used. In the neighborhood of 100 to 150 hp. the field will be disputed between the two according to existing fuel prices and local conditions. There has been developed a design which may be termed a "compromise" between the low-pressure hot-bulb and the Diesel high-compression engines, for which there is claimed a fuel economy comparative with the Diesel motors, although it employs lower working pressures.

The guarantees on oil engines ordinarily offered by the manufacturer in the past have been as follows, for loads between three-fourths and full-load rating:

Diesel

Oil consumption not to exceed 0.5 lb. per horsepower-hour.

De La Vergne

Oil consumption not to exceed 0.6 lb. per horsepower-hour.

Hornsby-Akroyd

Oil consumption not to exceed 1 lb. per horsepower-hour.

Oil not to contain less than 18,500 B.t.u. per pound.

In response to inquiries sent by the committee to a large number of plants many interesting replies were received relative to the operation, maintenance and cost in American plants using oil engines. These are herein tabulated.

The load factor in the table is taken as the average percentage of rated load carried by the engine during its period of service.

The operators registered several criticisms of the oil engines regarding minor troubles, but in no case was there any serious complaint made, with the exception of a few instances where breakages were directly traceable to the inexperience of the operators, or to the possible inadequacy of the earlier designs. Pistons and piston rings have been most prominent in the difficulties experienced with the engines, more particularly with the high-pressure type. The carbonization and consequent sticking of the piston rings on either very light or very heavy loads is a constant source of trouble, as is also the rapid wearing down of rings and cylinders, necessitating reboring about every three years, and reported as due either to faulty lubrication or lack of a crosshead in the use of trunk-type pistons.

Character of Load	Date, Units Put in Operat'n	Number of units	Individual b.hp.	Type	Vertical or horizontal	No. of cylinders	Diameter of cylinder and length of stroke	Rated speed, r.p.m.	A.c. or D.c.	Load factor, per cent.	Kind of Fuel and Source	Sp.gr., deg. Bé.	B.t.u. per lb.	Price of fuel, cents per gal.	Operating Cost (Cents per Kw.-hr. / Cents per B.hp.-hr.) Fuel	Labor	Lubricants	Supplies	Repairs and Maintenance	Total	Character of attendance necessary	Trouble	Remarks
Central station..	1905	3	225	Diesel	V	3	16x24	164	D.c	50	Penn fuel	30	19 832	3 0	0 33	0 3	0 05	0 01	0 3	0 99	Best	Some	
Central station...	2-1907 2-1910	4	225	Diesel	V	3	16x24	164	A.c	95	Fuel	19	18,000	3 5	0 42		0 024				High grade	Yes	
Industrial, 55-hr. week†	1910	1	170	De La Vergne	H	2	17x27½	180	A.c.	65	Penn. fuel	29	19,000	2 9	0 198	0 28	0 056				Medium	None	
Industrial, 10-hr. day†.	1911	1	125	De La Vergne	H	1	20x34½	155	D.c.	70	Texas fuel	27	19,500	3 0	0 205	0 4					Medium	None	Variable load
Industrial, 8-hr. day.	1911	1	225	Diesel	V	3	16x24	164	D.c.	70	Texas fuel	27	19,000	3 0	0 4	0 306	0 044			0 75	Medium	None	
Refrigerating and lighting	1911	2	120	Diesel	V	3	12x18	225	A.c.	60	Kansas fuel	34	19,600	2 75	0 28	0 16	0 09		0 003	0 533	Best	None	
Central station*.	1909	3	450	Diesel	V	3		150	A.c.	75	Mexican gas oil	38		3 8	0 364	0 321	0 076	0 055	0.152	0 968	High grade	None	
		3	500		V	3																	
Industrial, 10-hr. day..	1909	1	170	De La Vergne	H	2	17x27½	180	D.c.	54	Penn. residual Crude	26	19,500	2 25	0 236	0 371	0 061	0 034	0.151	0 853	Medium	None	
Industrial, 10-hr. day..	1910	1	85	De La Vergne	H	1	17x27½	165													Medium		
Industrial, 11-hr. day. .	1911	1	125	De La Vergne	H	1	20x34½	155	A.c. D.c.		Fuel			3 15							Medium	Yes	
Oil pumping.....		1	125	De La Vergne	H	1	20x34½	155															
Industrial, 11-hr. day	1905	2	75	Diesel	V	3	10½x15	250	D.c.	40	Illinois crude	30	19,000	2 3	0 27	0 113	0 071	0 019		0 773	High grade	None	
Central station..	2 1905	2	225	Diesel	V	3	16x24	167	A.c.	41	Illinois fuel	40	19,500	2 73	0 26	0 207	0 04	0 055		0 562	Medium	Some	
	1-1907	1	120				12x18	225															
Central station, 20-hr day		2	225	Diesel	V	3	16x24	164	A.c.	80	Fuel			3 0	0 38	0 29	0 03	0 11		0 81		Yes	
Industrial, 54-hr week	2-1906 2-1910	4	225	Diesel	V	3	16x24	164	A.c.	73	Penn. fuel	29	19,000	3 1	0 295	0 144	0 045	0 01	0 217	0 711	Medium	Some	
Refrigerating..		2	225	Diesel	V	3	16x24	164															
		1	120		V	3	12x18	220	D.c.		Fuel	28	19,000	4 0								
Irrigation, pumping...		1	180	De La	H	2	17x27½	200			California crude	17	18,500								Medium	None	
		1	260	Vergne	H	2	20x34½	164															
	1-1908																						
Industrial, 10-hr. day	3-1909 2-1910	6	225	Diesel	V	3	16x24	164	A.c.	83	Penn. fuel	28	19,443	2 75	0 283	0 176	0 084	0 07	0 09	0 703	Medium	None	
Central station†..	1911	1	170	Diesel	V	3	14x21	200	A.c.	25	Missouri fuel	19	18,000	3 56	0 339		0 049				Medium	None	
Central station..	1910	1	225	Diesel	V	3	16x24	164	A.c.	28	Illinois crude	25		1 77	0 219	0 484	0 075	0 002	0 05	0 83	Medium	Some	
Mining plant†....	1911	1	125	De La Vergne	H	1	20x34½	155		64	Kansas fuel	27	19,000	13 0	0 778	0 468	0 078	0 087		1 411	Medium	Some	
Industrial, continuous..	1907	3	225	Diesel	V	3	16x24	168	A.c.	80	Fuel	31		2 5	0 245	0 161	0 03		0 07	0 506	Best	Yes	Engines overhauled every 10 days
Industrial.....		1	15	Hornsby	H	1	12x16	260						2 5							High grade	None	
		1	25	Ackroyd		1	14x20	220															
Pumping.....														3 5							High grade	Yes	Steam plant in reserve
Industrial, continuous†		16	125	H.-A.	H	1	27x36	150	D.c.	75	Texas fuel	27	19,000	2 66	0 293	0 15	0 05		0 2	0 693	High grade	Yes	
		6	225	Diesel	V	3	16x24	165							0 184	0 15	0 06		0 25	0 644			
Central station, 131-hr. wk	1911	1		De La Vergne										2 0							Medium	None	
Industrial, continuous	1908	16	225	Diesel	V	3	16x24	164	A.c.	73				2 3	0 189	0 121	0 032	0 01		0 352	High grade	Very little	
Pumping, continuous.	1910	1	125	De La Vergne	H	1	20x34½	155		32	Colorado fuel	32	19,530	3 0							Medium	None	
Industrial, 10-hr. day.		1	85	De La Vergne	H	1	17x27½	164	A.c.	85	Penn. fuel			3 25							Medium	Very little	
Industrial		6	225	Diesel	V	3	16x24	164	A.c. D.c.		Texas fuel	29	19,500								High grade	Very little	
Central station.	1907	2	225	Diesel	V	3	16x24	164															
Central station .	1910	1	120	Diesel	V	3	12x18	225	A.c.		Indiana fuel			3 0							Medium	None	Auxiliary to steam plant.
Central station.	1906	1	170	Diesel	V	3	14x21	200	A.c.		Penn. fuel	29	19,000	3 5							High grade	None	
		1	225				16x24	164															
Industrial, 10½-hr. day.		1	225	Diesel	V	3	16x24	165	D.c.	48	Illinois fuel	30	19,200	2 2	0 25	0 55	0 05	0 32	0.14	1 31	Medium	None	Variable load
		1	120				12x18	220															
Industrial.....	1910	1	90	De La Vergne	H	1	17x27½	180		78	Illinois fuel	29	19,000	3 5							Medium	Some	
Oil pumping...	1911	1	85	De La Vergne	H	1	17x27½	180			Illinois fuel	32	19,100										
Central station.	1905	2	120	Diesel	V	3	12x18	225	A.c.	50	Illinois crude			2 0	0 25	0 33	0 082		0 5	1 163	High grade	Very little	
	1910	1	225				16x24	170															
Industrial, continuous. .	1910	2	195	Diesel	V	3	14x21	200	D.c.	60	Illinois fuel	29	19,000	4 0	0 501	0 278	0.103		0.049	0.931	High grade	Some	

*The 450- hp. engines were built by the Ausgburg Co. and the 500-hp. engines were built by Sulzer Bros. †The costs have been given on the b.hp.-hr. basis; all others are per kw.-hr.

The low-pressure type has been more or less free from difficulties but has suffered from carbonization of oil in the vaporizer, as well as a tendency of the engines to loosen on their foundations. In either type of oil engine, where the engine has more than one cylinder, great care must be exercised in correctly aligning the crankshaft and equalizing the work done by each cylinder. If these points are not regularly checked, liability of springing or damaging the crankshaft exists. In practically all cases the regulation has been reported to be satisfactory. Where the low-pressure oil engine is installed there does not seem to be the same stress placed upon the necessity for high-grade attendants. This is possibly due to the character of service deemed satisfactory where they happened to be used. In plants with engines of the high-pressure type it appears to be an almost unanimous opinion that skilled operators should be employed. The valves must be frequently reground to prevent loss of efficiency and capacity through leakage. There are other small adjustments which require attention to maintain the original quality of the engine.

National Gas Engine Association Convention

The convention of the National Gas Engine Association, held in Milwaukee, formally opened on Wednesday, June 19, with an address of welcome by Mayor Bading, of that city, and a response in behalf of the society by H. W. Bolens, of Port Washington, Wis.

Prominent in the morning's session was an address by P. S. Rose, of the publicity committee, who called attention to the possibilities of the conservation of our food supply through the displacement of farm-work animals by gas engines. He showed that fully one-fourth of the tillable land in the United States is given over to the support of horses and mules, useful for power alone, and emphasized the fact that as the gas engine displaces the horse, this land becomes available for agricultural purposes.

Two formal papers were read during the morning session, "The Adaptability of the Large Gas Engine in the City," by H. W. Jones, vice-president of the association, and "Some Lubrication Problems," by A. E. Potter, of New York. Mr. Jones gained instant attention by the statement that there were more gas engines built in Wisconsin last year than there were steam engines built in the whole United States during the same period. His paper was largely a plea for greater publicity regarding the economy and reliability of the gas engine and that its use completely solves the smoke nuisance.

Mr. Potter's paper was quite technical; he analyzed the difficulties of lubricating the various parts of different types of engines and dwelt on the characteristics of various oils. Interesting among his conclusions was that a gas-engine cylinder oil may be of too high a fire test, failing to burn in the combustion chamber, accumulating and fouling the ignition system and causing misfiring. He recommended the sale of oil to the consumer in sealed packages bearing trade names.

In the discussion which followed, it was stated that most engine manufacturers recommend "any good gas-engine oil" and leave the customer to struggle with the lubrication problem, a practice which Mr. Potter unqualifiedly condemned.

Following the presentation of these papers, E. L. Tracy, formerly state oil inspector of Wisconsin, spoke on the methods used in the state inspection. He contended that the determination and certification of the specific gravity of gasoline or oil protected nobody; that large purchasers could obtain the information much cheaper themselves and small consumers had to buy what was offered them, anyway. He showed that the inspection in all the Central and Western states was similar to that in Wisconsin, furnishing only fees for political purposes. After Mr. Tracy's talk, resolutions were adopted by the association condemning the inspection system now in vogue as a useless burden on the industry which the consumer had to carry.

Wednesday evening the regular banquet of the association was held in the fern room of the Hotel Pfister. A pleasant feature of the evening was the presentation of a pair of field glasses to O. M. Knoblock, treasurer of the association, the occasion being his birthday.

On Thursday, four formal papers were presented: "The Storage Battery in Isolated Lighting Plants," by Taliaferro Milton; "Oil Power," by Edward T. Adams; "Dirty Spark Plugs and Missed Explosions in the Sales Department," by Frank J. Whiteside, and "Electric Farm Lighting Systems," by Otto Borcherdt.

Mr. Milton called attention to the many uses for power about large country estates and advocated the installation of central electric plants with electric drive for the individual machines. A storage battery gives 24-hr. service for lighting and will also serve to run small motors when the main plant is shut down, resulting in increased convenience and economy and decreased fire risk as compared with direct gas-engine driven machines.

Mr. Adams analyzed the oil and gasoline situation, which is fast becoming acute, and showed that kerosene must soon supply the demand for power which gasoline has created and that we are even now entering the age of oil power.

A valuable point brought out by Mr. Whiteside in his paper on defects in selling organizations, was that too frequently letters of inquiry are not carefully read and answered, resulting in disgruntled customers and lost business. The meeting of Friday opened with a general discussion on how to best help the dealer increase his sales, during which the dominant and fraudulent advertisement were both severely scored. The day's program was given up to publicity and sales increase, the papers presented being: "Hit and Miss Publicity," by E. J. Baker; "Publicity through the Trade Journals and What It Means to the Manufacturer," by E. R. Shaw; "Essentials of Foreign Trade Development," by Franklin Johnson; "Direct Advertising to the Dealer," by C. H. Hall, and "The Small Farm Tractor," by L. W. Ellis. Mr. H. E. Miles also spoke on industrial education. The alarming growth of illiteracy in our native white population and the present state of social unrest moved Mr. Miles to advocate continuation schools, somewhat after the German type, in connection with factory organizations. This he pointed out would make for skilled and efficient workmen as well as better citizens. Before adjournment the association adopted resolutions asking the Post Office Department to modify its attempted ruling requiring technical magazines to be shipped by freight.

The Gas Engine Show

In connection with the convention there was held in the auditorium the first annual gas-engine show of the association, about seventy-five manufacturers of gas engines and accessories taking advantage of the occasion to display their products. The show was unique in being the first of the kind where engines could be run under normal operating conditions, making possible for the first time the satisfactory demonstration of systems of starting and control. Engines aggregating about one hundred cylinders were in operation and it is a tribute to the present perfection of the gas engine that so many engines could exhaust into a single room and the room remain habitable as a place of exhibition.

A tugboat engineer visited the engine room of a large ocean steamer. One of the engineers on watch showed him around the engine room and explained the operation of various devices. Stopping at the gage board, where all of the needles were pointing to zero, the visitor looked at the vacuum gage and said:

"What vacuum do you carry?"

"Anywhere from 24 to 26 in.," the engineer replied.

"Gee!" said the tug man, "is that all?"

"How much did you think we ought to carry?"

"Why we carry 24 on my little tug, and I thought a big ship like this would carry at least 40."

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Friction Formulas for Commercial Pipe*

By Ira N. Evans

To determine the best formula for the friction of water in commercial wrought-iron pipe as used in heating work, the results from several standard formulas are compared in the accompanying curves and tables. The results were figured analytically and plotted for sizes from 1 to 12 in. and velocities from 1 to 15 ft. per second inclusive.

Most of these formulas were based and developed on water-works practice for cast-iron pipe where a considerable increase in head is contemplated due to the tuberculating action of the water on the interior surface. Comparatively few tests have been made on wrought-iron pipe below 8 in. in diameter. Tests on galvanized-iron pipe show an especially high friction factor even when the pipes are new. Wrought-iron pipe has a comparatively low friction factor, and the results of the formulas vary between these limits. The greatest variation is found in small pipes, such as are most used in heating work. Here water has no action on the pipes if the piping is tight and the water changed infrequently. The high velocities and continuous service throughout the heating season tend to improve the surfaces and reduce the loss in friction head.

Table 1 gives the results from the several formulas so arranged that the loss in head for each size and velocity may be read according to each authority. The actual commercial diameters are used. Table 2 gives the discharge in cubic feet, gallons and pounds per minute for each size and velocity. The temperature of the water is taken at 100 deg. F. and the pounds and gallons per cubic foot as 62 and 7.48 respectively. Table 2 is probably not closer than the third or fourth figure to the actual results, due to the decimal expressing the area in square feet.

The curves are plotted from Table 1 and the averages of all results are given in the table and also plotted. In addition to the results obtained from the formulas several actual tests have been plotted and averaged for different sizes. At a glance the curves show which formulas are the closest to the average for the various sizes and velocities.

For hot-water heating by forced circulation the average of these formulas should give a safe and proper friction head. Each formula represents the best results of a long series of investigations by that particular authority so the average of all should be very nearly correct. Some of these formulas give high results for small sizes and low for large, and *vice versa*. The curves are differently related as the pipe sizes increase, the widest variation occurring with the high velocities.

The two tests by Poché on 1-in. wrought-iron and galvanized-iron pipes were evidently carefully made and no doubt are very accurate. Each test coincides exactly with one of the formulas.

A study of these curves in connection with some of the published matter on this subject will show the danger of using a table of friction heads and discharge quantities in designing work without first investigating and checking the data. Curves plotted to small scale cannot be read accurately so that anyone having continuous use for these data should work out tables independently. It will then be possible to judge the weight of each factor and use the data with confidence of obtaining the proper results for the governing conditions.

Tabular data and charts giving the discharge and friction head for certain pipe sizes are likely to lead to error, due to the many factors involved, unless these factors are specifically expressed. The plotting should be between friction head and velocity in feet per second. The actual diameter instead of the nominal does not affect the friction head as it does the discharge. If this suggestion is not followed it is almost impossible to check the curves and tables except by trial and error.

The author had two or three seemingly good charts on this subject, but they were useless as the plotting was between discharge in gallons and friction head. An article by J. N. LeConte in the *Journal of Electricity, Power and Gas* for Jan. 27, 1912, purporting to give a series of friction factors from Williams & Hazen's formula to be used in the old formula

$$h = f\frac{l}{d}\frac{V^2}{2g}$$

gave the author a great deal of trouble. The Williams & Hazen coefficients gave very high results when plotted, and when the original data were looked up it was found LeConte had used a friction factor of 100, which was for cast-iron pipe 17 years old and utterly unsuited for the purpose of this article.

Several have claimed that the viscosity of water affects the friction head and it may, but the roughness of the surface of the pipe seems to vary results more than any other factor and is the one most difficult to determine for any given case. It requires assuming a constant to fit the probable roughness of an unknown pipe.

Some of the tests given were conducted by Saph and Schoder, at Cornell University, but were on old pipes and galvanized in some cases, which made the results high. Those by Poché were published in Power, July 25, 1911, and were on pipes in perfect condition. The results are therefore nearer the requirements for the conditions under consideration.

Old tuberculated and galvanized pipe show such variable and uncertain results that the data obtained are practically useless for the purpose in hand. Anyhow, in hot-water heating work, if the piping deteriorates to any extent, it does so rapidly, manifests itself in leaks and has to be replaced.

Some authors have taken the liberty of varying the friction factor according to their personal judgment, and the reader using it sometimes errs again in adding an additional safety factor to be on the safe side. The friction factor may also vary greatly if care is not used in the erection of piping. The ends of pipes entering fittings and couplings should be carefully reamed to remove any burr left in cutting and threading.

In hot-water heating, as far as design of mains is concerned, the friction factor is not so important if the formula gives results which are not high for one size and low for another for changes in velocity. Using a high factor would not especially affect the circulation of the different circuits as they would still balance. The designs of the pumps and prime mover are, however, affected. If the head of the pump is greater than required for the system for a given discharge, the latter will be increased somewhat and the prime mover, if a motor, may be overloaded.

Formula 1, that of Williams & Hazen, seems to coincide with the average values throughout nearer than any of the others.

$$V = cr^{0.63}\left(\frac{h}{l}\right)^{0.54} 0.001^{-0.04} \qquad (1)$$

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TABLE 1. FRICTION HEAD PER 100-FOOT OF PIPE

Size Pipe, In.	Formula or Test	Velocity Ft. per Second														
		1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	Poche, galv. iron	1.17	2.83	6 16	10 83	16 7	23 3	31 3	40 2	50 05	59 9	70 2	81 0			
	Poche, wrought iron	0 5	1 82	4 16	7 17	10 14	13 09	19 41	25 0	31 00	37 53	45 6	54 3			
	Fanning	0 675	2.3	4 68	7 95	12 17	17 13	22 8	29 3	36 5	44 6	53 3	62 9			
	Meier	0.85	3 1	6 5	12 00	16 7	23 54	31 36	40 2	50 05	60 89	72 7	85 5			
	Harrison	0.45	1.81	4 08	6 83	10 2	14 3	19 0	24 5	30 5	37 3	44 7	52 8			
	Sercinsky	0 8	2.6	5 75	10 00	14 95	20 8	27 5	36 00	45 00	55 0	66 00	75 0			
	Darcy	0 691	2 8	6 22	11 05	17 3	24 9	33 85	44 2	55 6	69 0	83 6	99 5			
	Williams & Hazen	0 75	2.7	5 7	9 72	14 69	20 6	27 4	35 14	43 63	53 0	62 35	71 86			
	Sapp. Schroder, old galv.	0 76	3 04	6 84	11 9	18 4	26 1	34 8	45 5	56 7	70 0	82 5	97 0			
	Average	0 74	2.6	5 59	9 72	14 6	20 42	27 5	35 56	48 86	54 15	64 55	75 4			
1¼	Fanning	0 5	1.8	3 54	6 01	9 19	12 94	17 28	22 14	27 7	33 6	40 4	47 48			
	Meier	0 6	2.17	4 65	7 8	11 84	16 7	22 3	28 5	35 6	43 3	51 6	60 7			
	Harrison	0 3	1.09	2 44	4 34	6 79	9 77	13 3	17 4	21 98	27 14	32 84	39 1			
	Sergensky	0 49	1.65	4 35	6 85	11 00	15 4	20 5	27 00	34 0	40 8	48 5	56 5			
	Darcy	0 46	1 85	4 2	7 4	11 6	16 7	22 74	29 7	37 6	46 4	56 1	66 8			
	Williams & Hazen	0 54	1.95	4 15	7 05	10 66	14 98	19 87	24 77	31 68	38 45	45 72	53 42			
	Average	0 48	1 75	3 9	6 57	10 18	14 41	19 33	24 92	31 43	38 3	45 86	54 0			
1½	Fanning	0.414	1.47	3 02	5 14	7 83	11 03	14 72	18 93	23 58	28 8	34 53	40 6			
	Meier	0.49	1.8	3 75	6 5	9 8	13 8	18 4	23 6	29 4	35 65	42 67	50 2			
	Harrison		1.2	2 72	4 54	6 82	9 56	12 72	16 3	20 2	24 8	29 8	35 2			
	Serginsky	0.38	1.53	3.44	5 75	8 55	12 00	16 4	21 00	26 5	32 0	38 00	45 0			
	Darcy	0 37	1.5	3.67	5 98	9 35	13 4	18 3	23 94	30 3	37 4	45 25	53 96			
	Williams & Hazen	0.45	1.63	3.46	5 89	8 91	12 5	16 56	21 31	26 82	32 18	38 38	45 0			
	Average	0.42	1.52	3.34	5 63	8 54	12 05	16 2	20 85	26 13	31 8	38 1	45 0			
2	Fanning	0.286	1.06	2.25	3 88	5.94	8 33	11 17	14 3	17 87	21 7	26 04	30 7			
	Meier	0.36	1.31	2 75	4 75	7 2	10 12	13 5	17 3	21 5	26 2	31 23	36 72			
	Harrison	0.23	0.92	2 04	3 41	5.12	7 76	9 54	12 2	15 2	18 6	22 3	26 4			
	Serginsky	0.34	1.12	2.34	3 9	5 8	8 2	11 00	14 2	18 3	22 0	26 00	31 0			
	Darcy	0.27	1.07	2 41	4 29	6 7	9 61	13 13	17 15	21 7	26 8	32 4	38 6			
	Williams & Hazen	0.34	1.22	2 6	4 43	6.7	9.36	12.48	15 98	19 87	24 12	28 66	33 7			
	Saph., galv. iron	0.226	0.865	1.83	3 12	4 71	6 6	8 9	11 3	14 1	17 0	20 4	24 0			
	Schroder, old pipe	0.315	1.24	2.77	4 9	7 63	10.9	14 8	19 3	24 06	29.9	36 1	42 76			
	Average	0.3	1.1	2 37	4 08	6 23	8.88	11.84	15 22	19.08	23.3	28 0	33 0			
2½	Fanning	0.232	0.861	1.86	3 2	4 9	6.88	9 18	11.84	14 8	18 0	21 7	24 8			
	Meier	0.28	1.05	2.25	3 8	5 75	8 44	10 8	13 9	17 26	21 0	25 1	29 5			
	Harrison	0.24	0.72	1.63	2 72	4 1	5 72	7 62	9 8	12.22	14 92	17 9	21 0			
	Serginsky	0.255	0.795	1.7	2 89	4 35	6 2	8 2	10.9	13.6	16.5	19.8	24 0			
	Darcy	0.21	0.85	1.91	3 39	5 31	7 65	10.41	13.6	17.21	21 25	25 71	30 6			
	Williams & Hazen	0.27	0.98	2.1	3 57	5 39	7.56	10.05	12.89	16.03	19.5	23 22	27 25			
	Average	0.24	0.88	1.91	3.26	4.98	7.07	9 38	12.16	15.2	18.53	22.24	26 2			
3	Fanning	0.18	0.68	1.47	2 52	3 78	5 47	7 29	9.37	11.7	14 27	17 2	20 4			
	Meier	0.22	0.8	1.7	2.85	4.4	6 2	8 92	10.6	13 14	16 0	19 1	22 45			
	Harrison	0.151	0.604	1.36	2.28	3.41	4 78	6 36	8 16	10.1	12.4	14 9	17 2			
	Serginsky	0.202	0.645	1.35	2.32	3 5	5 0	6.7	8.8	10 9	13.2	15.7	19 0			
	Darcy	0.167	0.644	1.45	2.58	4 03	5.8	7.9	10.3	13.05	16.11	19.5	23 2			
	Williams & Hazen	0.216	0.77	1.63	2 79	4.22	5 9	7.85	10.12	12.54	15.21	18.4	21 28			
	Saph.& Schroder,w.i. pipe	0.161	0.609	1.32	2.29	3.5	4.96	6.7	8.6	10.7	13.2	15.8	19 0			
	Average	0.18	0.68	1.47	2.52	3.84	5.45	7.4	9.42	11.73	14.34	17 2	20 35			
3½	Fanning	0.153	0.58	1.25	2 15	3 29	4.64	6.21	8 0	9.97	12.15	14 6	17 2			
	Meier	0.18	0.66	1.42	2.45	3.65	5.2	7.0	9.0	10.9	13 2	15 9	18 6			
	Harrison	0.13	0.52	1.17	1.95	2 94	4.07	5.42	7.0	8.72	10.65	12 8	15 1			
	Serginsky	0.165	0.53	1.12	1.9	2.84	4.1	5.5	7.2	8.8	10 7	13 0	15 5			
	Darcy	0.134	0.535	1.2	2.14	3.34	4.81	6 55	8.56	10 83	13.37	16 18	19 25			
	Williams & Hazen	0.18	0.643	1.36	2.32	3.5	4.87	6.42	8.36	10.4	12.62	15 05	17.7			
	Average	0.17	0.58	1.25	2.15	3.26	4.62	6.20	8.02	9.93	12.12	14 6	17.2			
4	Fanning	0.132	0.503	1.09	1.87	2.87	4 04	5 41	7.0	8.68	10.6	12 75	15 0	17.46	20 1	22 9
	Meier	0.157	0.565	1.23	2.04	3.15	4.4	5.9	7.4	9.7	11.37	13.6	16 0	18.52	21 26	24 2
	Harrison	0.113	0.45	1.02	1.71	2.56	3.58	4.77	6.12	7 64	9.38	11 1	13 2	15 3	17 5	19 86
	Serginsky	0.133	0.45	0.955	1.6	2.4	3.4	4.65	6 0	7.4	9 1	10 9	13 2	15 2	17.64	20 25
	Darcy	0.115	0.46	1.04	1.85	2.88	4.15	5.65	7.38	9.34	11.53	13 96	16 61	19 6	22 6	25 95
	Williams & Hazen	0.155	0.55	1.19	2.02	3.06	4.29	5 71	7.66	9 09	11.04	13 16	15.4	17 86	20 45	23 18
	Saph. Schroder	0.118	0.46	1.01	1.76	2.72	3.88	5.24	6.8	8 55	10.5	12.7	15 0	17.46	20.1	22 9
	Average	0.132	0.5	1.22	1.83	2.8	3.96	5.33	6.91	8 62	10.5	12.6	14.92	17 34	19.52	22 75
5	Fanning	0.103	0.39	0.85	1.47	2.25	3.16	4.23	5.45	6.8	8.3	9 94	11 7	13 6	15 6	17 7
	Meier	0.118	0.425	0.91	1.6	2.35	3.35	4.4	5.75	7.0	8 6	10.2	12 0	13 96	16 0	18 23
	Harrison	0.097	0.39	0.816	1.36	2.05	2.86	3.81	4.9	6.11	7 46	8.95	10 5	12 3	14 15	16 11
	Serginsky	0.102	0.335	0.7	1.18	1.8	2.52	3 48	4.4	5.5	6 85	8 15	9 8	11 45	13 23	15 13
	Darcy	0.088	0.35	0.79	1.41	2.21	3.18	4 33	5.65	7 15	8.83	10.7	12 7	14 9	17 3	19 9
	Williams & Hazen	0.12	0.43	0.909	1.55	2.34	3.27	4.36	5.58	7 07	8 42	10.05	11 8	13 68	15 62	17 9
	Saph. Schroder	0.099	0.37	0.807	1.4	2.14	3.03	4 10	5.25	6 58	8 05	9.7	11.4	13 3	15 3	17 4
	Average	0.13	0.39	0.83	1.42	2.16	3.05	4.10	5.28	6.6	8.07	9.67	11 4	13 31	15 31	17.5
6	Fanning	0.084		0.69	1.2	1.84	2.58	3.46	4.5	5.54	6 77	8.12	9.58	11 06	12 82	14 76
	Meier	0.08		0.73	1.25	1.9	2.63	3.5	4.5	5.6	6 8	8.5	9.6	11 09	12 73	14 5
	Harrison	0.062		0.68	1.13	1.7	2.38	3.18	4.08	5.09	6.21	7.45	8 8	10 44	12 11	13 9
	Serginsky	0.06		0.552	0.945	1.43	2.01	2 76	3.5	4.4	5.45	6.5	7 7	9 0	10 4	11 84
	Darcy	0.071		0.643	1.14	1.79	2 75	3 5	4.57	5.79	7 1	8 64	10 3	12 07	14 0	16 07
	Williams & Hazen	0.096		0.73	1.25	1.9	2.64	3.53	4.51	5.62	6 84	8.1	9 44	10 94	12 56	14 36
	Saph. Schroder	0.067		0.506	0.88	1.36	1 94	2 62	3.39	4 26	5.23	6.3	7 5	8 74	10 09	11 52
	Darcy experiment	0.77		0.76	1.13	1.7	2.44	3.33	4 35	5.5	6.79	8.18	9.74	11.43	13 26	15 2
	Average	0.074		0.66	1.12	1.7	2.42	3.23	4.18	5.23	6.41	7.72	9.08	10.6	12 25	14 01
7	Fanning	0.065		0.586	1.02	1.55	2.19	2 93	3.78	4.71	5 74	6 9	8 16	9 52	11.0	12 56
	Meier	0.066		0.6	1.02	1.57	2.17	2 9	3.75	4.7	5.7	6.9	8 0	9 23	10.6	12 04
	Harrison	0.053		0.481	0.85	1.33	1.92	2 62	3.42	4.33	5.34	6.5	7 7	9 02	10 46	12 00
	Serginsky	0.051		0.46	0.78	1.28	1.67	2 28	2.89	3.6	4.4	5.25	6 2	7.2	8 3	9 45
	Darcy	0.06		0.54	0.97	1.51	2.17	2 96	3.87	4.89	6.04	7.31	8 7	10.21	11 84	13 6
	Williams & Hazen	0.081		0.62	1.05	1.59	2.22	2.96	3.8	4.71	5.72	6.83	8 02	9.3	10.67	12 12
	Average	0.063		0.55	0.95	1.47	2.06	2.78	3.6	4.5	5.5	6.62	7.8	9.08	10.48	11.96
8	Fanning	0.056		0.505	0.875	1.34	1.9	2 53	3.26	4 07	4.96	6 0	7.11	8 22	9 5	10 84
	Meier	0.058		0.52	0.875	1.34	1.89	2 5	3.2	4.0	4.8	5.8	6 9	7 86	9 02	10 26
	Harrison	0.057		0.51	0.854	1.28	1.79	2 38	3.06	3.82	4.66	5 59	6 6	7.91	9 2	10 6
	Serginsky	0.043		0.39	0.657	1.01	1.42	1 93	2.45	3.02	3 77	4 5	5 3	6 2	7 17	8 21
	Darcy	0.0524		0.47	0.84	1.31	1.89	2 57	3.35	4.24	5.24	6 34	7 55	8 85	10 27	11 79
	Williams & Hazen	0.07		0.53	0.91	1.37	1.92	2.56	3.26	4 06	4.94	5 9	6 9	8 01	9.19	10.44
	Average	0.56		0.5	0.835	1.27	1.81	2.41	3.1	3.87	4 73	5.7	6 73	7.84	9.06	10.36
10	Fanning	0.043		0.393	0.68	1.04	1.47	1.97	2.54	3.17	3.88	4.65	5 52	6 4	7.41	8 47
	Meier	0.04		0.38	0.65	1 0	1 42	1 87	2 43	3 0	3.6	4 4	5 1	5.92	6 8	7 73
	Harrison	0.047		0.408	0.683	1.02	1.43	1.9	2.45	3.05	3.73	4 47	5 28	6 15	7 25	8 32
	Serginsky	0.034		0.293	0.5	0.77	1.07	1.45	1.85	2.3	2.8	3.3	4 0	4 73	5 5	6 3
	Darcy	0.041		0.37	0.65	1.02	1.47	2 0	2.61	3.31	4 08	4 94	5 88	6 9	8 00	9 19
	Williams & Hazen	0.054		0.41	0.698	1.06	1.48	1.97	2.52	3.14	3 81	4.51	5 33	6.15	7.06	8 02
	Average	0.043		0.376	0.64	0.97	1.4	1.86	2.4	3.0	3.65	4.4	5 2	6.0	7 0	8 0
12	Fanning			0.32	0.55	0.843	1.2	1.61	2 07	2.58	3.16	3 8	4 48	5 24	6 10	6 89
	Meier			0.31	0.525	0 79	1.12	1.5	1.9	2 4	2 9	3.5	4 0	4 72	5 41	6 16
	Harrison			0.347	0.581	0 871	1.26	1 62	2 08	2.6	3.17	3 8	4 49	5 23	6 1	6 97
	Serginsky			0.235	0 4	0.615	0.87	1.15	1.5	1 8	2.3	2.7	3 1	3 5	3 92	4 5
	Darcy			0.30	0.54	0.839	1.21	1.64	2.15	2.72	3.36	4 06	4.83	5 67	6 58	7 55
	Williams & Hazen			0.33	0.562	0 85	1.19	1.58	2.03	2.53	3.07	3.65	4 28	5.05	5.9	6 69
	Average			0.31	0.53	0.8	1.14	1.52	1.96	2.44	3.0	3.6	4 2	4.9	5 67	6 46

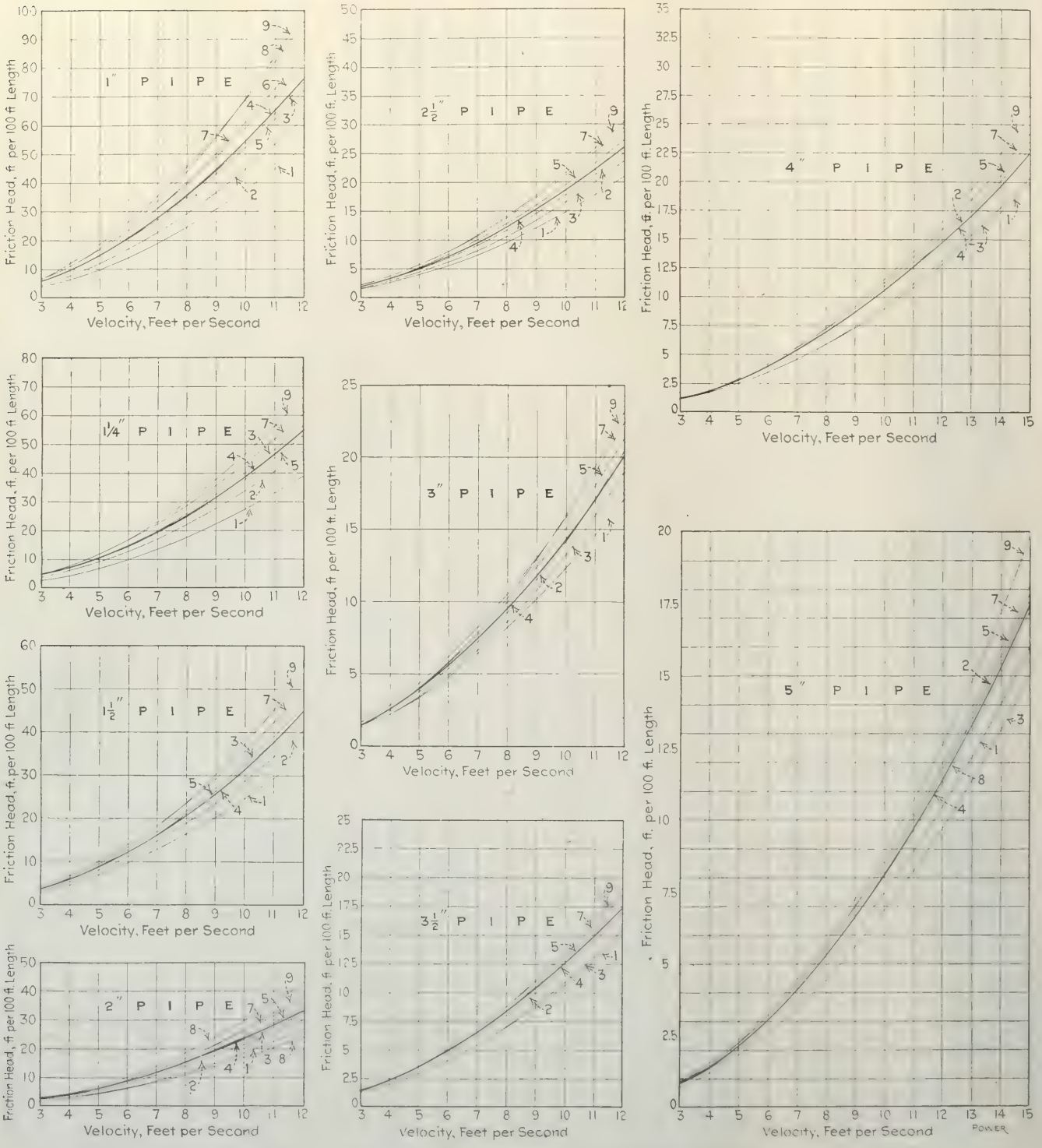
V = Velocity in feet per second;
c = Constant varying with condition of surface;
r = Hydraulic radius = $\frac{d}{4}$ in feet in this case;
h = Friction head;
l = Length of piping in feet;
0.001-0.04 = Factor added to make numerical value of c equal to c in Chezy formula, at a slope of 0.001.

The constants are: 140 for best cast-iron pipe, tin, lead, brass, etc.; 130 for new cast-iron pipe; 120 for cast-iron pipe 5 years old; 110 for cast-iron pipe 10 years old; 100 for cast-iron pipe 17 years old.

Substituting $\frac{d}{4}$ for r and extracting the 0.54 root,

$$h = \left\{ \begin{matrix} 0.0006 \\ 0.00043 \\ 0.00037 \\ 0.00032 \end{matrix} \right\} l \frac{1^{1.85}}{d^{1.166}}$$

1 Harrison Formula, also Poché Tests for 1" Pipe
2 Fanning Formula
3 Serginsky Formula
4 Average
5 Williams & Hazen Formula
6 Poché Tests Galvanized Iron Pipe
7 Meier Formula
8 Saph Schoder Tests
9 Darcy Formula



CURVES FOR FRICTION IN PIPES PLOTTED FROM DIFFERENT FORMULAS

the numerical values in the equation depending on the respective use of the following values for c: 100, 120, 130 and 140.

In the charts and tables the factor 120 gave results for most of the sizes higher than the average. The factor 125 would give results checking very closely all the way through. The greatest variation is shown in the chart for 1-in. pipe; here the average values are high as several tests of 1-in. old and galvanized-iron pipe were included.

No. 2 is Darcy's formula which gives high values throughout. It was never intended for very small pipe sizes, say under 3 in., and was based entirely on water-works practice. The formula is expressed in several ways, but all may be reduced to the same thing. One of the simpler forms of the Darcy formula is:

$$h = 0.003732\, l\, V^2\, \frac{D+1}{D^2} \qquad (2)$$

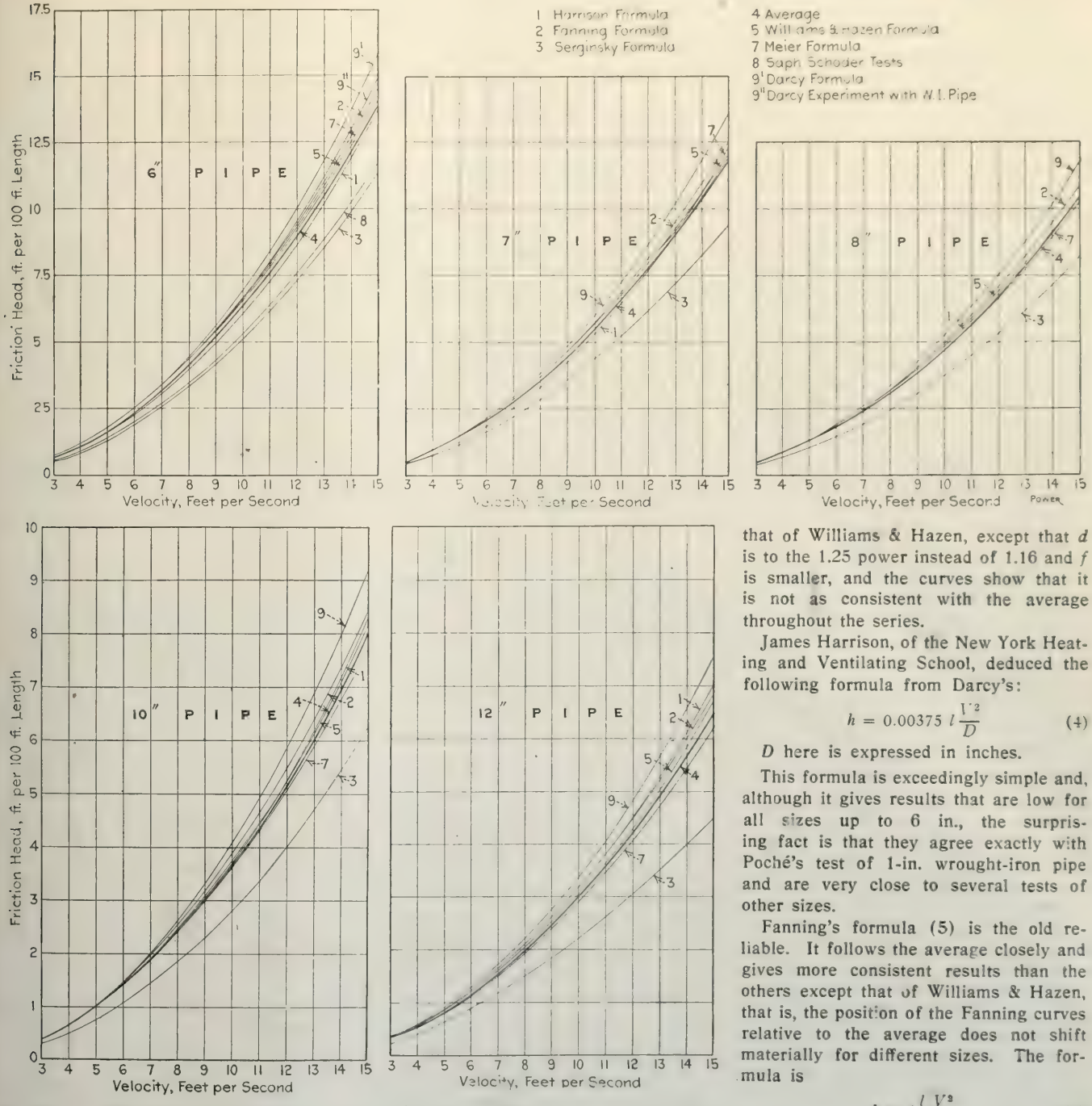
in which

h = Friction head;
l = Length of pipe in feet;
V = Velocity in feet per second;
D = Diameter in inches.

Meier's formula is

$$h = 0.0038\, l\, \frac{V^{1.86}}{D^{1.25}} \qquad (3)$$

Here D is expressed in feet. Mr. Meier increased this constant from 0.00038 to 0.0004 and introduced the factor $2g$ which complicates the formula. The increase of this friction factor or constant makes the results for all small sizes and great velocities high, and above 8-in. pipe it gives results more and more below the average. It may be noticed that this exponential formula is much the same as that of Williams & Hazen, except that d is to the 1.25 power instead of 1.16 and f is smaller, and the curves show that it is not as consistent with the average throughout the series.



CURVES FOR FRICTION IN PIPES PLOTTED FROM DIFFERENT FORMULAS (CONTINUED)

James Harrison, of the New York Heating and Ventilating School, deduced the following formula from Darcy's:

$$h = 0.00375\, l\, \frac{V^2}{D} \qquad (4)$$

D here is expressed in inches.

This formula is exceedingly simple and, although it gives results that are low for all sizes up to 6 in., the surprising fact is that they agree exactly with Poché's test of 1-in. wrought-iron pipe and are very close to several tests of other sizes.

Fanning's formula (5) is the old reliable. It follows the average closely and gives more consistent results than the others except that of Williams & Hazen, that is, the position of the Fanning curves relative to the average does not shift materially for different sizes. The formula is

$$h = f\, \frac{l}{d}\, \frac{V^2}{2g} \qquad (5)$$

Calculation with this formula is very burdensome, due to the varying values of f for each change in size and velocity. Tabulation of the final results might just as well be made as of the series of factors, and the formula would then be unnecessary. This formula is commonly used with a coefficient of 0.02 for all cases, but this is only an approximation.

For the tables and curves a complete set of factors for each size and velocity was interpolated and the latest commercial exact sizes for the diameters of the pipe.

Serginsky's formula was compiled from the latest German authorities. Although it was not available the author has confidence in the results as far as computation is concerned. These were read from a logarithmic chart and appear to be high for small sizes and low for the larger sizes. The curve changes its position with reference to the average from high in the 1-in. chart to the lowest on the 12-in. chart.

The data in this article are not in convenient shape to use in practice but charts or tables may be prepared according to any one of the formulas by plotting the results given in Tables 1 and 2. As stated before, the most consistent results are from Williams & Hazen's formula with 120 to 125 as the constant. In the use of this factor there should be a sufficient allowance for practical increase in friction over ideal conditions, but not enough to allow for careless or poor construction.

It is interesting to notice that all of these formulas involve a numerical constant, which is very close to 0.0037 in all cases.

CORRESPONDENCE

Mr. Jahnke's Proposed Coil

In Mr. Jahnke's arrangement of heating coils, shown in Fig. 2, page 842 of the June 11 issue (repeated herewith), appears at least one fault. A coil of 200 ft. of 3-in. pipe will expand considerably and expand more than the return line; therefore he will find the conditions much the same as before. He should put in more ells between the branch tees and the manifolds to allow for expansion.

W. H. H. PLOWMAN.

Philadelphia, Penn.

When Mr. Jahnke changes over in the manner he describes he will be troubled considerably on account of the steam "short-circuiting" unless he has separate drips on each of the coils.

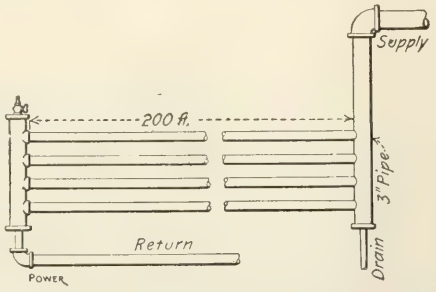


FIG. 1. PRESENT COIL

It would be better to use horizontal manifolds, as in Fig. 2, on the steam and also on the drip of the present coil, Fig. 1. By using elbows and having the drops from and to the headers about 3 or 4 ft. long to allow for expansion, he will not have any trouble with the old coil. The coil should pitch as much as convenient toward the drip, which should be of sufficient size to take care of the condensation.

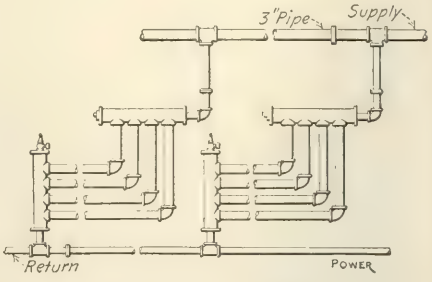


FIG. 2. PROPOSED COIL

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Ipswich, Mass.

TABLE 2. CAPACITIES OF PIPES AT VARIOUS VELOCITIES

Nominal Dia., In.	Actual Dia., In.	Dia., Ft.	Area, Sq.Ft.	Cap. per Min.	Velocity Feet per Second 1	2	3	4	5	6	7	8	9	13	14
1	1.05	0.0875	0.0062	Cu.Ft.	0.372	0.744	1.12	1.5	1.86	2.23	2.6	3.00	3.35	4.8	5.2
				Gal.	2.783	5.6	8.33	11.11	13.9	16.7	19.5	22.24	25.00	36.16	38.9
				Lb.	23.1	46.2	69.3	92.4	115.5	138.6	161.7	185	208	300	323
1¼	1.38	0.115	0.0104	Cu.Ft.	0.624	1.25	1.87	2.5	3.12	3.7	4.4	4.99	5.62	8.11	8.74
				Gal.	4.67	9.34	14.00	18.7	23.3	28	32.7	37.4	42.03	60.71	65.4
				Lb.	38.69	77.4	116	155	193	232	271	309	348	503	542
1½	1.61	0.134	0.0141	Cu.Ft.	0.846	1.69	2.54	3.4	4.23	5.1	5.92	6.77	7.61	11	11.86
				Gal.	6.33	12.66	19	25.3	31.6	38	44	50.6	57	82.3	88.62
				Lb.	52.45	104.9	157	210	262	315	367	420	472	682	735
2	2.06	0.172	0.0236	Cu.Ft.	1.416	2.83	4.25	5.7	7.08	8.5	9.9	11.33	12.7	18.4	19.9
				Gal.	10.592	21.2	31.8	42 4	53	63.6	74.1	85	95.3	137.7	148.3
				Lb.	87.79	175.6	263.4	351	439	527	615	702	790	1141	1230
2½	2.46	0.205	0.0332	Cu.Ft.	1.992	3.98	6	8	10	12	14	16	18	26	27.9
				Gal.	14.9	29.8	44.7	60	74.5	89.4	104.3	119.2	134	194	209
				Lb.	123.5	247	371	494	617.5	741	865	988	1111	1606	1730
3	3.06	0.255	0.0513	Cu.Ft.	3.078	6.16	9.24	12.3	15.4	18.5	21.55	24.6	27.7	40	43.1
				Gal.	23.023	46	69.1	92.1	115.1	138.2	161.2	184.2	207.2	299.3	322.3
				Lb.	190.84	382	572	763	954	1145	1336	1527	1717	2480	2672
3½	3.56	0.297	0.0687	Cu.Ft.	4.122	8.2	12.4	16.5	20.6	24.8	28.9	33.0	37.1	53.60	57.70
				Gal.	30.8	61.6	92.4	123	154	185	205.7	246	277.2	400.4	431.2
				Lb.	255.6	511	767	1022	1278	1534	1789	2044	2301	3323	3578
4	4.02	0.335	0.0884	Cu.Ft.	5.304	10.61	15.91	21.22	26.52	31.8	37.1	42.44	47.7	68.95	74.25
				Gal.	39.674	79.4	11.9	159	198	238	278	318	357	516	555
				Lb.	328.85	658	987	1316	1645	1973	2303	2631	2960	4275	4604
5	5.04	0.42	0.139	Cu.Ft.	8.34	16.7	25	33.4	41.70	50	58.4	66.7	75	108.4	116.8
				Gal.	62.38	124.76	187.	250	312	374	437	500	561	811	874
				Lb.	517.1	1034	1551	2068	3586	3102	3620	4137	4654	6722	7239
6	6.06	0.505	0.201	Cu.Ft.	12.06	24.1	36.2	48.2	60.3	72.4	84.4	96.4	108.6	156.8	169
				Gal.	90.21	180	271	361	451	542	632	722	812	1173	1263
				Lb.	747.7	1496	2243	2991	3739	4486	5234	5982	6729	9720	10,468
7	7.02	0.585	0.269	Cu.Ft.	16.14	32.3	48.4	64.6	80.7	96.8	113	129.2	145.2	209.8	226
				Gal.	120.73	242	362	483	604	725	846	967	1088	1570	1691
				Lb.	1000.7	2001	3002	4,003	5,004	6,004	7,005	8,006	9,006	13,009	14,010
8	7.97	0.665	0.347	Cu.Ft.	20.82	41.6	62.5	83.3	104.1	125	145.8	166.6	187.5	270.7	291.5
				Gal.	155.74	311.5	467.2	623	779	935	1,090	1,246	1,402	2,025	2,180
				Lb.	1290.8	2582	3872	5,163	6,454	7,744	9,035	10,326	11,616	16,783	18,073
10	10.01	0.834	0.55	Cu.Ft.	33	66	99	132	165	198	231	264	297	429	462
				Gal.	247	494	741	988	1,235	1,482	1,729	1,976	2,223	3,211	3,458
				Lb.	2046	4092	6138	8,184	10,230	12,276	14,322	16,368	18,414	26,580	28,644
12	1 .2	1	0.787	Cu.Ft.	47.22	94.44	141.66	188.9	236.1	283	330	378	425	614	661
				Gal.	353.21	706.4	1059.6	1,421.8	1,766	2,119	2,472	2,825	3,179	45,916	4,945
				Lb.	2928	5856	8784	11,712	14,640	17,568	20,496	23,424	26,352	38,060	40,990

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POWER

The Turbine Contest

We trust that the participants in the turbine contest are well endowed with patience. The articles submitted have passed through the hands of two of the judges and by the time this paper reaches the subscribers will be in the hands of the third and last. The judges are all busy and important men, and we are sure that the contestants, as well as the editors, appreciate their kindness in consenting to serve. The conscientious reading and comparing of the merits of the large number of manuscripts submitted involves much time and labor. There shall be no unnecessary delay in arriving at a decision, and it will be announced and the prize awarded, as soon as the judges have spoken.

Unreliable Data

In the office of a prominent consulting engineer, a draftsman, laying out the heating mains for a large building, asked how he should determine the pipe sizes. The consulting engineer brought out a blueprint of tabular results supposed to give the proper size for different lengths and capacities for one pound drop in pressure. Asked if he was sure of the data, he replied: "I guess the figures are all right as they are used by a large company engaged in this line of work." It so happened the table did not apply, as it was based upon altogether different conditions. Such procedure is quite common and has resulted in the design of many unsatisfactory, or at least inefficient systems.

There is hardly a branch of engineering where more actual guessing is employed, than in the sizing of conduits for fluids, as applied to the practice of heating. Even when tables are employed the user seldom knows their origin or limitations and as infrequently checks them carefully. If the results have failed to give trouble through a period of use it is sufficient.

Actual tests on friction of fluids in commercial pipes are comparatively few in number and meager in detail. And in most cases the tests relate to water works practice where the kind of pipe and surface differs from that for heating. What tests there are give a wide variation of results.

Most of the friction formulas are fairly accurate if the correct constant for roughness can be assumed or determined, but calculating a series of results for final sizes and velocities is so laborious that ordinarily pipe sizes for steam and water heating are guessed at, which may be as good as guessing at a coefficient for roughness.

Some interesting data on this subject appear on page 54 of this issue. The article compares in detail the results obtained from several standard formulas for the loss of head in friction due to the flow of water in standard commercial wrought-iron pipe. The results are tabulated and also plotted in curves, and an average in both cases has been determined.

Many authorities no doubt have investigated the subject in much the same way but the results of their investigations appear only in their formulas. The present article is arranged so that the reader can make his own comparisons and form his own conclusions. He can also vary the constant of any formula and judge of the effect on the general result. The tables and curves have been arranged more for purposes of comparison than for easy practical use, but the data can be rearranged easily to suit individual conditions. It is to be hoped that the matter will prove as valuable as it was tedious and laborious for the author to compile it.

The purpose of the investigation was to determine the formula and constant best suited for designing hot-water heating systems and in this connection it would be interesting if readers of POWER in possession of special formulas or tests on commercial wrought-iron pipe would send them in for comparison with the data presented in the article.

Ash and Clinkers

"Little satisfactory information regarding the impurities found in coal, their nature and appearance, action in the furnace and effects on combustion, is to be found in textbooks, or technical literature generally. When the subject is mentioned it is usually dismissed in a few lines, or, at most, in two or three short paragraphs, which discuss merely the effect that ash in general has upon fuel economy and combustion, but do not go into the question of the nature of the ash, or describe the effect which its different constituents have in reducing the efficiency of the coal, and in causing the formation of clinkers," says a writer in *Coal Age*.

This is all too true and, moreover, perhaps due to the paucity of literature on the subject, it is too superficially considered by steam users. The greatest number of heat units per ton at the lowest price is desired in a coal. If it is high in ash it is generally worth less per unit of weight and is far more expensive to handle. But because it is low in ash does not signify that it is really the cheapest coal for the purpose. The nature of the ash may be such that when the bed of ash on the grate reaches the fusing temperature, the most troublesome kind of clinkers will form, especially if the coal contains much sulphur, lime, iron oxide, silica and other impurities which tend to lower the fusing point of the ash.

Ash is not only a total loss when the thermal value of a fuel is considered, but a great amount of heat in the coal is wasted in raising the temperature of the ash to that of the fuel bed. Additional expense is due to the labor necessary to handle it, and the rapid deterioration of the furnace walls and grate caused by the chemical impurities of the coal and accelerated by the frequent use of the slice and tamping bars.

When present in large quantities in the furnace, ash necessitates that the fire-doors be open for a considerable time while removing it. This allows large volumes of air to enter, seriously lowering the CO_2 and conducing to undesirable stresses in the boiler due to extreme temperature variations in its structure.

On the other hand, if the ash readily fuses, it clogs the air spaces of the grate, causing imperfect combustion, smoke and further thermal losses. Some coals form an ash which readily fuses and becomes heavy, black and ropy when hot. Its viscosity is such that it runs over the grate, filling the air spaces and finally crystallizing when cold. The most skillful fireman cannot get good results from such coal, however high it may be in heat value. It is also next to impossible to effectually clean grates once subjected to this treatment.

There is much in this subject of the impurities and clinker-forming properties of ash, and it is to be hoped that the Bureau of Mines and the various engineering colleges and societies will take it up and give publicity to the results of their investigations.

Learning by Doing

The article appearing on another page, "Preparing for License Examination," contains many ideas pertinent to the welfare of steam engineers who would progress in their work.

There are two ways of learning, by study and by experience, but both used together make a strong combination and a power for advancement.

In the case under consideration, study and experience were combined. After reading how to do a job the actual work was performed, and the idea firmly fixed in the mind.

The engineer used problems found about his plant, figured them out, and when he got through, knew a lot more than before the study period began.

Instructions for setting the valves on a slide-valve engine are published in most handbooks dealing with steam engines, but a beginner could read them several times and not obtain so complete an understanding of how the work should be done as would be obtained by performing the operation with an engine, using the printed explanation for a guide.

Other power-plant performances can be worked out in a similar manner. The idea is sound and practical. It can be adopted with profit by those who aim to accomplish things in the steam-engineering field.

On May 7 a conference was held between the Board of Public Utility Commissioners of the State of New Jersey and representatives of nine railroads operating in that state. A memorandum issued by the commission states that "The representatives present were unanimously of the opinion that there was no mechanical device that would prevent the dropping of crown-sheets in locomotive boilers, thereby preventing explosions."

On June 20 the Jacobs-Shupert U. S. Firebox Co. let the water in a locomotive boiler with one of their fireboxes go out of sight and kept up a roaring oil fire in the box until the pressure went down from 230 pounds to not enough to blow the fire without any outward manifestation of distress. A radial-stay boiler subjected to the same treatment dropped its crown-sheet with a great outpouring of steam and flying of hot firebrick and rivet heads when the water line had fallen 14.5 inches below the crown-sheet.

Is it time to call another conference?

The straight-flow steam engine, brought into prominence in recent years, through the work of Prof. Stumpf, of Charlottenburg, which has produced such astonishing results in the use of steam, was patented in England by Todd, in 1886, his patent bearing the number 2136. The Willans engine, which was constructed upon somewhat different lines, involves the same principle.

It is said that the 25,000-kw. Parsons turbine-driven unit recently purchased by the Commonwealth Edison Co., of Chicago, will be run by four boilers, each having 12,000 sq.ft. of heating surface and 250 sq.ft. of grate. This is a ratio of heating to grate surface of 48 to 1, and 1.92 sq.ft. of heating surface and $^1/_{25}$ of a square foot of grate surface per kilowatt.

A card in a Boston electric power house, says the Boston *Transcript*, reads: "To touch these wires means instant death! Anyone failing to respect this warning will be prosecuted and fined." It would be decidedly interesting to know the verdict rendered on the first case successfully prosecuted. We might even be willing to pay the fine.

At the end of fifty-seven years of activity, the Manchester Steam Users' Association is able to boast that no life has ever been lost by the explosion of any boiler under its care.

New Analysis of Indicator Diagrams

Report of the discussion upon J. Paul Clayton's paper before the A. S. M. E. entitled "A New Analysis of the Cylinder Performance of Reciprocating Engines."

This analysis, as explained in POWER, June 18, page 858, is made by transposing an ordinary indicator diagram to logarithmic coordinates.

DISCUSSION

In discussing J. Paul Clayton's paper on the above subject at the recent meeting of the American Society of Mechanical Engineers, Arthur L. Rice, of the *Practical Engineer,* said that he had had occasion to apply the same method of analysis to some diagrams taken upon tests of engines which were of only one hour's duration and found a difference of from 2 to 6 per cent. between the water rate as indicated by the analysis and that found by observation.

Prof. F. E. Cardullo, of the New Hampshire College, said that the law $pv^n = c$ is no doubt a very close approximation to the truth in the case of gas engines and air compression, although it probably is not exactly true. In the case of the adiabatic expansion of steam, if the relation of the pressure and volume is plotted on logarithmic paper the expansion curve is convex toward the origin but the curvature is very slight.

If, instead of assuming that n is a constant it is assumed that n is a variable and that the range of variation during any given expansion is very slight and nearly uniform, we shall have a still more exact statement of the case and it is unquestionable that any appreciable irregularity in the logarithmic expansion or compression line is *prima facie* evidence of leakage or of some other disturbing condition.

The author's statement that the skin surface of the cylinder must be heated once every cycle from the temperature acquired by contact with the exhaust steam nearly up to the temperature of the admission steam is misleading, since the range in temperature of the metallic surface is known to be comparatively small, being only from a fifth to a tenth of the range of the temperature of the steam.

From the author's statement it is to be inferred that the phenomena occurring after cutoff are independent of the form of the cylinder and the relation between the volume and the clearance surface. Professor Cardullo believes that a further study will lead to a change of view in this regard. He did not believe that the method is sufficiently accurate to warrant its use in determining steam consumption, although it will probably be very useful in determining losses from leakage and in showing whether or not an engine is operating under the most economical conditions possible.

He asked how the method would check itself when the steam consumption of the high-pressure and low-pressure cylinders of a compound engine is determined independently by its use. Does it give the same steam consumption for both cylinders as it should? The method is not applicable when the load on the engine is varying, since the temperature conditions of the cylinder walls are those acquired from a previous revolution and the quality at cutoff and the form of the expansion line will be different from what it would be were the engine operating with the same cutoff under a steady load. He offered what he referred to as a "more elegant, more accurate and more simple method for determining the clearance than that proposed," a description of which will be found upon pages 33 to 35 in Professor Cardullo's "Practical Thermodynamics," and will appear in the next issue of POWER.

There is no question but what the logarithmic analysis will show evidence of leakage in most cases. In certain cases, however, when there are simultaneous leaks of inlet and exhaust valves, it is quite possible that no leakage can be detected. There is also the case where the leakage takes place by the bodily transfer of water under the surface of a slide valve, which is the form of leakage most common in high-speed engines. In his opinion there is no possibility of approximating the amount of leakage from the logarithmic diagram. Data in regard to the relation of the quality of the steam at the cutoff and the exponent of expansion obtained from forms of valves known to be subject to leakage should be regarded with suspicion.

If experience justifies the claims of this form of analysis we may shortly expect to see indicators on the market which will draw the logarithmic diagrams directly upon the indicator card. The mathematical relations required are quite simple and there ought to be no difficulty in constructing an accurate and simple apparatus for the purpose. Of course, such an indicator will not draw correct diagrams unless the clearance of the engine has been determined and the apparatus adjusted accordingly. If, however, the apparatus can be adjusted until the expansion lines are straight, it will be possible to use the apparatus without determining the clearance. With regard to hooked diagrams, Professor Cardullo said that he had found a hook due to cylinder condensation in the compression curve of a diagram from a 7x12-in. Straight Line engine running at 250 revolutions per minute. These hooks will always occur with any type of engine when the compression is sufficiently great, is more pronounced in engines having small clearance volumes, and the higher the rotative speed the higher the compression pressure at which they occur.

R. C. Stevens, of the Skinner Engine Co., called attention to the failure of the method to give any account of the steam which leaked from the steam chest through the exhaust side of a piston or slide valve, without ever going into the cylinder. He called attention to the large amount of such leakage shown in the paper presented by Messrs. Dean and Wood to the society in June, 1908, entitled "Economy Tests of High-Speed Engines," and to the large amount of leakage shown by the experiments of George Mitchell and reported in POWER of Oct. 11, 1910, in an article entitled "Leakage Through a Piston Valve," in which the leakage in some cases ran as high as 22 per cent. of the total consumption. Further, in POWER for Nov. 8 of the same year, A. L. Ide & Sons state that the steam consumption of an engine was increased from 26 to 32.1 lb. per indicated horsepower-hour by a valve ten-thousandths of an inch under size, as compared with one having ordinary commercial fit. This represents a leakage of 23 per cent.

What is true of piston valves is also true of the flat-balanced valve with the pressure plate. Tests on one such engine of 80 hp., which the owners considered in good condition, showed 624 lb. of steam leakage per hour, or about 31 hp. wasted, or over 25 per cent. of leakage. It is common practice to allow 3 to 4 thousandths of an inch freedom between the valve and pressure plates when newly fitted, as in operation the valve wears small and the seat wears large. The experiments of Callendar and Nicolson in Canada as early as 1897 (Institution of Civil Engineers, *Proceedings,* 1897-1898, Vol. CXXXI) called attention to the importance of valve leakage as affecting economy and to the fact that the leakage from the steam chest directly into the exhaust is very considerable when the engine is in operation and much more at this time than when the engine is at rest. He had proved that the substitution of an expanding valve which is and remains steam-tight, produces a saving of 20 per cent. or more as compared with the usual plug valve, whether piston or flat, and he would not except the piston valve with packing ring or the Corliss or oscillating valve as used in four-valve high-speed engines from this unfavorable compari-

son, if these engines had seen a few months' service.

He challenged, therefore, Mr. Clayton's assertion that the results of the application of his methods made up to the present, with the restricting conditions imposed, tend to show that the steam consumption of engines may be approximated from the indicator diagrams to within an average difference of less than 4 per cent. from test results, and was forced to believe that Mr. Clayton's tests were made on engines in much better condition than is usual, even with new engines, or that some of his data are unreliable. For, if 20 per cent. or more of the steam that enters the throttle goes out of the exhaust without doing useful work or in any way making its presence felt on the indicator, it is difficult to understand how any analysis of the indicator diagram, or its derived logarithmic diagram, can establish the actual steam consumption of an engine to within even 20 per cent. of accuracy.

Sandford A. Moss, of the General Electric Co., said that this paper was the first new thing in its line that we have had for many years and he believed that it would be a permanent addition to steam-engine theory. Leakage must be taken into account, as Mr. Stevens has said.

Mr. Clayton's method will disclose cases where there are leakages by comparison of actual weight with figured weight upon the slope of the logarithmic lines. It seemed to him that the tests did not warrant any distinction between different pressures and that the authors should use the equation from the average of all pressures, rather than the chart presented. (See POWER, June 18, p. 863.)

Prof. William G. Ennis, of the Brooklyn Polytechnic Institute, said that having been in possession of an abstract of this paper some weeks before its appearance in the *Journal*, he had been able in one or two instances to verify Mr. Clayton's law. A small automatic engine gave $n = 0.96$ with a dryness factor at cutoff of 0.62. A Corliss engine gave $n = 1.02$ average, corresponding with an indicated steam rate of 26 lb. per i.hp.-hr. noncondensing. Some of the indicated tests in Mr. Barrus' book on "Engine Tests" give values of n which seem to be out of harmony with the Clayton law.

We may hope that the accuracy of the method will be firmly established and that a satisfactory expression may be found for the exponent of the expansion curve under other conditions, so that this method of testing may not be limited permanently to the case of engines running noncondensing. Should this hope be realized the general average of indicator practice will have to be greatly improved.

A study of values of n for the compression curve is interesting. Values exceeding 1 have frequently been found along with such absence of high final temperatures as demonstrates that the customary Hirn assumption of initial dryness equal 1 must be incorrect. He had found in two cases with small automatic engines n for compression equal to 1.85 and 1.80.

From the thermodynamic side the paper is of particular importance from the new light it throws on the question of heat interchange. We have been taught in the books that during the early part of expansion the steam continues to give up heat to the cylinder walls; afterward, as its temperature falls, again receiving the heat. But, if the expansion line of the steam diagram conforms to the law, $pv^n = con.$ (and he had before him 18 transferred diagrams in which it does), there can be no such reverse transfer. The steam must be either steadily gaining or steadily losing heat, and, according to Mr. Clayton, it loses or gains according as it is dry or wet at the point of cutoff.

For some time past he had been trying to find a case where this reversal of transfer occurred; that is, where the expansion line recrossed the adiabatic. The only places where this seemed to happen are in the books; and in the light of the present paper he was inclined to think that the apparent reversals have been generally due to leakage. In the Callendar and Nicolson experiments, the steam temperature dropped to that of the metal on the cylinder head in $^1/_{18}$ of a second after cutoff. Considering the lag and dampening of wall-temperature fluctuation, the condition thus favors a gain of heat by the steam practically throughout expansion. Strangely enough, also, the wall temperature taken at a point in the sidewall just inside the face of the head began to fall while the walls were about 17 deg. F. cooler than the steam.

The subject of steam leakage is one of the most important which the paper suggests. In this connection the discussion is especially complete because the Clayton analysis leads to conclusions regarding leakage in both condensing and jacketed engines. A large part of what we have been discussing as cylinder condensation for some years past has certainly been piston and valve leakage. Tests for tightness with the parts stationary give no indication of the leakage conditions in an engine when well warmed and running. We have been in the dark on the whole question not only as a quantitative but even as a qualitative matter.

Leakage between two chambers will, other factors being constant, be proportional to the pressure difference and inversely proportional to the seal of the valve. The diagram sketches these variables and shows that leakage is most likely to occur:

(a) Through the *steam* valve just after cutoff or just before admission.

(b) Through the *exhaust* valve during the early part of a stroke.

(c) Through the *piston* during the early part of a stroke.

In cases *b* and *c* the diagrams are scarcely likely to show leakage until after cutoff has occurred.

Leakage through the steam valve occurring just after cutoff may be shown by a steam-chest diagram. It raises the expansion curve during the early part of expansion. When it ceases the true polytropic curve begins.

Preadmission leakage at this point raises the compression curve and may take place at such a rate as to give a curve apparently polytropic with an extremely high n value. A steam-chest diagram may confirm the diagnosis.

Leakage through either piston or exhaust valve occurring just after cutoff depresses the expansion curve. If it is the piston that leaks, a corresponding rise of pressure may in some cases be noted on the exhaust line of the companion diagram for the other end of the cylinder.

If such leakage occurs before cutoff the diagram in question will not show it, but the compression diagram may show a rapid rise of pressure during the early part of its exhaust period.

Mr. Clayton's study, important and thoroughly presented as it is, is thus far but a beginning. The bearing of the new law on the application of Hirn's analysis and the temperature-entropy diagram for actual tests will compel us to revise our judgment as to the utility of these forms of investigation. A new and undoubtedly fruitful field has been opened to steam engineering.

S. W. Marquis, of the engineering-experimental station of the University of Illinois, spoke of the advantages offered by the method in testing locomotives. The paper contained but one locomotive test, but Mr. Clayton was making a thorough study of the application of his methods to locomotive engines and finding a slightly different relation between the exponent of the curve and the quality of the steam at cutoff than those reported in the present paper.

J. B. Stanwood, of the Houston, Stanwood & Gamble Co., called attention to the logarithmic treatment of the expansion curve in Prof. Perry's discussion of Mr. Willan's paper on "Non-Condensing Engine Valves." (See *Proceedings* of the Institute of Civil Engineers of Great Britain, Vol. XVIII, 1888.) He felt that there was need for further experimental work and that the relation between the exponent and the quality of the steam should be investigated for engines of other classes and under other conditions.

In the instance at hand, leaving leakage out of account, it would seem that the initial quality of steam at cutoff is dependent upon two things: First, the amount of steam actually condensed up to cutoff, and, second, the quality of steam in the clearance space just before the admission. If the quality is based

simply on the difference between steam as accounted for by the indicator at cut-off, and the water consumption as determined by the amount condensed in the condenser he feared that the result is not correct, as no consideration has been taken of the quality of the cushion steam. It may be possible that the quality of cushion steam may be as low as 10 per cent., especially if Mr. Willan's theory is true that water retained on the clearance surface forms an important factor in creating cylinder condensation. Professor Zeuner has stated that if he knew absolutely the quality of steam at any point during the compression he could determine the steam consumption of a steam engine from the indicator card, leakage neglected. This question of the quality of cushion steam may have something to do with the variations in the results obtained by Mr. Clayton in his comparison of different engines. There is quite a difference of opinion on this point of wet or dry cushion steam. Some engineers maintain that the steam is dry during compression; others that there is more or less water contained in the clearance space.

The writer is of the opinion that there are few, if any, engines which are operating without some leakage. In regard to the 12x30-in. engine, tested by Mr. Clayton, the speaker thought there must be quite an appreciable amount of leakage. Mr. Clayton states that this engine, with 109 lb. absolute pressure at the throttle, with 25 per cent. cutoff, has a quality of steam of 58.8 per cent., which means a cylinder condensation of 41.2 per cent., which is high for this cutoff in comparison with most tests that have been made on Corliss engines. George Barrus in his book on "Steam Engine Tests," estimates that the amount of condensation for 25 per cent. cutoff on an average is 24 per cent. If, therefore, there was no excessive leakage in this engine, the experiments show a remarkable difference between condensation in small engines and condensation in larger engines of the same general type.

This question of leakage has been very thoroughly investigated by a special committee of the Institution of Mechanical Engineers of Great Britain, Professor Capper, chairman. (See *Proceedings* for 1905, beginning with page 171.) The experiments conducted by him show the amount of leakage of the valves at rest as compared with the leakage of the valves in motion, and corrections were applied so that the leakage was differentiated from the initial condensation. It is this fact that leads the speaker to believe that the 12x30-in. engine should be tested for leakage, both through the steam-valve system and exhaust valve.

The discovery of Mr. Clayton may lead to the solution of some problems heretofore unsolved. We may be able to determine after some experience, by means of data heretofore unobtainable, the actual quality of steam in the compression spaces.

By comparison of different cylinders of the same general type, manufacturers may be able to predict the steam consumption, especially as to the influence of the size of cylinder.

We may be able to discover the effect of clearance and clearance spaces on the economy of the steam engine by varying the same on a given engine and then from the character of the expansion curve determining the effect of cylinder condensation.

Designers may be able to improve the proportion and construction of steam cylinders and valve-gears by the character of the expansion curve under different conditions analyzed by this logarithmic method.

The speaker feared that a determination of steam consumption of many types of engines by this method would not be very reliable. There are many cylinders in which the steam is short-circuited directly from the steam side of the cylinder to the exhaust side of the cylinder, and no evidence of this loss appears on the diagram, so that the use of Mr. Clayton's method would not be at all satisfactory.

Prof. R. C. H. Heck, of Rutgers College, submitted a discussion dealing particularly with the character of the heat interchange between cylinder walls and steam and drawing the conclusion that except for the extreme cases of very low speed and very short cutoffs, there was nothing in the showing to antagonize the empirical result of practically constant exponent *n*.

In summing up, Mr. Clayton said that he was fully aware of the extent of valve leakage and was at work upon that phase of the subject. Valve leakage by his method in certain cases, by accounting for what was in the cylinder, has been approximated to within 6 per cent. in cases with which he has had experience.

Standard Pipe Flanges and Fittings.

Edward B. Denny, president of the National Association of Master Steam and Hot Water Fitters, in his presidential address at the recent convention at Atlantic City, alluded to this subject as follows: "Most particularly would I call your attention to the report of the Committee on Standardization. It will describe in detail our efforts to establish "The 1912 U. S. Standard" schedule of Standard Weights and Extra Heavy Flange Fittings and Flanges.

"In view of the fact that some violent opposition to this schedule has been expressed by some of the manufacturers of such fittings, and that some misleading statements have been made relative to its adoption, it may not be amiss for me to state a few facts concerning the matter.

"There has been no contention as to the desirability and need of a universal standard for flange fittings and flanges. Neither has there come to our notice any criticism of the schedule adopted as to its intrinsic merits, its adaptation to the requirements of the work for which such fittings are used. On the contrary, we have received many letters of commendation from the very highest authorities for not only working out and adopting a schedule but for the evident care with which "the 1912 U. S. Standard" schedule has been prepared, reflecting, as it does, the theoretical demands of the engineer and the practical requirements of the practical steam fitter.

"So far as we are informed, the opposition to the acceptance of this schedule is based either upon the method of its adoption or the expense entailed for the necessary alterations of some patterns. As to the latter objection, it is a self-evident proposition that no standard schedule can ever be made and come into general use without requiring such changes in patterns. While this may seem a hardship to the manufacturer, he has known for years that it was inevitable when those whose interest demanded the reform should be sufficiently aroused to insist upon having it and had there been a reasonable response upon his side to the overtures made by us as long ago as 1905, this incidental expense would have been avoided to the extent of the cost of many new patterns made since that time. As to the method of the adoption of this schedule, if it shall be said (as it has been) that ample notice was not given to the manufacturers of our purpose to prepare and adopt such schedule, such a statement is not in accordance with the facts, as will be clearly shown by the reports of your secretary and the Committee on Standardization.

"If the right and authority of those who made and adopted the schedule be questioned, we would ask what higher authority can there be on the subject than that of the American Society of Mechanical Engineers, whose members design most of the plants in which such fittings are used and are responsible for their proper dimensions and strength, and the National Association of Master Steam and Hot Water Fitters, whose members install such fittings and are likewise responsible for loss of life and property resulting from so called standards fixed by those whose only interest consists in the profit that may be made on the sale of the goods?

"The subject is one that should receive your careful consideration in the light of all the information contained in the reports of your officers and committees and I recommend that the sentiment of our association in convention assembled be voiced with no uncertain sound."

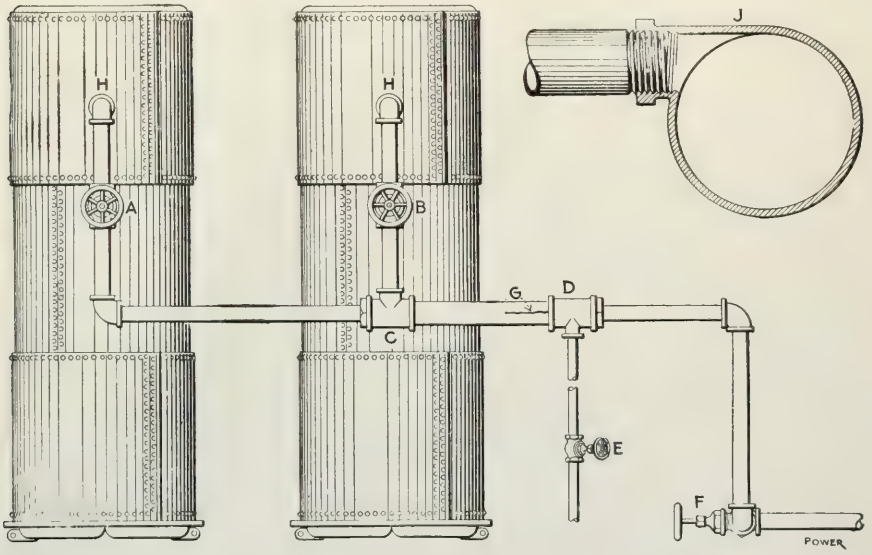
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Dangerous Steam Piping

The illustration shows the piping for a pair of tubular boilers. Each 4-in. riser is followed by an ell and a globe valve, with the stem in a vertical position, as at *A* and *B*. This is bad because it prevents water from draining automatically from the steam pipe. If globe valves are preferred, the stem should be set horizontally. From each globe valve a 4-in. pipe extends, discharging into the 4x6x4-in. tee *C* over the second boiler. This is followed by 6-in. pipe connecting with the 6x4x1-in. tee *D*.

The 2-in. horizontal pipe, which takes steam from *D*, is fitted with a globe valve *E*. Beyond this valve and nipple an ell looks downward, followed by more pipe and a throttle valve used to control a small engine.



SHOWING BROKEN PIPE OF BOILER CONNECTIONS

With a full pressure on these boilers, the valves *E* and *F* were closed for several hours; then *E* was suddenly opened to supply steam for the small engine. This was a cast-iron valve, and as soon as it was opened the whole bottom was blown off; simultaneously the 6-in. pipe burst at *G* for a distance of 10 in. When the pressure was shut off at *A* and *B* no further damage was found.

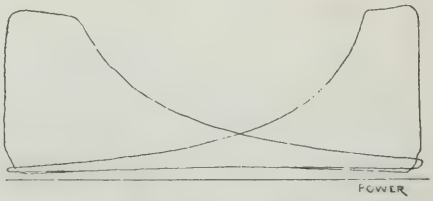
The stop valves *A* and *B* should have been replaced by angle valves, connected where the ells *H* are located. The damaged piece of pipe *G* was replaced with a 6x6x2-in. tee followed by more 6-in. pipe, a 6-in. ell and another piece of pipe carrying a 6x4-in. reducing ell looking downward. This arrangement eliminates some of the danger, as water cannot collect as before. The globe valve *E* was omitted, but a drip pipe was put in the throttle valve and a valve in it is kept open when the engine is shut down. The tee should have been of the eccentric type on the 2-in. outlet, as shown at *J*, as the pipe would have been better drained.

W. H. WAKEMAN.

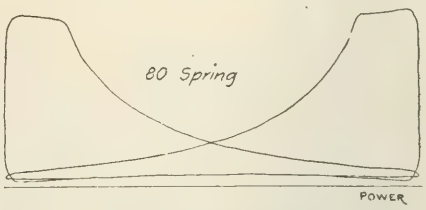
New Haven, Conn.

A Cutoff Problem

These diagrams were taken from our 20 and 40 by 36-in. cross-compound Corliss pumping engines running at 42 r.p.m., at 130 lb. steam pressure and 25.5 in. vacuum. Fig. 1 was taken after blocking the governor in running position and setting the cutoff correctly by the stroke. Fig. 2 was taken after adjusting the cutoff by the indicator diagram. The head-end steam valve cuts off the same at all times no matter at what speed the engine runs, while the crank-end valve holds on until the engine is up to speed. This fault manifests itself in both engines on both high- and low-pressure cylinders. Will some readers please tell me how to correct the fault and what causes it. The valve links on the steam valves are the same length and so are the links on the exhaust valves. The wristplate and rocker-arm swing equal distances past the center. The cutoff on the low-pressure cylinder is controlled by hand. Why cannot equal cutoff be obtained as in Fig. 2 without the head-end valve giving trouble?



SHOWING UNEQUAL CUTOFF



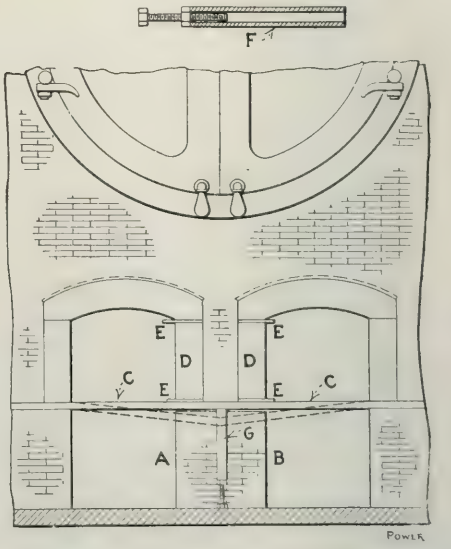
CUTOFF SOMEWHAT IMPROVED

LEROY WHITNEY.

Oakford, Penn.

Repairs to a Boiler Furnace

The piers *AB* in the illustration needed attention, but I was not willing to remove them. Examination showed that the pier *B* did not reach or support the dead plate *C*, causing the latter to bend under the weight of the upper brick work, as shown by the dotted lines, and leaving the pieces



SHOWING REPAIR TO FURNACE WITH SCREW JACK AND SHEET IRON SHIMS IN POSITION

DD loose, which at times would move out into the doorway and were likely to be broken. To remedy the trouble I cleaned all the loose material from behind the pieces *DD* and placed them in their original position, and then slipped pieces *EEEE*, of sheet iron, in the top and bottom. It was out of the question to straighten the dead plate, yet I managed to prevent it from getting worse. I then made a screw jack *F* of a 1-in. bolt and nut, and a piece of 1¼-in. pipe. This was placed at *G* on the ashpit floor under the dead plate and used to force the plate up enough to tighten the upper pier and arches.

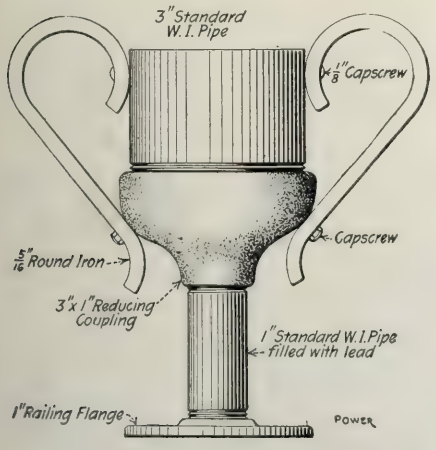
This jack and brickwork have been in place now for two years with no signs of falling down. I am surprised at the success of sheet iron as a substitute for fireclay. Many arch piers and dead plates fail on account of being inadequately supported underneath as the plate will not sustain much weight in such a hot place.

W. J. SELBY.

East Orange, N. J.

Home Made Take-down Loving Cup

This loving cup made from odds and ends found about the plant has many advantages over the common jewelry-store variety in that it is very substantial and



A UNIQUE LOVING CUP

may be easily taken apart and packed away for future use, the only tools necessary being a monkey wrench, a vise and a pair of chain tongs. This is, in my opinion, the only take-down loving cup extant.

PHILIP PARKER.

Fitchburg, Mass.

Careless Pipe Work

Fig. 1 illustrates the rear end of a tubular boiler, showing part of the blow-off pipe and connections. A reinforcing plate is riveted to the shell, to strengthen it, and a 2-in. pipe thread is cut in both plate and shell. As the combined thickness is ¾ in., and a 2-in. pipe has 11½ threads to the inch, this method gives nine threads to hold in this important but inaccessible place.

The steam fitter used a nipple that could not be screwed through the reinforcing plate; consequently it did not hold at all in the shell, as shown in the illustration. The engineer protested against this careless work, and eventually removed the nipple himself, and would not allow it to be put back. The owners then authorized the engineer to have the boiler piped according to his best judgment, which eliminated all discord between

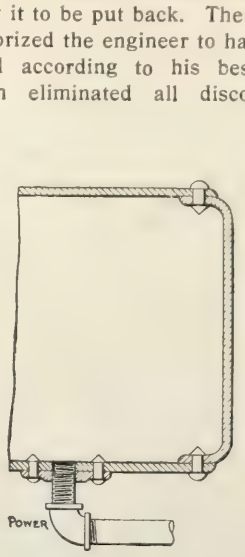


Fig. 1

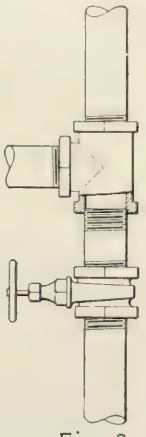


Fig. 2

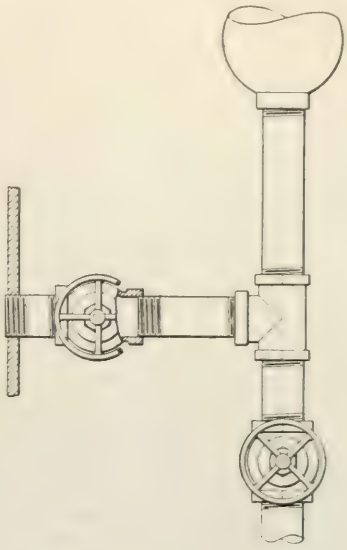


Fig. 3

SHOWING POOR PIPE CONNECTIONS

these two parties. The engineer had a nipple put in that extended completely through the shell but did not rise above it, thus making a flush joint that held with nine threads.

Fig. 2 shows part of a patented boiler setting consisting of 2-in. extra-heavy pipe and fittings. The gate valve shown is one of three blowoff valves required by this setting. After this boiler had been in service 14 years, the joint above the gate valve began to leak. The boiler was laid off and the water removed, and the nipple below the tee was unscrewed. When it had been given less than two turns it fell to the floor. This caused no surprise, because the nipple was supposed to be badly corroded. Closer examination showed that it was made of very thick iron pipe, and was practically free from corrosion. It had never been screwed in more than two threads, yet had held firmly and did not leak through this long term of service.

As the engineer that took it down had operated it ever since it was installed, his feelings can be better imagined than described, when he thought of the consequences if that nipple had blown out under full pressure while he was closing the valve. Such a blunder is usually made by attempting to make up both ends of the nipple at once, by putting a monkey wrench on the valve. It screws into the brass readily but is soon tight in the iron fitting; hence the failure to make two safe joints is not noticed. It should be made up with a pipe wrench in the fitting first, and the number of threads used observed; then the valve should be screwed on. A 2-in. pipe tap was screwed into the tee, and the old nipple put back, but now it holds by five threads.

Fig. 3 shows the lower connections to a water column. After it had been in service several years, it was necessary to take the column down and change its location. The heavy brass 1¼-in. nipple connected into the horizontal valve had only been screwed in less than two threads yet had held while the boiler was worked under full pressure, and its condition was only discovered when the nipple was unscrewed. Had this failed under working conditions, the result can be imagined.

W. H. WAKEMAN.

New Haven, Conn.

What Ails the Belt?

In our plant there is a 30-in. three-ply belt on one of the engines and it gives considerable trouble due to slippage. The belt is old and overloaded and more or less oil soaked. I put on a 10-in. single-ply belt as a rider and it is this rider that perplexes me. It runs with the slack side next to the tight side of the old belt and where the old belt curves down over the pulleys the rider runs as tight as a fiddle string. This engine runs "over" and the tight side of the rider is on top. I have two other belts carrying riders and they both run without trouble.

Will some interested reader state the cause of my difficulty?

A. SPENCER.

Kingston, N. C.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Lubricator Experience

The letters on lubricators have reminded me of an experience I once had with them. In the plant where I am employed are eight lubricators of various makes and sizes. When I took charge not one of them was in good condition, and the owner would not buy anything to fix them with nor get new ones, so I had to do the best I could with what material I could find.

Four of the feed-glass drain plugs were gone entirely so I took four ¼-in. pipe plugs and formed a disk on the end of each with solder. I put old globe-valve handles on what valves and filling plugs needed them and soldered seats on two of the filling plugs that would not hold tight. Not a glass had a rubber gasket on it; the glasses had been packed with asbestos wick. This was replaced with rubber gaskets. Two of the drain plugs were missing, so pet-cocks from an old pump were used instead.

The other drain plugs had been tightened with a wrench until the fiber disks had been worn, causing them to leak. I cleaned them out and ran in a mixture of lead and babbitt metal, 2: 1, and now the drain and filling plugs can be made tight by hand.

With some of the same mixture I made new disks on the try-cocks on the water column of the boiler and they have given excellent service. I put babbitt in the mixture as lead alone is too soft and wears out quickly, and babbitt alone is too hard and does not seat readily.

Since I fixed the lubricators and several other things the proprietor has not refused to get me what I needed to keep things in repair.

EDWARD AMSDEN.

Martins' Ferry, Ohio.

Failure of Boiler Tubes

In May 28 issue, Mr. Maxwell complains of failing tubes in water-tube boilers, especially at the baffle wall when it is 15 or 18 in. from the tubes.

When boiler tubes are subjected to great heat which rises above a certain intensity at any part, the walls of the tube cannot conduct all the heat to the water rapidly enough. Therefore, the outside of the tube will be hotter than the inside, which causes unequal expansion between the two surfaces and a grinding of the molecules on one another. This action prolonged for considerable periods destroys the cohesion of the particles of the tube and it fails. I had to replace the two bottom rows of tubes in four water-tube boilers on account of this action.

In each boiler these tubes had two grooves in each, extending from the front headers to the baffle bricks. These tubes had scale in the bottom half of them, and the grooves developed where the clean metal joined the portion covered by the sediment; in some places the grooves had nearly cut through the tubes and some of them developed pin holes and small leaks.

W. A. COOIL.

Ladysmith, B. C.

Corliss Engine Troubles

In the issue of May 28, E. C. Sutton, under the above heading, explains why he cannot give negative lap to the steam valves on a double-eccentric Corliss engine. If he will remove the valve-stem keys, and with the steam valve and dashpot rods in normal position, place the wristplate plumb, he can, by slipping the valve stem around in the valve crank, get any amount of negative lap he desires, but before cutting new keyways he should make sure that the valves fully close and cut off when the dashpots are down; if the valve stems are exact duplicates, he may change the head-end stem to the crank end, and the crank-end stem to the head end, and then the new keyways will come in a new place on the stems.

Unless the load on the engine requires a later cutoff, I would not recommend giving the valves negative lap, for the short lift of the dashpots on light loads is likely to cause them to give trouble. I have an engine on which the valves have negative lap and when running with light loads I have to throttle the engine.

C. P. LAMONS.

Lawton, Okla.

Mr. Sutton should place the steam wristplate at its center of travel, and remove the keys from the steam valve stems and also set the valves with the desired negative steam lap, allowing the steam valve stems to turn in the cranks without altering the length of the valve rods. Offset keys should be used to key the valve cranks in their new position on the valve stems and the steam eccentric moved back to coincide with the new position of the valves. This will give the desired negative lap and prevent the dashpot rods from striking the steam valve cranks.

Most double-eccentric Corliss engines are designed so that when the wristplate is at its center of travel the steam cranks are horizontal and a line drawn through the center of both steam cranks will cut the center of the valve stems and dashpot-rod pins.

FRED R. HAWK.

Kansas City, Mo.

Fallacy of Submerged Flotation

Neither the article on page 816 on the "Fallacy of Submerged Flotation" nor the editorial on the same subject, I believe, is explicit enough to satisfy anyone who is inclined to believe in submerged flotation, so I offer the following for such persons.

In the first place, it must be remembered that the pressure of any liquid is exerted evenly and equally in all directions, either within a receptacle containing the liquid, or on the surfaces of any object submerged in the liquid. This in itself is enough to overcome the idea which so many have that when an object sinking in the ocean reaches a certain depth the great pressure of water sustains it, for, the downward pressure being equal to the upward pressure, the object is in perfect balance so far as pressure is concerned.

Another thing to keep in mind is that any object placed in any liquid is buoyed up by a force equal to the weight of the volume of liquid displaced. This holds good whether the object floats or sinks. If an object weighs more than the liquid displaced it must sink, but if it displaces its weight of water without becoming submerged in will float.

Take a 1-ft. iron cube weighing, say, 480 lb., and assume it submerged to the depth in the Pacific Ocean mentioned in Mr. Nelson's letter, where a cubic foot of water will weigh about 67 lb. About 7.2 cu.ft. of water would have to be displaced to equal the weight of the block but as it only displaces one cubic foot of water it is evident that the buoyancy is not sufficient to keep the iron from sinking to the bottom.

If a cubic foot of iron could be cast as a hollow sphere having a volume of

7.2 cu.ft. and strong enough to withstand the enormous pressure at the depths mentioned by Mr. Nelson, it would sink only to the assumed depth of 5.9 miles, where the buoyancy would be equal to its weight.

A cubic foot of wood or anything weighing less than a cubic foot of water, if forced to any depth in the ocean and then released, would rise to the surface.

Some think a ship in sinking would be crushed to a shapeless mass while still others believe there would be numerous explosions, due to air being trapped in various compartments of the ship, the pressure of the air being raised as the pressure of the water increases.

No doubt the boilers or anything of like construction would collapse, due to the enormous external pressure, but in my opinion there would be no explosions after a ship got below the surface.

If a ship sinks bow foremost, air trapped in the stern might reach a pressure sufficient to cause an explosion before the stern gets below the water, but after the ship is completely submerged all trapped air in pockets having openings through which the water may exert its pressure will have its pressure counterbalanced by external water pressure preventing any collapse or explosion of the compartment.

Solid pieces of metal would show very little if any effect of the compression, owing to the pressure acting equally in all directions on the metal. It may be assumed also that wood or any porous substance would show no effects of the pressure as the water could saturate them, preserving the balance of pressure within and without.

As to the effect on a human body, such bones as the skull apparently would be crushed and all the bones would be crushed as greater depths were reached. The flesh being porous would show very little effect of the pressure after the water once got through the skin.

We may safely assume that submerged porous objects will not collapse or explode if not sinking too rapidly for the water to saturate them.

The question of "Submerged Flotation" is not one of pressures but of volumes and weights. Only a bed of dense silt or slime in the greater depths of the ocean could prevent an object sinking to a hard bottom.

L. A. TUCKER.

Philadelphia, Penn.

Since that great international misfortune, the sinking of the "Titanic," the question of whether a substance will float submerged at any depth in the sea has aroused considerable interest and may still be regarded an active controversy. Such an object as an ocean steamship, once beneath the surface, will sink until it meets something with greater sustaining powers than sea water. But, if this were true of all objects, fishes could not raise and lower themselves in the water with the ease they do. It is all a question of bulk for bulk and weight for weight.

An object much lighter than the same volume of water at its surface density will remain with a large part above the water. If its weight is precisely the same as that of the water it will displace, the top of the object will exactly coincide with the water surface. If now it be made a mere trifle heavier without changing its volume it will sink beneath the surface without going to the bottom. A log of water-soaked oak containing a few small bolts or spikers may be seen floating in still water with just a film above it. Another small nail will submerge it still farther. If it be forced down to such a level that the water pressure will materially reduce its volume the log will continue to sink, because its bulk is reduced by the pressure, due to the depth, much more than that of water. To float submerged in the ocean an object must be of the same specific gravity as the water at the given level and it must resist the compression due to that pressure.

The simplest object lesson which will demonstrate these principles is to submerge an egg in a tall glass vessel containing brine. Strengthen the brine by the addition of salt and the egg will approach the surface. Weaken it by pouring in fresh water and the egg will go to the bottom. In this way the egg can be held at any desired level within the limits of an ordinary sized vessel.

E. D. H. WILLARS.

Pittsburgh, Penn.

I have been much interested in the articles upon the "Fallacy of Submerged Flotation" appearing on pages 810 and 816. I would be glad if Mr. Nelson would answer the following questions:

What other great depths have been sounded in the oceans?

What are the temperatures of the water at the deep points?

What is the greatest depth from which fishes have been taken, and also the greatest depth from which other forms of life have been obtained?

How does the pressure of 13,752 lb. to the square inch at the greatest known sea depth compare with that used in hydraulic machinery?

H. S. ARNOLD.

Brooklyn, N. Y.

Mr. Tucker's statement that the question regarding an object sinking to the depths of the sea is one of relative volume and weight and not of pressure is true, but pressure tends to reduce the volume of a submerged object and also to saturate the object with water, thereby increasing its specific gravity and adding to its tendency to sink.

In regard to his suggestion concerning the hollow sphere, Mr. Tucker should remember that the difference in weight between a cubic foot of water at the surface of the sea and a cubic foot of water immediately below it is so small that the viscosity of the water as well as the friction would be sufficient to overcome the tendency to sink from one level to the next and that the object must be heavy enough to sink in the first place. If a cubic foot of water at the surface of the ocean weighs 64 lb., and a cubic foot of water at a depth of 31,614 ft. weighs 67 lb., the increased weight of a cubic foot of water per foot of depth would be $\frac{3}{31,614}$ lb., or, in round numbers, less than ¾ of a grain. This increase of weight is so small in proportion to bulk that its effect in buoying up an object is negligible. The editorial on page 810 well states the truth as follows: "Even considering a sinking body of closely the same specific gravity as water, it is inconceivable that its structure could resist the enormous hydrostatic pressure at great sea depths sufficient to prevent its specific gravity from increasing faster than that of water as it continues to sink."

Mr. Tucker is not correct in stating that after a ship is completely submerged all trapped air or air pockets having openings through which the water may exert its pressure will have a pressure equal to that on the outside of the air pocket, for the reason that the compressed air would be at an equal pressure throughout its entire bulk, whereas the pressure on the outside of the vessel would be increased about ½ lb. per square inch for each foot of depth, so that an air pocket of, say, 50 ft. in the stern of a vessel would have an interior pressure 25 lb. to the square inch greater at the top than the surrounding water. This pressure might be sufficient to throw off a deck, but such a happening would not be expected to occur, owing to the numerous deck openings, which would prevent the formation of such an air pocket, as the air would always have a tendency to force its way through any opening, no matter what pressure of water surrounds it. A vessel would at first have a tendency to sink slowly, but it would gain in velocity, and after a depth of 1500 or 2000 ft. had been reached the pressure would have increased to such an extent that boilers would collapse or their tubes rupture and the boilers fill; piping would be flattened or broken; and coal and cargo saturated. The speed of the descent would be so great that human bodies, instead of permitting saturation, due to porosity, would undoubtedly be compressed in bulk to that of a man's arm. Partially filled sealed bottles, casks, etc., would be shattered. Water at a pressure of 6000 lb., which pressure would be obtained in 2000 fathoms, would have a velocity of 1000 ft. per second; so that the smallest in-

terstices which might form pockets would be quickly penetrated and completely filled.

Mr. Willars refers to the experiment fo an egg in a glass vessel containing brine. This experiment is really useful in showing that the heaviest brine remains at the bottom and the lightest at the top. If the brine were of uniform density the egg would either float at the surface or sink to the bottom, and the fresh water referred to, which being added would cause the egg to go to the bottom, must be poured in through a tube reaching to the bottom, to produce that effect.

In reply to the questions in Mr. Arnold's letter the depth of 5269 fathoms, to which I referred, was sounded by the U. S. S. "Nero" near the island of Guam in the Western Pacific Ocean. This depth exceeds the height of the highest mountain. There are now known to be 43 areas with a depth exceeding 3000 fathoms. Depths exceeding 4000 fathoms have been found in eight different places, and two soundings have been made in depths exceeding 5000 fathoms.

The temperatures of the deeper parts of the sea are always close to the freezing point. Records of 34 deg. F. are not uncommon. Animal life which exists in the great depths is subject to enormous pressure, total darkness and intense cold.

The U. S. S. "Albatross" in the North Atlantic Ocean dredged fishes from a depth of 2949 fathoms, and it also dredged other forms of life in the South Pacific Ocean from a depth of 4173 fathoms. The same vessel also obtained 800 fishes at a single haul of the dredge from a depth of two miles in Behring Sea, which shows the abundance of animal life in great depths.

Pressures of 15,000 to 17,500 lb. per square inch are not uncommon in practical use, and special high-pressure hydrostatic apparatus have used pressures as high as 45,000 lb. per square inch.

JAMES W. NELSON.

Brooklyn, N. Y.

Carbonine

On page 266 of POWER for Feb. 20, there is an editorial headed "Get Rich Quick Schemes," the latter part of which tells about "Carbonine" and how the use of that same powder mixed with water and sprinkled on the coal is supposed to produce a saving of 25 or 26 per cent. in the coal bill. Salem is not the only city in Massachusetts where the promoters of Carbonine have caught some victims, as Fall River has two or three plants which its promoters claim are using Carbonine and for which they claim a saving of 26 per cent. in the coal pile. There are other plants which cancelled their first order for Carbonine, as I have been told, when the preposterousness of the claims of its promoters was explained to them.

It seems that the right way to use Carbonine is to buy one ton, which costs $80. Then take one pound of Carbonine powder and mix it with one gallon of water (hot or cold) it makes no difference. Then to each ton of coal sprinkle eight gallons of this mixture and then you save 26 per cent. of your coal bill. One of the promoters was asked how it was figured out that adding 67 lb. of water to a ton of coal aided the combustion of the coal, and he answered that he did not know. "You could use Carbonine without water," he said, but adding water made the mixture go farther. You could get the same results by using it dry. The writer agrees with him, and also believes he can get the same results by leaving out the Carbonine.

It seems to me that the only way to prove out this Carbonine matter, as the editorial claims, is to get some reliable mechanical engineer to make a test of the evaporative qualities of coal with and without the use of Carbonine. Then if they can add 20 or 25 per cent. to the evaporative qualities of the coal, we will believe in Carbonine, but not until then. If any man can get up a mixture that will add 20 or 26 per cent. of value to coal, then it will be, as my friend Snyder says: "We can do away with 26 per cent. of our boilers, also 26 per cent. of the boiler-making works and the Coal Barons would kidnap the discoverer."

MICHAEL H. HARRINGTON.

Fall River, Mass.

[The sample accompanying the letter below was sent without comment to Dr. C. E. Lucke, of Columbia University, for analysis, and by him submitted to a firm of industrial chemists whose report is reproduced herewith.—EDITOR.]

M. C. WHITAKER, M Sc — F. J. METZGER, PH D.

CONSULTING INDUSTRIAL AND ENGINEERING CHEMISTS

BROADWAY AND 116TH STREET
NEW YORK CITY

June 11, 1912.

Prof. C. E. Lucke,
Columbia University.

Dear Dr. Lucke:

We have made a qualitative analysis of the sample of material forwarded on June 5th with your letter from Power. We find that this material is practically all common salt mixed with some potassium permanganate and some zinc compound. We also note small quantities of carbonaceous material and a little sulphate.

The stuff looks to us very much like some of the fake ash burning compounds which are put out and which we have had occasion to examine before.

Hoping that you will find this satisfactory,

Very truly yours,

M. C. Whitaker

Water Broke the Piston

The pistons mentioned by A. J. Haight in the May 7 issue were undoubtedly broken by a charge of water from the steam or exhaust line. One cannot say which as the piping arrangement is unknown.

Engines that exhaust into a common exhaust header are subject to serious accidents from the presence of considerable water in the header and piping. I have seen high-speed engines wrecked the same as that mentioned by Mr. Haight and have traced the cause to faultily erected headers and piping.

C. R. MCGAHEY.

Baltimore, Md.

Wavy Expansion Line

In the May 28 issue, Leighton Johnson, in commenting on the cause of the wavy expansion line of the diagrams mentioned in the Apr. 16 issue, refers to a 1½-in. pipe forming the connection to the indicator. This is undoubtedly a draftsman error as this and the other connections are of ½-in. pipe and fittings.

J. W. DICKSON.

Memphis, Tenn.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Capacity of Stranded Cables

How is the safe current carrying capacity of a stranded copper conductor computed? From the wiring tables, a No. 12 B. & S. rubber-covered copper wire will safely carry 17 amperes. Would a cable made of 19 strands of No. 12 wire carry

$$19 \times 17 = 323 \text{ amperes}$$

or would it be determined by multiplying 6530 circ. mils (the area of a No. 12 wire) by 19, which would equal 124,070 circ. mils. The tables show a rubber-covered cable having a cross-section of 124,074 circ. mils to be capable of safely carrying only 141 amperes. Why this difference?

J. B.

The reason for 19 independent No. 12 rubber-covered copper wires being able to carry 323 amperes, whereas a stranded cable composed of the same number of wires of the same size would safely carry only 141 amperes, is that the independent wires present a greater radiating surface than the cable; hence the temperature rise is less. The combined circumferences of the independent wires is about 4.8 in. as against 1.1 in., the circumference of the cable.

The formula for figuring the current carrying capacity of single-conductor rubber-covered, stranded cable is

$$I = 490\sqrt{\frac{TD^3}{r}}$$

where

D = Diameter of cable in inches;
T = Temperature rise in cable core in degrees Centigrade;
I = Current in amperes;
r = Specific resistance of wire in ohms per inch-foot at the final temperature.

Value of Flue Gas Analysis

Is flue-gas analysis of as much importance when firing with natural gas as when using coal as fuel? Would employment of a CO_2 recorder be of benefit using natural gas for fuel with a steady load?

D. C. S.

Flue-gas analysis is as valuable for determining the completeness of combustion when burning gas as when coal is used as fuel. Steady load does not insure uniform conditions for combustion because gas and air pressure and atmospheric conditions are likely to vary, cracks may develop in furnace, or passages be left open which allow air to get inside of the setting. A CO_2 recorder affords means for determining whether operating conditions are being maintained.

Butt and Double Strap Joints

In double-riveted butt and strap joints of boilers, why is the narrower strap placed on the outside of the shell?

H. C.

Because the narrower strap is the one which has proper pitch of rivets for calking the joint.

Bleeder Turbine

What is a bleeder turbine?

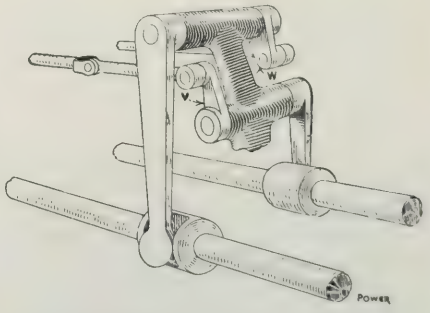
G. C. W.

A steam turbine in which provision is made for bleeding steam from one of the intermediate stages, for use in heating systems or manufacturing processes.

(See POWER, Aug. 1, 1911, page 193.)

Valve Motion of Duplex Pump

In the valve motion of a duplex pump, why are the valve-rod rocker-arms, *V* and *W*, placed one above and the other below



VALVE MOTION OF DUPLEX PUMP

its rocker shaft? Could the pump be made to work with each valve-rod arm on the same side of its shaft?

N. H. J.

In ordinary duplex pumps, the steam valve of one side is a duplicate of the other, and the valve-rod arms are placed on opposite sides in order to impart opposite direction of motion. Before reaching the end of its stroke one side of the pump has to move the valve of the other side, so the latter will move in opposite direction and the reversing action and operation of the pump are thus made continuous.

If the arms were the same, both sides of the pump would be as one side without means of reversing itself.

Chamfering of Tube Holes

Why should the edges of tube holes in a boiler-tube sheet be chamfered?

H. C.

The tube holes should be chamfered or rounded to a radius of about $\frac{1}{32}$ in. so that in expanding the tubes they will not be sheared by the edges of the tube holes.

Steam Used by An Ejector

What is the best method of determining the number of pounds of steam used per hour by an ejector?

C. A.

By taking the difference between the amount of water raised and the total discharged.

Testing Capacity of Safety Valve

What practical test can be made to determine whether the safety valve on a boiler is large enough?

E. M. S.

Making proper allowance for correctness of the steam gage, at a time when the gage indicates that the safe limit of pressure has been reached, the safety valve should be set so as to lift at that pressure. If the valve is large enough, then at any time after the proper limit of pressure has been reached, it should liberate steam so fast that the pressure cannot be raised above that point with the hottest fire that can be made and when all other escapes for steam are closed.

A test of the capacity of the valve should not be made unless the rate of firing can be increased very gradually and it is well established that the intensity of the fire is under complete control and provision is made for liberating the steam by an auxiliary outlet that can be opened for escape of steam in case the safety valve fails to work properly. For a boiler burning coal with natural draft the auxiliary outlet should have an area of not less than 1 sq.in. for each 2 sq.ft. of boiler grate surface.

Study Questions

This Week's Questions
Last Week's Answers

(41) While water is being drawn from a house tank at a uniform rate a pump running at 60 strokes per minute can fill the tank in 8 hr. and at 72 strokes per minute can fill it in 4 hr. With pumping stopped, how long would it take the full tank to empty?

(42) Under the same conditions as above, how long will it take to fill the tank again with the pump running at 84 strokes per minute?

(43) Two wires which have respective resistances of 5 and 7 ohms are joined in parallel so that a total current of 38 amperes flows through them. What current flows through each wire?

(44) What will be the rise of temperature of 1 lb. of water at 55 deg. F. when an ounce of copper at 212 deg. F. is immersed in it?

(45) Under a pull of 50 lb., how much will a steel tape stretch that is 100 ft. long, ½ in. wide and 0.02 in. thick if the modulus of elasticity of steel is 30,000,000?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(36) It is first necessary to find the tonnage capacity of the compressor, which is equal to the number of cubic feet of gas actually displaced per minute multiplied by the number of British thermal units of cooling effect available per cubic foot, divided by 200, the B.t.u. per minute equivalent to one ton of refrigeration per 24 hr. The actual displacement is equal to the volume swept through per minute by the piston, multiplied by the volumetric efficiency, 65 per cent. Therefore, the actual displacement would be

$$\frac{0.7854 \times 9^2 \times 13}{1728} \times 180 \times 0.65 = 56 \textit{ cu.ft.}$$

per minute. With a condenser pressure of 180 lb. and a suction pressure of 15 lb. the cooling effect per cubic foot of displacement will amount to 50 B.t.u. Then

$$\frac{56 \times 50}{200} = 14 \textit{ tons}$$

which is the tonnage capacity of the compressor. To find the rating of the engine necessary to drive the compressor, neglecting friction, multiply the tons capacity by 1.5,

$$14 \times 1.5 = 21 \textit{ hp.}$$

To include friction multiply by 1.7

$$14 \times 1.7 = 23.8 \textit{ hp.}$$

To coincide with commercial ratings it would then require a 25-hp. engine.

(37) With each forward and return stroke, taking 1 sec., the pump leaves 1 gal. in the standpipe. In 19 sec. the pipe has been filled to within 3 gal. of the top and on the next forward stroke water reaches the top, taking altogether then 19⅜ sec.

(38) Since the drop in voltage through transmission is 5 per cent. of the initial voltage and that at the building is 550 volts, the voltage at the power house must be

$$\frac{550}{0.95} = 578.95 \textit{ volts}$$

Hence the actual drop is 28.95 volts.

A load of two thousand 60-watt lamps and 50 hp. in motors is equivalent to

$$2000 \times 60 = 120{,}000 \textit{ watts (lamps)}$$
$$50 \times 746 = 37{,}300 \textit{ watts (motors)}$$

Total 157,300 *watts.*

Since the voltage is 550 the current will be

$$157{,}300 \div 550 = 286 \textit{ amp.}$$

From Ohm's law $I = \frac{E}{R}$ where I is the current, E the voltage drop and R the resistance, substituting,

$$286 = \frac{28.95}{R}$$

whence

$$R = \frac{28.95}{286} = 0.1012 \textit{ ohm}$$

is the resistance per mile of the line. The resistance per mile of wire would be $0.1012 \div 2 = 0.0506$ ohm.

(39) The fireman was trying to say that the injector had been fixed six days and was out of order 16 days. The calculation is a simple matter of addition and subtraction, using the plus sign for "after" and minus sign for "before" and adding a day for "tomorrow" and subtracting one for "yesterday," thus

$$+ 1 + 1 - 7 - 1 = - 6$$

so it had been fixed six days *before*

$$+ 1 - 35 + 17 - 1 + 2 = - 16$$

and it was out of order for 16 days *before* that. The calculation can be easily checked with the aid of a calendar.

(40) Any unit length of the pipe may be considered. Take it as 1 in. The pressure tending to burst such a ring 24 in. in diameter, separating it on any diameter, is ½ of $24 \times 1 \times 80 = 960$ lb. (See answer to study question No. 20, June 11, page 852.) This is resisted by a section of ⅜ × 1 in. = ⅜ sq.in. Then

$$\frac{960}{\frac{3}{8}} = 2560 \textit{ lb. per square inch}$$

While Colorado's coal production has decreased, its production in 1911 was over 10,000,000 tons.

One of the "Imperator's" immense rotors contains 50,000 blades, weighs 135 tons and can develop 22,000 hp.

Convention of National District Heating Association

On June 25 to 27 the National District Heating Association held its fourth annual convention at Detroit, Mich., with headquarters at the Cadillac Hotel. In every way, the meeting was successful, the attendance was 168, a number of good papers were presented and the entertainment provided was enjoyed by all.

At 10 a.m. on Tuesday the first session was opened with an address by Alexander Dow, of the Edison Illuminating Co., of Detroit. G. W. Wright, of Baltimore, responded. President Spencer's address will appear in an early issue.

Papers

Chairman A. P. Biggs read the report of the meter committee, which briefly described meters of interest to central heating companies. It included a discussion on system troubles which prevent correct metering and instructions on inspection, installing, reading and removal.

E. Darrow, in his paper on "Common Sources of Trouble in Customers' Installations," laid especial stress on lack of radiation and insufficient pipe capacity for zero-weather conditions. The determination of these factors is no longer left to architects and piping contractors. All plans are now submitted to the company's engineering department to be checked up before a heating contract is accepted. Too little attention is paid to radiator air valves. They must be sensitive to respond quickly to the cooling that takes place with an air accumulation. Every customer's installation must have at least one thermo-control valve set for 70 deg., usually placed near the entry of the service. For street mains, Mr. Darrow states, the three important requisites for construction are drainage of water accumulations from possible contact with the steam lines, provision for expansion and insulation.

In his paper on "Quality of Steam as Served by Central Stations to Their Customers," A. C. Shepherd gave his attention to tests of steam at different points along the lines of the Cleveland Electric Illuminating Co., the object of the tests being to determine the variation of quality at different points in the distributing system. Results and methods were given, using barrel and jet continuous condensing calorimeters.

R. D. DeWolf and J. L. Hecht, forming the committee on rates, presented a long report. Two sets of questions, one adapted to steam-heating systems and the other to hot-water systems, had been sent to 284 heating companies in the United States and Canada. Replies from 107 steam-heating and from 40 hot-water heating companies were received and tabulated in the report, the results indicating a wide range both in rates and methods followed in giving service. Contracts from

the various companies had also been requested and the contract clauses used by the various companies were summarized.

In "Analysis of Heating Rates," by A. E. Duram, the author did not attempt to determine any specific rate of charge, but rather suggested a method for such determination based upon the actual cost of the service rendered plus a fair percentage of profit. The great diversity in rates was shown from tabular data from various cities, and for rapid comparison a chart with a number of curves had been incorporated in the paper to show the net rate per square foot per season for all rates of consumption from 100 to 1200 lb. per square foot.

It is the duty of the committee on central-station records to collect data from all heating company members for the purpose of establishing standard forms of station and distribution records. This year and last it did not receive enough data to make complete records and it was decided to embody in the report a description of the system used by the Peoria Gas & Electric Co. This system is unusually complete and will undoubtedly furnish suggestions to other companies on which to start or improve their systems.

"Heating in Connection with Steam Turbines," by August H. Kruesi, presented some developments of the Curtis turbine bearing on the extraction of steam for district steam and for industrial heating, such as that required in bleaching, washing and dyeing. Features of operation in connection with such work were also discussed.

Edwin D. Dreyfus read a paper covering much the same subject, the turbine, however, being of the Parsons type. Both papers and most of the others herewith will be published in later issues.

J. F. Lewis described the power plant of the Murphy Power Co., Detroit, which was afterward inspected by many of the delegates. It carries a lighting and power, refrigerating and heating load which makes an excellent combination to maintain a uniform load factor.

A hot-water plant at Oak Park, Ill., was described by G. E. Chapman. This plant has 4800 hp. in boilers, 2000 hp. in turbines, 500 hp. in engines and approximately 12 miles of two-pipe heating mains.

"Sectional Steam Heating in Chicago" was by S. Morgan Bushnell. To supply steam for heating in the Loop district of Chicago, the Illinois Maintenance Co. was organized in 1900. Contrary to the usual practice, this company, with one or two exceptions, has installed no boilers of its own. It simply contracts to operate boiler plants already installed in buildings to furnish steam for heating and other purposes. Each plant, therefore, covers a very limited territory, one set of boilers being used in some cases for one building, although it is the intention to serve several adjacent buildings from each plant whenever contracts permit.

Each plant has its own operating force and all are supervised by a general engineer. The heavy investment for street mains is avoided, the transmission losses reduced, the operating efficiency increased over private plants and supplies in large quantities are obtained for less money. The company is giving satisfaction. Including steam for elevators, ice machinery, pumps, hot-water heaters, etc., it is supplying an equivalent of 800,000 sq.ft. of radiation.

In "Depreciation in Underground Distribution Systems," William Jennings, Harrisburg, Penn., gave a short account of his own experience in the Harrisburg Steam, Heat & Power Co. plant, originally installed in 1887. In making changes as late as 1911 it was found that there had been very little deterioration in the original system, which verified the author's claim that a properly installed system should last for 25 years.

A symposium on underground insulation by representatives of the various companies engaged in that line completed the list of papers.

Election of Officers

The officers for the ensuing year are president, R. D. DeWolf; first vice-president, J. F. Lewis; second vice-president, J. L. Hecht; third vice-president, E. Darrow; secretary-treasurer, D. L. Gaskill; executive committee, W. A. Wolls, H. R. Wetherell and A. D. Spencer, the retiring president.

Indianapolis as the next convention city was favored by the convention, but as requests had been received from numerous other cities, the final choice was left to the executive committee.

Entertainment

There was a theater party on Tuesday evening. On Wednesday the steamer "Pleasure" took the entire delegation, including the ladies, for a day's sail to the Flats and up the St. Clair River, ending at Belle Isle in the evening. Brief sessions were held on board to give time for a musical program and dancing. There was a buffet luncheon on the steamer and a crab and chicken dinner at the Casino, and in the evening a band concert on the island terminated a most enjoyable program. On Thursday at the close of the last session, touring cars took the delegates to "Delray."

Exhibits

These manufacturers exhibited heating specialties of all kinds, pipe insulation and meters: American District Steam Co., North Tonawanda, N. Y.; Builders Iron Foundry, Providence, R. I.; Central Station Steam Co., Detroit; Chamberlin Metal Weather Strip Co., Detroit; Jenkins Bros., New York; H. W. Johns-Manville, Milwaukee; Michigan Pipe Co., Bay City; the Richards-Wilson Pipe Covering Co., Grand Rapids; A. Wyckoff & Son, Elmira, N. Y.

Missouri N.A.S.E. Convention

On June 20 to 22, the Missouri State Association of the National Association of Stationary Engineers held its second annual convention at Joplin, and was well attended.

Allen McReynolds, of Zinc Belt Lodge No. 3, Joplin, was chairman at the opening session. After the Rev. Charles A. Weed's invocation, Mayor Jesse F. Osborne delivered the address of welcome. The response was made by National Secretary Fred Raven, as John W. Lane, editor of the *National Engineer*, was unavoidably absent. J. J. White, state president, spoke briefly of the work of the association during the year.

In the evening, at the exhibit hall, on the mezzanine floor of the Hotel Connor, F. E. Ransley, president of the Central States Exhibitors' Association, welcomed the visitors. An informal reception and luncheon were given by the exhibitors at which there were speaking and music.

At the Friday morning session Fred Raven, in one of his characteristic "keynote" addresses, spoke on the purposes of state conventions. State President White, in his report, told of his success in increasing the membership throughout the state. The secretary's report showed that there are six subordinate associations in good standing with a total membership of over 500 and that the prospects are bright for forming several more associations in the immediate future.

The committee on revising the constitution recommended several important changes. The *per capita* tax was raised from 10 to 15c. and provision was made to pay the delegates mileage with the funds left after all convention expenses have been met.

In the afternoon the ladies were taken for an automobile ride about the city. Four interesting lectures were given during the open session. George K. Elliott, chief chemist and metallurgist for the Lunkenheimer Co., spoke on "The Metallurgy of the Modern Steam Valve." He showed that the commercially successful valve must resist steam and strains, and the metal must be of such a nature as to cast easily and machine well. Pine copper is unsatisfactory as it is too soft to resist wire drawing, etc.; hence tin is added to produce the proper degree of hardness and strength. Zinc is used to properly finish the metal and make it steam-tight. Lead is afterward added to make the metal easy to machine.

The effects of sulphur, phosphorus and graphite on the qualities of iron were also discussed. Mr. Elliott illustrated his lecture with diagrams, tables and valves, showing the analysis and characteristics of various valve metals.

L. W. Pearce then read his paper, "An Engineer." This engineer is of the old school; he is little more than a starter and stopper; a handy man, with but little mental equipment to piece out his manual skill. He is rapidly passing, said Mr. Pearce, and but few of his type exist today. The progress made in steam engineering demands that a man be familiar with the theoretical part of the science of power generation and transmission. In conclusion, he said that manliness was the keystone of success in steam engineering and, coupled with good habits, this quality would give anyone a long start.

Herbert Taylor, of the Du Pont Powder Co., spoke briefly on "New Application of Concentrated Energy, 'Dynamite.'" Crop yields have been increased over 300 per cent. by thoroughly dynamiting a field to a depth of from 3 to 5 ft. By thus loosening the soil, more moisture is absorbed and retained and the plants are better nourished.

A. Herron, of St. Louis, read F. H. Munsberg's paper, "The Educational Feature of the N. A. S. E. and the Benefits Derived from Attending Meetings," as Mr. Munsberg was unable to be present.

In the evening the delegates and visitors were entertained at Lakeside Park, an amusement resort about 10 miles from Joplin, where the Essany Film Manufacturing Co., Chicago, gave a special moving-picture show.

At the executive meeting on Saturday morning, Springfield was selected as the next meeting-place. These newly elected officers were installed by National Secretary Raven: President, F. H. Munsberg, St. Louis; vice-president, D. M. Trimble, Kansas City; secretary, J. H. Chestnutt, Springfield, reëlected; treasurer, A. P. Maeder, Joplin; conductor, O. M. Evans, Springfield; doorkeeper, W. E. Rogers, St. Louis, and trustees, D. Spoerndli, St. Joseph; F. B. Daggett, St. Joseph, and Charles Parkinson, St. Louis. A joint meeting of the delegates and exhibitors was then held, at which members of both associations spoke.

On Saturday afternoon a trolley trip was taken to the American Zinc, Lead & Smelting Co.'s mining plant, where the power plant was inspected and the process of recovering the lead and zinc ores was observed. The power plant and employees' club house of the Southwestern Missouri Railway Co. were then visited, after which the party returned to Joplin.

The evening was spent in visiting the numerous and interesting exhibits. The exhibitors, members of the Central States Exhibitors' Association, were: American Steam Gauge & Valve Co.; V. D. Anderson Co.; Crandall Packing Co.; Dearborn Drug & Chemical Works; Garlock Packing Co.; Greene, Tweed & Co.; Hawk-Eye Compound Co.; Hills, McCanna Co.; Home Rubber Co.; Jenkins Brothers; H. W. Johns-Manville Co.; Keystone Lubricating Co.; *National Engineer;* Osborne Valve & Joint Co.; William Powell & Co.; POWER; Schaeffer & Budenberg Co.; C. E. Squires Co.; United States Graphite Co.; Joplin Supply Co.; Rogers Foundry & Manufacturing Co.; the Missouri Fidelity & Casualty Co.; Landreth Machinery Co.; Waters-Pierce Co.; DuPont Powder Co.; Nowlin Machinery Co.; George B. Carpenter & Co.

Connecticut N. A. S. E. Convention

On June 28 and 29, the seventeeth annual convention of the Connecticut State Association of the N. A. S. E. was held at Bridgeport, the headquarters being the Stratfield Hotel. About 50 delegates were present from the 10 local branches, besides many prominent visitors from New York, New Jersey and Massachusetts. The meetings were held in Colonial Hall, which was tastefully arranged for the mechanical exhibit, fully 50 firms occupying booths.

Friday morning at 10:30, the convention was formally opened by State President F. Chapman. Mayor C. B. Wilson welcomed the engineers and their guests to the city, National President Edward Kearney responding. F. J. Mulcahy, vice-president and general manager of the Crane Valve Co., Bridgeport, addressed the convention at length. The several committees were then appointed, and the convention adjourned until Saturday at 9 a.m.

On Saturday morning, business was quickly disposed of. The reports showed a big increase in membership and that the organization was in a good financial condition. The officers for the ensuing year are Thomas J. Phillips, president; Bernard Reynolds, vice-president; Charles M. Dowd, secretary-treasurer; Herman Offinger, conductor; Jacob Miller, doorkeeper; William Holt, George P. Thomas and Roscoe Granger, trustees. F. D. Hastings was reëlected state deputy.

A meeting of the supplymen was held on Friday afternoon, President F. S. Bulkley in the chair, and a new organization, the Connecticut Supplymen's Association, was formed, for which a charter will be asked. Its aim is to assist the local committees in arranging for exhibits and in entertaining the delegates and guests. The following officers were elected: F. P. Upson, president; E. S. Birge, vice-president; W. L. Larash, secretary-treasurer; John H. Foote, F. S. Bulkley, R. H. Stiles, J. B. Harrington and C. H. Tower, executive committee.

A visit was made to the plant of the Crane Valve Co. on Friday afternoon and during the evening, in the exhibition hall, Past National President Herbert E. Stone talked on "The Relationship of the Engineer to the Supplyman." C. Cooley, superintendent of power of the New York, New Haven & Hartford R.R., gave an address on "Recent Installations of Power Plant Equipment." H. J. Maynard, of the General Electric Co., read a paper on "Artificial Illumination." Interspersed between the speeches were songs by Frank Corbett, of the Consolidated Valve Co., and William Murray, of Jenkins Brothers. Frank J. Quinlan, chairman of the local committed, presided.

On Friday night the delegates were entertained by the supplymen with a smoker in the rathskeller of the Stratfield Hotel. On Saturday afternoon the company enjoyed a pleasant sail on Long Island Sound. It was decided to hold the next annual convention at Meriden.

EXHIBITORS

The exhibitors were the American Steam & Valve Manufacturing Co., Ashton Valve Co., Armstrong Manufacturing Co., Albany Lubricating Co., Bridgeport Testing & Laboratory Co., Crandall Packing Co., Connecticut Metal Boiler Compound Co., Crocker-Wheeler Co., C. A. Claflin Co., Crane Valve Co., Durabla Manufacturing Co., Dearborn Drug & Chemical Works, M. T. Davidson Co., General Electric Co., Garlock Packing Co., Greene, Tweed & Co., Harrison Safety Boiler Co., A. W. Harris Oil Co., Handy Manufacturing Co., Hartford Mill Supply Co., T. Hawley & Co., Home Rubber Co., Hawley Hardware Co., W. J. Hyland Manufacturing Co., Heine Safety Water Tube Boiler Co., The Ironworks Co., Jenkins Brothers, H. W. Johns-Manville Co., Keystone Grease Co., Lunkenheimer Co., Lyon & Grumman, E. H. Ludeman, Estate Edward R. Ladew, Mason Regulator Co., McLeod & Henry Co., C. S. Mersick, N. Y. Belting & Packing Co., Nightingale & Childs, *National Engineer, New England Engineer,* National Flue Cleaner Co., Otis Elevator Co., Perfection Grate Co., Patterson Allen Co., POWER, Peerless Rubber Manufacturing Co., Quaker City Rubber Co., Roto Co., J. R. Robinson, Smooth-On Manufacturing Co., *Southern Engineer,* Standard Regulator Co., Schaeffer & Budenberg Manufacturing Co., Skinner Engine Co., Traveler's Insurance Co., George H. Thacher Co., and W. R. Winn.

PERSONAL

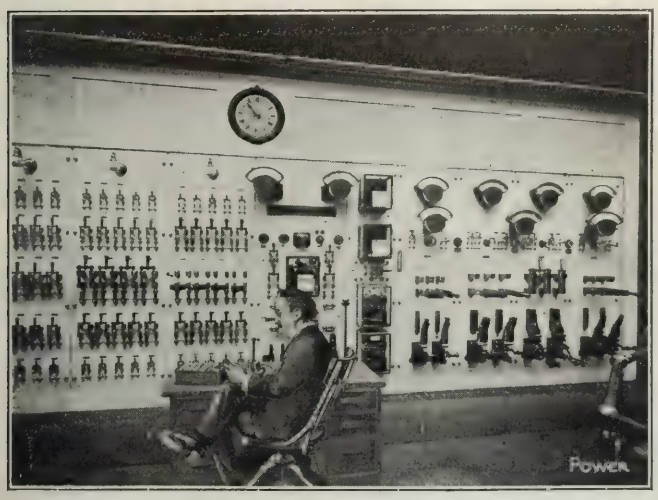
Dr. Leonard A. Levy, head of the chemistry and research department of Alexander Wright & Co., Ltd., has just been awarded the degree of Doctor of Science by the University of London.

POWER

Vol. 36 NEW YORK, JULY 16, 1912 No. 3

The Electrician and the Engineer

The electrician has it all over the engineer. He is given a handsome marble switchboard with ammeters and voltmeters and wattmeters and all sorts of instruments, fixed and



portable, for measuring the current which he handles. The engineer is given a steam gage and a water column. The electrician, with all of the instruments, cannot make any more current than the engineer gives him, nor make it go any farther.

The engineer, who is responsible for the input and output of the plant, often has not so much as a pair of scales to tell if he is getting as much coal as he is paying for; no calorimeter to test the quality of the coal; no water meter to show how much water he is evaporating; no pyrometer to test the temperature of his furnace or of the escaping gases; no draft gage; rarely, unless the maker of the heater gives it to him, a thermometer to tell how hot the feed water is; no Orsat or other apparatus to tell how complete the combustion of the fuel has been and how much unnecessary air he is heating up to be sent out of the chimney; no indicator to tell how the valves of his engine are set or how much power the engine is developing; no engineer's level to tell if the engine and shafting are in line. If he asks for these things he is likely to be told that he is getting fussy, scientific and unpractical and to be sent back discredited to his steam gage and water column.

An engineer can hardly be expected to do the best that can be done when he has no means of knowing what he is doing; to proceed intelligently to better conditions when he has no means of determining what the existing conditions are.

It is no justification for an employer to say that his engineer would not know what to do with such instruments and apparatus if he had them. Try him and more than likely you will find that he does know these instruments and how to make good use of them, and that he can save you many times their cost by using them. If he cannot or will not, there are those who can and will.

New San Joaquin Power Plant

By Howard H. Bliss

A 16,000-kw. hydroelectric plant operating under a static head of 1405 ft.

Three-phase current is generated at 2300 volts and a large part of it is taken by motors.

To supply the rapidly increasing demand for current in the great central valley of California, the San Joaquin Light & Power Corporation has recently installed a new hydraulic plant and made extensive improvements in its reservoir and ditch system. The new plant replaces the first generating station of the company, installed in 1896, which was once famous for using the highest head of water in existence. Modern practice and high efficiency are well exemplified in the new installation.

The power house is on the main San Joaquin River, about 30 miles from Fresno. It is of steel with double walls of cement plaster and reinforced-concrete floors and roof. The floor dimensions are 148 ft. by 71 ft. 9 in. Current is generated by four 4000-kv.-a. generators, three-phase, 60-cycle, 2300 volts. These are driven by Doble impulse wheels, one to each unit, running at 400 r.p.m. Water under a static head of 1405 ft. is delivered to the wheels through fixed needle nozzles, having auxiliary nozzles which waste the water into the tailrace when the governors close the needles at higher than a predetermined speed. The relief device has a slow-closing mechanism which shuts off the waste water at a safe rate. Each pair of units is supplied from one pipe line, 34 in. in diameter at the bottom, ¾ in. thick, lap welded; and 44 in. at the top, ¼ in. thick, lap-riveted. The horizontal length of the penstock is 3900 ft.

At the head of the pipe line is a forebay regulating reservoir eight acres in area, to which the water is brought through a seven-mile conduit. Concrete-lined tunnels make up two miles of this, a steel flume 966 ft., a concrete flume 715 ft., and the rest is an earth ditch in disintegrated granite. Owing to the serious annoyance from gophers and ground squirrels which had caused many breaks in the ditch while the old power house was in commission, a concrete core-wall was built into the downstream bank when the ditch was enlarged to the present dimensions. The foundation of the core-wall was placed 2 ft. or more below the bottom of the ditch, and the wall carried up 6 in. thick to a point 1 ft. above the flow line for 100 sec.-ft. Two small ponds near the intake have been utilized as sand traps.

FIG. 1. INTERIOR OF SAN JOAQUIN POWER HOUSE

At the head of the ditch is a small diverting dam, placed at the junction of the two northern branches of the San Joaquin River. Several miles up the northern or larger fork is a reservoir capable of storing 51,000 acre-feet. The

Crane Valley which it occupies has been used for water storage since the company began operations, and the old dam is inclosed within the new and higher one necessitated by the new installation. The new dam is 2000 ft. long and 150 ft. high, with a concrete core-wall reinforced with square steel bars. On the upstream side of the dam an earth-fill of disintegrated granite and red adobe clay was deposited by hydraulic means and riprapped 2 ft. thick. On the downstream side rock fill is used for about half the length of the dam in the deepest part of the cañon and earth-fill for the remainder. A concrete tower in the reservoir near the dam is used for controlling the outlet water, which is sent through a tunnel beneath the dam. At the end of the tunnel the water can be sent down the river channel several miles to the forking, where the diverting dam starts it toward the new power house, or it can be turned into the ditch line leading to the company's power station No. 3, four miles away.

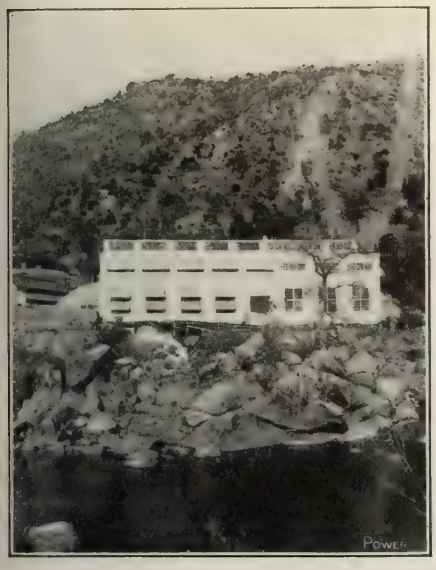


FIG. 2. EXTERIOR VIEW OF STATION

This plant contains two 1000-kw. units driven by impulse wheels working under 400-ft. head. The tailrace empties into the channel of the north fork and the water used in No. 3 station is used again in the new power house. At present, No. 3 plant is not operated but is held as reserve. An additional fall of 350 ft. between this place and the diverting dam can be utilized later if necessary by constructing another station.

In the new power house are four banks of 1500-kw. water-cooled transformers to step up the voltage from 2300 to 60,000. At present the voltage is raised only to 30,000 at this plant. Two lines carry the power 18 miles to the Coppermine switching station in the San Joaquin Valley, where a part of it is stepped up to 60,000 for transmission 195 miles to Bakersfield. From the Coppermine station two 30,000-volt lines run to Fresno, 17 miles, and one to Madera, 28 miles. The latter is continued 35 miles beyond to Merced, and 22 miles beyond that to Merced Falls. There the voltage is reduced to 10,000 for transmission to a mining district in Mariposa County. The company's service reaches seven counties: Mariposa, Merced, Madera, Fresno, Tulare, Kings and Kern. There are 676 miles of transmission line and 563 miles of distributing line.

A large part of the power is taken by motors running irrigating pumps, oil-well and mining machinery, street cars and machine shops. Hence the load factor is exceptionally favorable, though the power factor is generally low. Table 1 shows the operating data for 11 months in 1911, where the load factor is computed by dividing the kilowatt-hours for the month by the product of the number of hours in the month by the monthly peak.

FIG. 3. DAM 2000 FT. LONG IMPOUNDING 51,000 ACRE-FEET

TABLE 1. OPERATING DATA OBTAINED IN 1911.

Month	Kw.-Hr.	Monthly Peak	Load Factor, Per Cent.
January	3,100,000	6,300	66.1
February	2,910,000	6,350	68.2
March	3,360,000	6,500	69.4
April	3,050,000	7,400	56.8
May	3,690,000	7,950	62.4
June	3,830,000	7,650	69.5
July	4,200,000	7,050	80.0
August	4,290,000	8,850	65.1
September	4,090,000	9,250	61.4
October	4,560,000	10,000	61.3
November	4,260,000	10,000	59.2

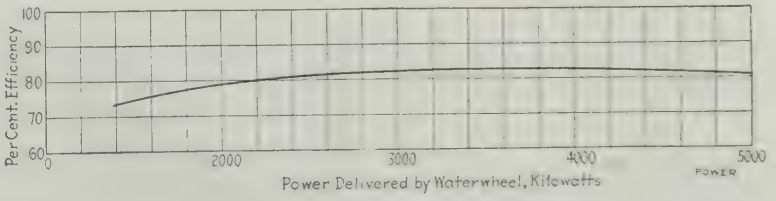


FIG. 4. EFFICIENCY LOAD CURVE

At Bakersfield the company has just finished installing a steam-auxiliary plant of 7000 kw. capacity. It contains one General Electric turbine and generator of 2000 kw., and an Allis-Chalmers turbine of 5000 kw. It is intended to run these units as rotary condensers when not needed for power, to correct the very low power factor.

Before the final acceptance of the waterwheels for the main plant, a thorough efficiency test upon the installation was made by the J. G. White Co., which had charge of the design and construction work for the power corporation. Two units were tested, No. 2, which had been in operation a year, and No. 4, which had just been set up. The former showed higher efficiency than the latter, due to the smoothing of the bearings by wear, and both machines considerably surpassed the contract guarantees, although the friction and windage of the generators were charged against the waterwheels.

A rather unusual method of loading was resorted to and found eminently successful. The current from the generators was stepped up to 30,000 volts in the transformers and run to a raft anchored in the tailrace. The three wires were connected to three 1½-in. pipes which projected down vertically into the water. They were 5 ft. apart and at the apices of an equilateral triangle. A previous trial had shown how deep the pipes should

go for each load desired. It was found that the current was perfectly divided between the three phases and absolutely constant during the 20-min. test runs. The reason for adopting the scheme was that the commercial load was too variable and at low power factor.

The input to the wheels was measured by gaging the pressure head behind the nozzles, adding the velocity head, and multiplying by the quantity of water as measured by a venturi meter in the pipe line. The output was taken as the mean of the readings of three wattmeters on the low-tension current, plus the armature loss, core loss and load losses, which last, in accordance with the recommendations of the American Institute of Electrical Engineers, were taken equal to one-third of the short-circuit core loss. The generator losses were calculated from the results of tests upon them by the J. G. White Co., at Schenectady, before acceptance.

Following is a tabulation of the results for the wheels tested:

Load		Guar.	Realized Efficiency	
Hp.	Kw.	Eff.	Unit No. 4	Unit No. 2
3,050	2,275	80%	80.0	80.6
4,575	3,413	81%	82.2	83.2
6,100	4,551	81%	82.2	82.4

Finally a building-up test of the efficiency from three-fourths load to 27 per cent. overload was run, resulting in the efficiency curve shown. This was on the newer, less efficient wheel, and as in the previous tests the windage and friction of the generator were neglected.

An Old Mine Pump

Modern mine pumps are constructed along much the same lines, but, as in the case of other mining machinery, decided changes have been made in their design during recent years. In Fig. 1 is shown the upper part of an old type of mine pump, which has been in operation for a period of 50 years. New valves and piston cylinders have been installed from time to time, but the original method of operation is still in use at the Barnum No. 2 mine of the Pennsylvania Coal Co., Pittston, Penn.

The two plungers and cylinder are down approximately 200 ft. in the shaft. Each plunger is 15 in. in diameter and outside packed; the stroke is 6 ft. The plungers are actuated by a series of wooden plunger rods, each 8 in. wide and 6 in. thick, and they are connected by hook-and-eye straps secured to the end of each rod.

The upper end of each series of plunger rods is connected to the inner corner of a triangular walking-beam 7 ft. long on each side. The bottom corner is fitted with a shaft which oscillates in suitable bearings. The top corner of each triangle is connected to a crossarm, Fig. 1, which is 14 in. square at the center and is strengthened on both ends by an iron band which is made with an eye connection through which a pin passes as well as through corresponding holes in the angle of the walking-beam. The pump-plunger rods are connected in the same manner. The walking-beams receive motion from a rod connected with one corner of the triangle at one end and to a crank at the other. The crank is attached to a crankshaft upon which is mounted a large gear wheel at the other end. This gear meshes with a spur gear which is keyed to the engine shaft, as shown in Fig. 2. The reduction in speed between the engine and counter crankshaft gives a speed to the pump plungers of 18 strokes per minute.

The engine which operates this pump has a single eccentric and is of the old hook-valve-gear type. It was built in 1849, and, although the type of engine has become practically obsolete, this particular unit is still in good condition and performs its duty daily.

A peculiar feature is noticeable in the manner of adjusting the bearings on both the engine bearings, crankshaft and walking-beam bearings. This is accomplished by wedges which drive in from the top and are held in place by set-screws.

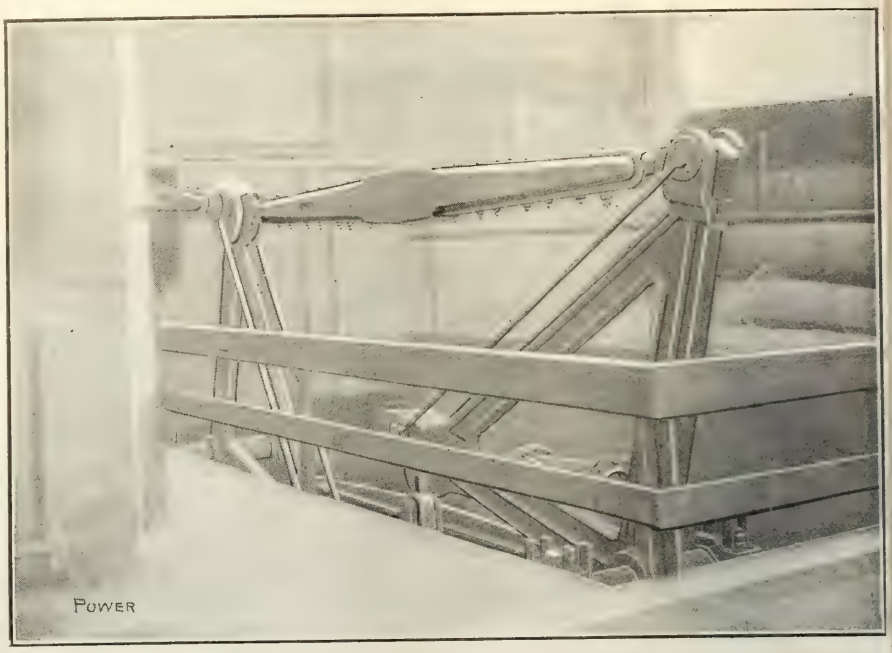


FIG. 1. UPPER WORKS OF OLD-TIME PUMP

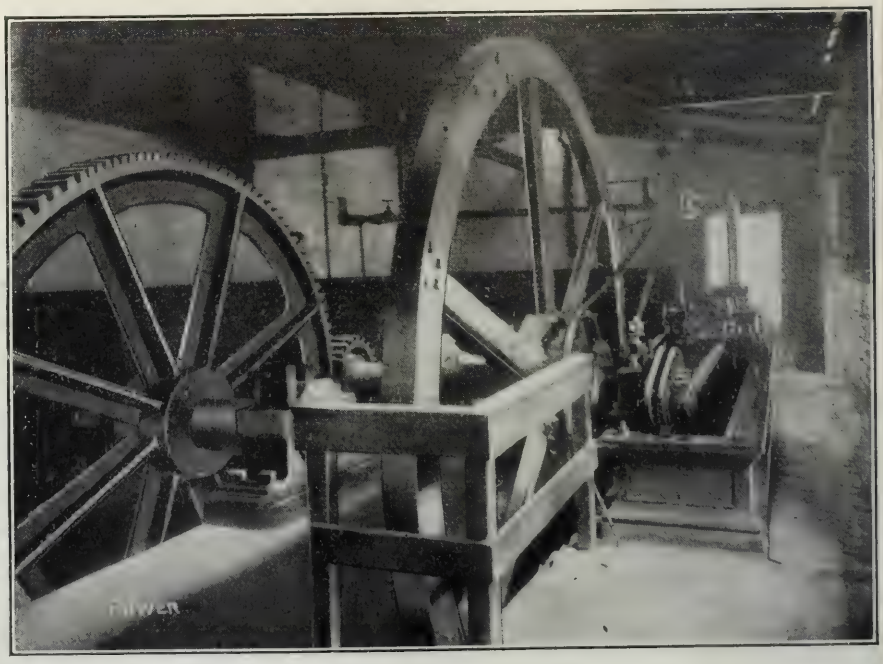


FIG. 2. OLD ENGINE WHICH OPERATES THE PUMP

A Modified Orsat Apparatus

By E. J. Billings

Many means are used for collecting samples of flue gases covering appreciable periods of time, the most common of which probably is that shown in Fig. 1, where water or brine is allowed to flow by gravity from one bottle to another, thereby drawing a sample of gas into the upper bottle. An aspirator may or may not be used with this device. Many users of flue-gas analysis apparatus do not attempt to collect continuous samples at all but analyze "instantaneous" samples or samples drawn directly from the boiler into the measuring burette of the apparatus. The Aug. 25, 1911, issue of Power, in a résumé of the Bureau of Mines Bulletin No. 12, describes a method ordinarily used in boiler tests made by the U. S. Government.

Below is a description of a combined apparatus for sampling and analyzing furnace gases as used in boiler-plant tests by Arthur D. Little, Inc., Boston. This apparatus was designed with the following points in view:

(a) To combine in a portable apparatus the sampling and analyzing devices.

(b) To find a simple means for collecting samples over any period of time up to 24 hours, which did not have the defects found in some, such as the absorption of carbon dioxide (CO_2) by the displacing liquid; the inability to always regulate the flow of liquid so as to obtain at a constant rate a sample covering the desired period; and the diffusion of gases between the sample being collected and the sampling line or boiler.

(c) To construct the apparatus so as to be compact and to contain no easily damaged parts or pieces which could not be readily replaced.

Fig. 2 shows the general plan of the apparatus designed to meet these requirements. The right half of the apparatus will be readily recognized as the ordinary Orsat apparatus, consisting mainly of three pipettes and a water-jacketed measuring burette. The left portion constitutes the sampling attachment. The gases are drawn from the sampling pipe in the flue or breeching through the rubber tube (2) into the sampling pipette (3) and out through the rubber tube (1) to the aspirator. The latter may be operated by steam or water. When a sample is being collected the three-way cock on the glass header is closed and the mercury in the sampling tube (4) is allowed to drain through the movable overflow (5) into the mercury container (7). The overflow (5) is lowered at a constant rate by clockwork (6). Two driving pulleys afford seven different rates of movement downward of the overflow (5). As the capacity of the sampling tube is 250 c.c. and only about 125 c.c. of gas at atmospheric pressure is needed for analysis, a sufficient sample may be collected at a constant rate over any period from ½ hr. to 24 hr.

To pass the gas sample collected from the sampling tube (4) into the measuring burette of the Orsat apparatus, the latter is filled with water from the leveling bottle in the usual way, the gas or air escaping through the three-way cock to the atmosphere. The upper glass cock on the sampling pipette (3) and that on the overflow (5) are closed. The mercury container (7) is then raised and set on top of the apparatus and the mercury allowed to drain back into the sampling tube (4), thereby forcing the gas through the three-way cock, previously opened, into the measuring burette. During this process the hand-leveling bottle is kept in the position shown in the illustration, and the spent gases are removed from the header by manipulating the three-way cock. To shift the sample and start the sampling apparatus to working again requires about two minutes. Instantaneous or "grab" samples may be drawn off and analyzed as often as desired and with

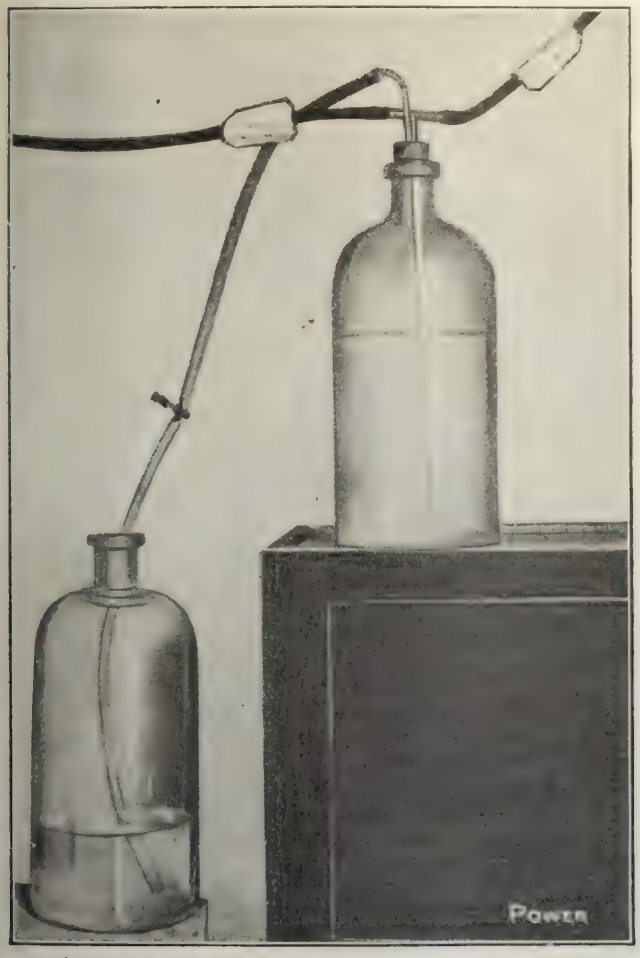


Fig. 1

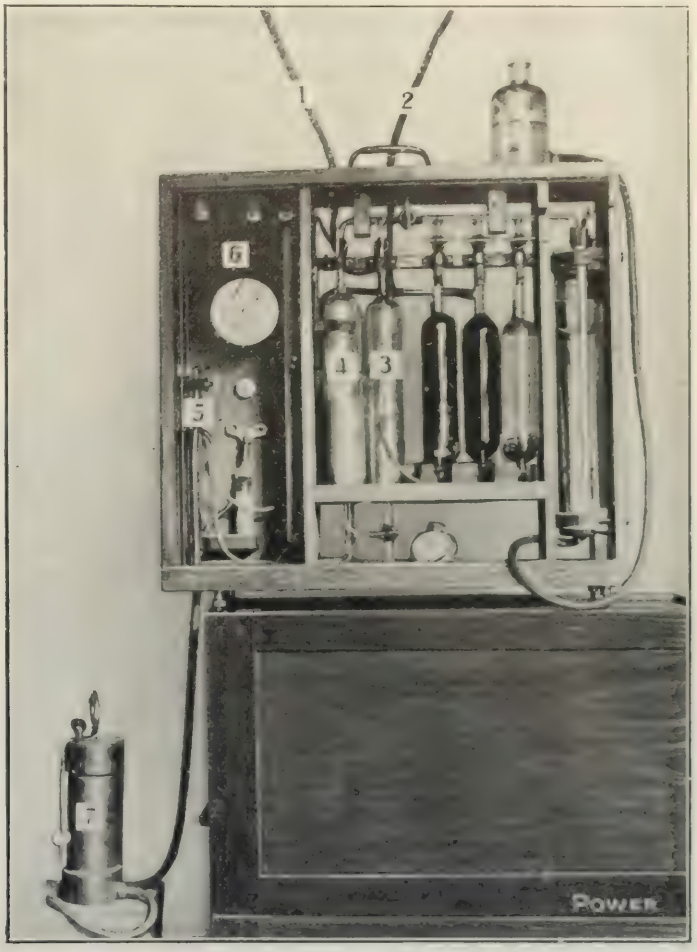


Fig. 2

A Modified Orsat Apparatus

practically no delay to the continuous sample.

Several features in the construction of the apparatus deserve particular attention; one is a rubber-tube connection between the sampling pipette (3) and the overflow (5) which prevents unevenness in the flow of mercury due to fluctuations in draft pressure in the former. Another is the special construction of the sampling pipette (3) within which is a water seal adjustable from the outside. The gas bubbles through this seal in passing into the sampling tube (4), thereby preventing diffusion of the gases between the latter and the sampling pipette (3). Before entering the pipette from the tube (2), the gases pass through a glass wool filter which removes the soot but allows the moisture to pass on into the pipette from which it may be removed from time to time through a cock at the bottom. The measuring burette has an especially large bulb and small column, permitting graduations in tenths up to 24 c.c., which is ample for flue-gas analyses. The front cover, when in place, prevents the glass cocks jarring loose and falling out during transportation.

Much care was taken to secure the glass parts firmly and so as not to be readily broken. The apparatus has been in almost constant use for six months and but two breakages have occurred. The header broke of its own accord due to improper annealing strains and one pipette was broken in filling. The sampling device has given practically no trouble and has always been in working order.

In transportation the container, with the mercury, weighs about 15 lb. and is carried separately. The apparatus proper is 18x20x5 in. in size and weighs 21 lb.

Standard Flanges and Flanged Fittings

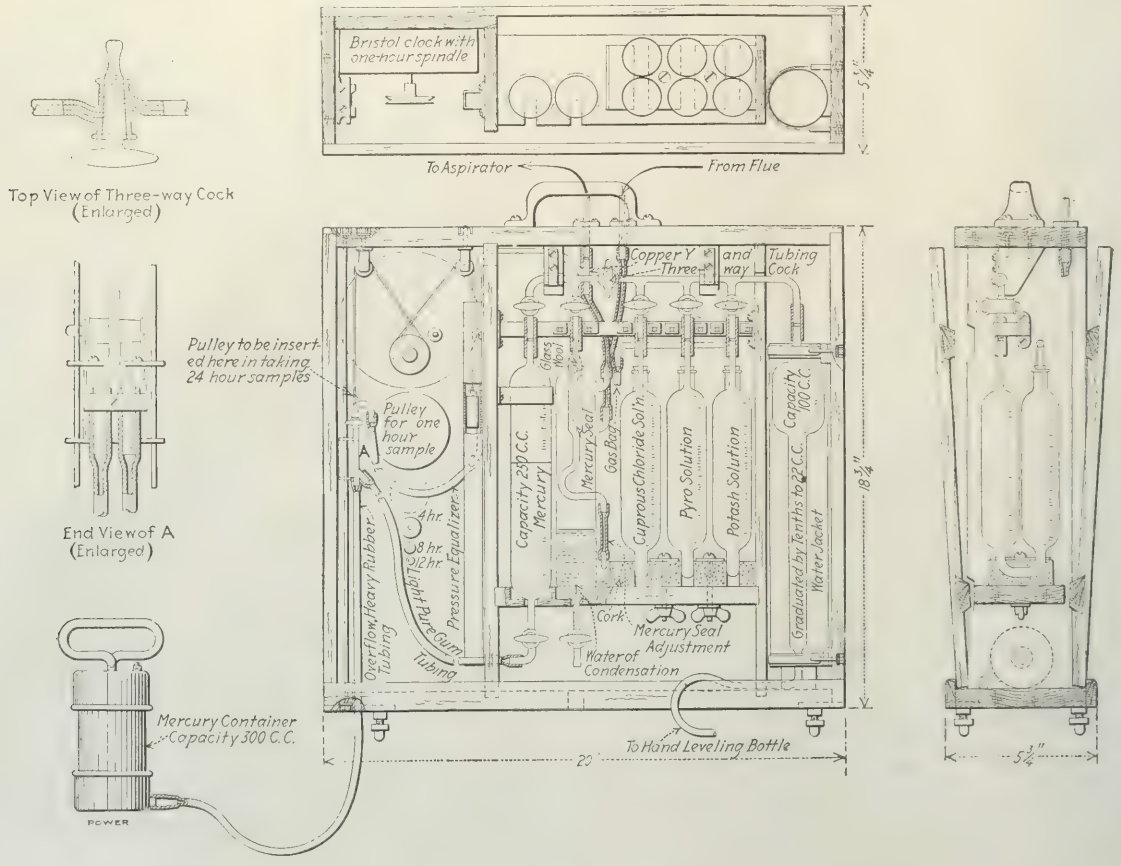
On June 17 a conference was held at the Navy Department in Washington at the request of the "Manufacturers Committee" of the makers of pipe flanges and flanged fittings to hear their protest against the adoption by the government of the "United States 1912 Standard." Present to defend it were Henry G. Stott, chairman of the committee of the American Society of Mechanical Engineers, and Henry B. Gombers, secretary of the National Association of Master Steam and Hot Water Fitters, a committee of which association worked in conjunction with that of which Mr. Stott is chairman in drawing up the 1912 standard.

The argument of the manufacturers' committee was to the effect that the standard had been adopted without sufficient notice to and discussion with them; that they have a large amount of money invested in patterns and catalogs for the present sizes, that the sizes now in use are in most cases in excess of those required by foreign standards, that they have been adopted after thorough tests and calculation and are believed to be adequate, but that if any change is to be made it should be done, in justice to the manufacturers, only after such investigation and discussion as to insure its being permanent.

Those responsible for the standard showed that they had made every reasonable attempt to interest the manufacturers in it previous to its adoption, that it represented the results of experience with and use of fittings, by those who are responsible for the erection and operation of large power plants, that each size has been thoroughly studied with regard, not only to its sufficiency, but the practicability of the design. The three principal points which were set forth were:

(1) Interchangeability of fittings in the new standard. For example, if one has a 12-in. line having a 4-in. tee on it, he can drop the fitting out and substitute another having a 6-in., 8-in., or 10-in. tee, without altering the main line. In other words, all fittings, as far as the main lines are concerned, are interchangeable, so that it is not necessary to cut or put in new lengths on the main run. This interchangeability is one of the leading points in favor of the new standard and no other standard attempts to provide it.

(2) It has been shown conclusively that when the pulling-up stresses on the



A Modified Orsat Apparatus

bolt, which are necessary to make a tight joint, are allowed for (leaving out of consideration the expansion and contraction stresses) the ordinary fitting, as now used, has a factor of safety of only 2 or less, while the pipe itself has a factor of safety of from 16 to 20. As a result of experiments carried out at Cornell University on the stresses placed upon the bolts in making steam-tight joints, it was found that the stresses varied inversely as the diameter of the bolt. In order to get in the larger bolts, it was necessary to enlarge the bolt circle. This, of course, meant enlarging the flange diameter.

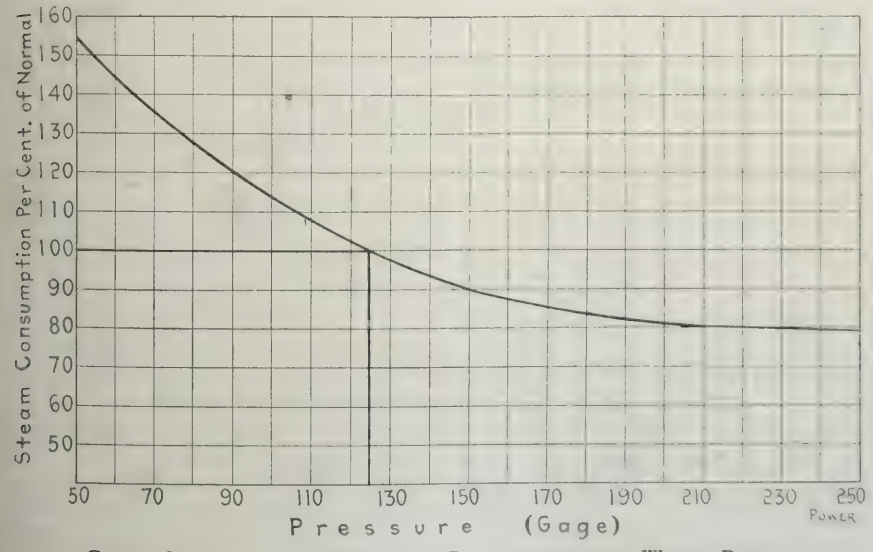
(3) The committee had in mind also that the extra-heavy fittings would in all probability be made of steel, and that too short a fitting would make a very difficult casting for the steel foundries. It is well known that it is not advisable to use cast iron with superheated steam, and as practically all turbine plants use superheated steam at a high pressure it would have been a mistake not to take this into consideration.

Mr. Sparrow, of the New York Edison Co., has conducted tests to measure the movement of a steam main, due to expansion and contraction under load conditions, which show the enormous strains put upon the flanges in the everyday work of a steam line.

If action was taken by the government authorities it has not been announced, but several of the departments are specifying the 1912 standard.

Effect of Pressure on Water Rate of Steam Turbine

The chart herewith, for which POWER is indebted to the engineers of the B. F. Sturtevant Co., shows the effect of a variation of initial pressure upon the water rate of the Sturtevant turbine.



CURVE SHOWING THE EFFECT OF PRESSURE ON THE WATER RATE OF A STEAM TURBINE

The normal pressure is taken at 125 lb., at which pressure the water rate, shown in the left-hand column, is 100 per cent. If the pressure drops to 90 lb., the steam consumption may be expected to increase to 120 per cent. of that at the normal pressure. If the pressure is increased to 220 lb., the consumption will be reduced to 80 per cent. of that at 125 lb.

Method for Filing Literature

BY LEWIS BEEBEE

Several articles giving methods for filing miscellaneous literature from papers and magazines have been published in POWER from time to time. As the writer has had a satisfactory system in service for the past two years which differs from those previously mentioned, it is believed that a detailed description of his method will prove its simplicity and completeness.

The magazines and papers from which the articles are taken should be unbound and the pages containing the articles kept whole; the remainder of the magazine or paper can be kept in a storeroom for possible future reference for some article not already selected.

The size of the cabinet, the folders and the cards will in each case need to be chosen to contain articles without folding for the specific literature to be filed, and the filing cabinet should have as many pockets as it has subjects. The standard letter-file folder of heavy material is satisfactory. The index to the file can be made up of any standard-size sheets best found in the user's line of work. The first pages in the index should contain an alphabetical list of the subjects chosen, and the pages following should be in the order of subjects indexed and numbered, and give the source from which the article was received, with the date of publication.

On the top of each folder must be printed or written the same reference name and number as the subject in the alphabetical index, as shown in Fig. 1. The cards are numbered as shown in Fig. 1 and will have the articles attached as outlined on page 1 of the index under "Air." The card shown in Fig. 2 will have the articles attached as outlined on page 2 of the index under the heading "Boilers."

The articles can be fastened to the card by any standard clip, and when a card has as many articles attached as desired, the use of the split-paper brad is very good as these articles will probably always remain attached to this individual card. To continue filing subjects under this head, the second card should have reference numbers as shown in Fig. 3.

FIG. 1 FIG. 2 FIG. 3

If two subjects have the same alphabetical letter, as "Engine" and "Elevator," then index the first as follows: 5—Engines, 5A—Elevators, 5B—Etc.

When a card having a set of articles as indexed and filed under "Air" has been taken from the file, and at some later date it is desired to replace it in the filing cabinet, it is only necessary to observe the first figure, which refers to the subject and the folder, and from this reference it can be filed direct without referring to the index.

When an article is wanted, go to the index, find the subject under which the article is filed (use subject "Air"); then turn to the following itemized sheets containing the articles in this file, and when the article is found in the index, note the first number 1, which should be in larger type and have a dash between it and the following numbers, and the number 2 following the dash. Then go to the file and find the card having as its reference figures 1 — 1, 2, 3, etc., which will have the article desired.

For example, suppose an article on boilers contains some good data on bituminous coals. This article is filed under "2 — Boilers — 2" and on the card having the reference number "2 —

METHOD FOR FILING

Page 1	*Subject 1—Air—1*	*Source of Article*	*Date*
1—1	Air compressors	POWER	September 5, 1911
1—2	Air drills	POWER	
1—3	Etc.		
Page 2	*Subject 2—Boilers—2*	*Source of Article*	*Date*
2—1	Boiler construction	POWER	September 5, 1911
2—2	Boiler data	POWER	
2—3	Etc.		
Page 3	*Subject 3—Condensers—3*	*Source of Article*	*Date*
3—1	Condensers, jet type	POWER	September 5, 1911
3—2	Condensers, barometric type	POWER	
3—3	Condenser pumps		
3—4	Etc.		
Page 4	*Subject 4—Dust Blowers—4*	*Source of Article*	*Date*
4—1	Fan blowers, general type	POWER	September 5, 1911
4—2	Fan blowers, engine driven	POWER	
4—3	Fan blowers, h.p. required		
4—4	Etc.		
Page 5	*Subject 5—Engines—5*	*Source of Article*	*Date*
5—1	Engines, high speed	POWER	September 5, 1911
5—2	Engines, slow speed	POWER	
5—3	Engine horsepower		
5—4	Etc.		
Page 6	*Subject 6—Fuel—6*	*Source of Article*	*Date*
6—1	Fuel, anthracite	POWER	September 5, 1911
6—2	Fuel, bituminous		
6—3	Fuel compositions		
2—2	Boiler data (Cross-indexed, see explanation.)		
6—4	Fuel		

1, 2, 3, Etc.," is itemized in the index as "2 — 2." That this information may be available under bituminous coal, a reference should be made, as shown on page 6, subject "6 — Fuel — 6," as "2 — 2," boiler data. This will show that this article is a main subject on boiler data and that it contains information on this subject—Fuel.

A card index can be used in place of the sheet form, but the sheet form is handled more readily and can be placed within the file and be at hand.

The subjects would of necessity have to be chosen to cover the line of work or information desired for filing.

The accompanying table shows the method of indexing the subjects and their sources. Of course, it is to be understood that each subject chosen is to be indexed as are those given as an illustration.

Low Pressure Steam Tables*

By Lionel S. Marks

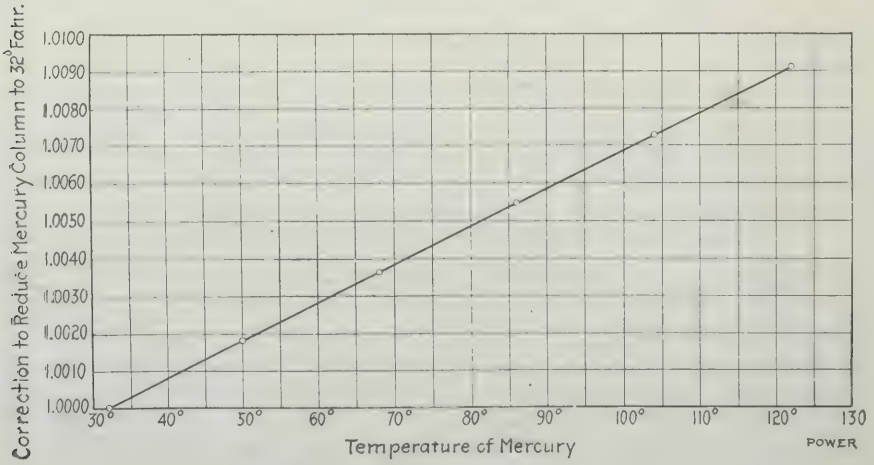
The data for condenser work are usually given in inches of mercury while the available steam tables advance by even pounds pressure and even degrees of temperature. The values for a given vacuum or pressure expressed in inches of mercury are frequently considerably different from those corresponding with the even pounds and degrees, making interpolation necessary. Prof. Marks has calculated, at our request, the following tables, giving for the pressures below that of the atmosphere the properties for even degrees, even tenths of pounds and tenth-inches of mercury.

In condenser practice, vacuums are commonly stated in inches of mercury. The vacuum is the difference between the absolute pressure of the atmosphere and that in the condenser. Knowing the atmospheric pressure as indicated by the barometer and the given or observed vacuum in inches, the absolute pressure in the condenser may be found by taking the difference.

The tables upon the following pages give the properties of dry and saturated steam at pressures which are expressed in inches of mercury as well as in pounds per square inch. Furthermore the values are given for even tenths of pounds pressure and even tenths of inches of mercury in the earlier part of the table, where the differences in the other values are considerable for comparatively small differences in temperature. The tables are an expansion of the Marks-Davis tables in this subatmospheric portion of the range and have been calculated especially for POWER by Prof. Marks.

It should be noted, however, that these pressures, as are those in the regular Marks-Davis tables, are expressed in terms of mercury at the standard temperature, 32 deg. At that temperature the standard atmosphere of 760 mm. of mercury, or of 14.696 lb. per square inch (approximately 14.7 lb.), corresponds to 29.921 in. of mercury; consequently 1 lb. per square inch corresponds to 2.036 in. of mercury, and 1 in. of mercury to 0.4902 lb. per square inch.

If the mercury is at a higher temperature its density will be less and it will take a greater height of column to balance a given pressure. If extreme accuracy is desired, the heights of the columns of mercury by which the vacuum and atmospheric pressures are measured should be reduced to their equivalent heights at 32 deg. before the subtraction is made. The accompanying chart will assist in such reduction. It is necessary simply to divide the given or observed vacuum or height by the number on the scale at the left corresponding with the height of the intersection of the line of the diagram with the ordinate representing the temperature.



As an example assume

Barometer	Mercury column on condenser	Uncorrected difference
30.42 in.	27.42 in.	3 in.

Suppose the temperature of the barometer to be 70 deg. and of the mercury column on the condenser to be 80 deg. The correction factor for 70 deg. is seen to be 1.00383 and for 80 deg. 1.00485. The corrected values would therefore be

$$\frac{30.42}{1.00383} - \frac{27.42}{1.00485} =$$

$$30.3 - 27.29 = 3.01 \text{ in.}$$

TABLE OF THE PROPERTIES OF DRY AND SATURATED STEAM AT PRESSURES BELOW THAT OF THE ATMOSPHERE—By Lionel S. Marks

Enlarged from the "Steam Tables" of Marks and Davis, with the permission of the publishers, Messrs. Longmans, Green & Co.

Temp. Fahr.	Pressure, lbs. per sq. in.	Pressure, inches of Mercury (at 32° F.)	Sp. vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	External Work of Evaporation in B.t.u.	Internal energy of Evaporation	Internal energy of Steam	Entropy of Water	Entropy of Evaporation	Entropy of Steam
t	p		v	1/v	h	L	H		I	E	n	L/T	N
32.00	0.0886	0.1804	3294	0.000304	0.00	1073.4	1073.4	54.1	1019.3	1019.3	0.0000	2.1832	2.1832
34.57	0.0980	0.2000	2987	0.000335	2.59	1071.9	1074.5	54.3	1017.6	1020.2	0.0053	2.1690	2.1743
35.03	0.1000	0.2036	2935	0.000340	3.05	1071.7	1074.7	54.4	1017.3	1020.3	0.0062	2.1666	2.1728
40.00	0.1217	0.2477	2438	0.000410	8.05	1068.9	1076.9	54.9	1014.0	1022.0	0.0162	2.1394	2.1556
41.00	0.1265	0.2575	2350	0.000425	9.05	1068.3	1077.4	55.0	1013.3	1022.3	0.0182	2.1341	2.1523
42.00	0.1315	0.2677	2266	0.000441	10.06	1067.8	1077.8	55.2	1012.6	1022.7	0.0202	2.1287	2.1489
43.00	0.1366	0.2782	2185	0.000458	11.06	1067.2	1078.3	55.3	1012.0	1023.0	0.0222	2.1234	2.1456
44.00	0.1420	0.2890	2107	0.000475	12.06	1066.7	1078.7	55.4	1011.3	1023.4	0.0242	2.1181	2.1423
44.98	0.1470	0.3000	2035	0.000492	13.05	1066.2	1079.2	55.5	1010.6	1023.7	0.0262	2.1125	2.1390
45.00	0.1475	0.3002	2033	0.000492	13.07	1066.1	1079.2	55.5	1010.6	1023.7	0.0262	2.1127	2.1389
46.00	0.1532	0.3118	1961	0.000510	14.07	1065.6	1079.6	55.6	1010.0	1024.0	0.0282	2.1074	2.1356
47.00	0.1591	0.3238	1892	0.000529	15.07	1065.0	1080.1	55.7	1009.3	1024.4	0.0301	2.1022	2.1323
48.00	0.1651	0.3363	1826	0.000548	16.07	1064.5	1080.5	55.9	1008.6	1024.7	0.0321	2.0970	2.1291
49.00	0.1715	0.3492	1763	0.000567	17.08	1063.9	1081.0	56.0	1007.9	1025.0	0.0341	2.0917	2.1258
50.00	0.1780	0.3625	1702	0.000587	18.08	1063.3	1081.4	56.1	1007.3	1025.4	0.0361	2.0865	2.1226
51.00	0.1848	0.3762	1643	0.000608	19.08	1062.8	1081.9	56.2	1006.6	1025.7	0.0381	2.0814	2.1195
52.00	0.1917	0.3903	1586	0.000630	20.08	1062.2	1082.3	56.3	1006.0	1026.1	0.0401	2.0763	2.1164
52.67	0.1960	0.4000	1551	0.000645	20.75	1062.0	1082.6	56.4	1005.5	1026.3	0.0413	2.0729	2.1142
53.00	0.1989	0.4049	1532	0.000653	21.08	1061.7	1082.7	56.4	1005.3	1026.4	0.0420	2.0712	2.1132
53.15	0.2000	0.4072	1524	0.000656	21.23	1061.6	1082.8	56.5	1005.2	1026.5	0.0423	0.0704	2.1127
54.00	0.2063	0.4201	1480	0.000676	22.08	1061.1	1083.2	56.5	1004.6	1026.7	0.0440	0.0660	2.1100
55.00	0.2141	0.4357	1430	0.000700	23.08	1060.6	1083.6	56.6	1004.0	1027.1	0.0459	2.0609	2.1068
56.00	0.2219	0.4518	1381	0.000724	24.08	1060.0	1084.1	56.7	1003.3	1027.4	0.0478	0.0559	2.1037
57.06	0.2301	0.4684	1335	0.000749	25.08	1059.5	1084.5	56.8	1002.7	1027.7	0.0498	0.0508	2.1006
58.00	0.2385	0.4856	1291	0.000775	26.08	1058.9	1085.0	56.9	1002.0	1028.1	0.0517	2.0458	2.0975
58.81	0.2450	0.5000	1257	0.000792	26.89	1058.4	1085.3	57.0	1001.4	1028.3	0.0532	2.0418	2.0951
59.00	0.2472	0.5034	1249	0.000801	27.08	1058.3	1085.4	57.0	1001.3	1028.4	0.0536	2.0408	2.0944
60.00	0.2562	0.5220	1208	0.000828	28.08	1057.8	1085.9	57.1	1000.7	1028.7	0.0555	2.0358	2.0913
61.00	0.2654	0.5410	1168	0.000856	29.08	1057.2	1086.3	57.2	1000.0	1029.1	0.0574	0.0308	2.0882
62.00	0.2749	0.560	1130	0.000885	30.08	1056.7	1086.8	57.3	999.3	1029.4	0.0593	2.0258	2.0851
63.00	0.2847	0.580	1093	0.000915	31.07	1056.1	1087.2	57.4	998.7	1029.8	0.0612	2.0209	2.0821
63.95	0.2940	0.600	1060	0.000944	32.02	1055.7	1087.7	57.5	998.1	1030.1	0.0630	2.0163	2.0793
64.00	0.2949	0.601	1058	0.000946	32.07	1055.6	1087.6	57.6	998.0	1030.1	0.0631	2.0160	2.0791
64.49	0.3000	0.611	1041	0.000961	32.56	1055.3	1087.9	57.6	997.7	1030.2	0.0640	2.0136	2.0776
65.00	0.3054	0.622	1024	0.000977	33.07	1055.0	1088.1	57.7	997.4	1030.4	0.0650	2.0110	2.0760
66.00	0.3161	0.644	991	0.001009	34.07	1054.5	1088.5	57.8	996.7	1030.8	0.0669	2.0062	2.0731
67.00	0.3272	0.667	959	0.001043	35.07	1053.9	1089.0	57.9	996.0	1031.1	0.0688	2.0013	2.0701
68.00	0.3386	0.690	928	0.001077	36.07	1053.4	1089.4	58.0	995.4	1031.4	0.0707	1.9965	2.0672
68.42	0.3430	0.700	916	0.001092	36.47	1053.1	1089.6	58.0	995.1	1031.6	0.0715	1.9945	2.0660
69.00	0.3504	0.714	899	0.001112	37.06	1052.3	1089.9	58.1	994.7	1031.8	0.0726	1.9916	2.0642
70.00	0.3626	0.739	871	0.001148	38.06	1052.3	1090.3	58.3	994.0	1032.1	0.0745	1.9868	2.0613
71.00	0.3751	0.764	843	0.001186	39.06	1051.7	1090.8	58.4	993.4	1032.4	0.0764	1.9821	2.0585
72.00	0.3880	0.790	817	0.001224	40.05	1051.2	1091.2	58.5	992.7	1032.8	0.0783	1.9773	2.0556
72.37	0.3920	0.800	808	0.001238	40.42	1051.0	1091.4	58.6	992.4	1032.9	0.0790	1.9756	2.0546
72.91	0.4000	0.814	794	0.001259	40.96	1050.5	1091.5	58.6	992.1	1033.1	0.0800	1.9730	2.0530
73.00	0.4012	0.817	792	0.001263	41.05	1050.6	1091.6	58.6	992.0	1033.1	0.0802	1.9726	2.0528
74.00	0.4148	0.845	767	0.001304	42.05	1050.0	1092.1	58.7	991.4	1033.4	0.0821	1.9678	2.0499
75.00	0.4288	0.873	743	0.001346	43.05	1049.5	1092.5	58.8	990.7	1033.8	0.0840	1.9631	2.0471
75.90	0.4410	0.900	722	0.001385	43.94	1049.0	1092.9	58.8	990.2	1034.1	0.0856	1.9590	2.0446
76.00	0.4432	0.903	720	0.001389	44.04	1048.9	1093.0	58.9	990.1	1034.1	0.0858	1.9585	2.0443
77.00	0.4581	0.933	698	0.001433	45.04	1048.4	1093.4	59.0	989.4	1034.4	0.0876	1.9538	2.0414
78.00	0.4735	0.964	677	0.001477	46.04	1047.8	1093.9	59.1	988.7	1034.8	0.0895	1.9491	2.0386
79.00	0.4893	0.996	657	0.001523	47.04	1047.3	1094.3	59.2	988.1	1035.1	0.0913	1.9445	2.0358
79.12	0.4912	1.000	654	0.001529	47.16	1047.2	1094.4	59.2	988.0	1035.1	0.0915	1.9440	2.0355
79.68	0.5000	1.153	642	0.001555	47.71	1046.9	1094.6	59.3	987.6	1035.3	0.0926	1.9413	2.0339
80.00	0.505	1.029	636.8	0.001570	48.03	1046.7	1094.8	59.3	987.4	1035.4	0.0932	1.9398	2.0330
81.00	0.522	1.063	617.5	0.001619	49.03	1046.2	1095.2	59.4	986.7	1035.8	0.0950	1.9352	2.0302
82.00	0.539	1.098	598.7	0.001670	50.03	1045.6	1095.6	59.5	986.1	1036.1	0.0969	1.9306	2.0275
82.06	0.539	1.100	598.0	0.001673	50.09	1045.6	1095.7	59.5	986.1	1036.1	0.0970	1.9303	2.0273
83.00	0.557	1.134	580.5	0.001723	51.02	1045.1	1096.1	59.7	985.4	1036.4	0.0987	1.9260	2.0247
84.00	0.575	1.171	562.9	0.001777	52.02	1044.5	1096.5	59.8	984.8	1036.8	0.1005	1.9215	2.0220
84.76	0.588	1.200	550.0	0.001819	52.78	1044.1	1096.9	59.8	984.3	1037.3	0.1019	1.9180	2.0199
85.00	0.594	1.209	545.9	0.001832	53.02	1044.0	1097.0	59.9	984.1	1037.1	0.1023	1.9169	2.0192
85.32	0.600	1.222	540.0	0.001850	53.34	1043.8	1097.1	59.9	983.9	1037.2	0.1029	1.9159	2.0180
86.00	0.613	1.248	529.5	0.001889	54.01	1043.4	1097.4	60.0	983.4	1037.4	0.1041	1.9124	2.0165
87.00	0.633	1.289	513.7	0.001947	55.01	1042.8	1097.9	60.0	982.8	1037.8	0.1060	1.9079	2.0139
87.21	0.637	1.300	510.0	0.001960	55.22	1042.7	1098.0	60.1	982.7	1037.9	0.1064	1.9070	2.0134
88.00	0.654	1.331	498.4	0.002007	56.01	1042.2	1098.3	60.2	982.1	1038.1	0.1078	1.9034	2.0112
89.00	0.675	1.373	483.6	0.002068	57.00	1041.7	1098.7	60.3	981.4	1038.4	0.1096	1.8989	2.0085
89.61	0.686	1.400	474.0	0.002030	57.61	1041.4	1099.0	60.4	981.0	1038.6	0.1108	1.8962	2.0070
90.00	0.696	1.417	469.3	0.002131	58.00	1041.2	1099.2	60.4	980.8	1038.8	0.1114	1.8944	2.0058
90.18	0.700	1.425	467.0	0.002143	58.18	1041.1	1099.3	60.4	980.7	1039.0	0.1117	1.8935	2.0052
91.00	0.718	1.462	455.5	0.002195	59.00	1040.6	1099.6	60.5	980.1	1039.1	0.1133	1.8890	2.0023
91.83	0.735	1.500	444.9	0.002250	59.83	1040.1	1099.9	60.6	979.5	1039.3	0.1148	1.8863	2.0011
92.00	0.741	1.508	442.2	0.002261	60.00	1040.0	1100.1	60.6	979.4	1039.4	0.1151	1.8856	2.0007
93.00	0.765	1.556	429.4	0.002329	60.99	1039.5	1100.5	60.7	978.8	1039.8	0.1169	1.8812	1.9981
93.90	0.784	1.600	418.0	0.002391	61.90	1039.1	1101.0	60.8	978.2	1040.1	0.1185	1.8771	1.9956
94.00	0.789	1.605	417.0	0.002398	61.99	1039.0	1101.0	60.9	978.1	1040.1	0.1187	1.8767	1.9954
94.46	0.800	1.629	411.0	0.002431	62.45	1038.7	1101.2	60.9	977.8	1040.2	0.1195	1.8747	1.9942
97.76	0.882	1.800	374.0	0.002675	65.74	1036.8	1102.5	61.2	975.6	1041.3	0.1248	1.8603	1.9858
98.00	0.891	1.813	371.4	0.002693	65.98	1036.7	1102.8	61.2	975.5	1041.4	0.1253	1.8592	1.9851
98.33	0.900	1.832	367.9	0.002719	66.31	1036.5	1102.8	61.3	975.3	1041.5	0.1260	1.8578	1.9843
99.00	0.918	1.869	360.9	0.002771	66.97	1036.2	1103.2	61.4	974.8	1041.8	0.1272	1.8549	1.9826
99.50	0.931	1.900	355.9	0.002811	67.47	1035.9	1103.4	61.4	974.5	1041.9	0.1281	1.8527	1.9813
100.00	0.946	1.926	350.8	0.002851	67.97	1035.6	1103.6	61.5	974.1	1042.1	0.1290	1.8505	1.9800
101.00	0.975	1.985	341.0	0.002933	68.97	1035.1	1104.0	61.6	973.5	1042.4	0.1308	1.8463	1.9776
101.25	0.980	2.000	338.6	0.002954	69.22	1035.0	1104.2	61.7	973.3	1042.5	0.1312	1.8452	1.9769
101.84	1.000	2.036	333.0	0.003000	69.80	1034.6	1104.4	61.7	972.9	1042.7	0.1323	1.8427	1.9754
102.00	1.005	2.045	331.5	0.003017	69.96	1034.5	1104.5	61.8	972.8	1042.8	0.1326	1.8420	1.9750
102.89	1.029	2.100	323.4	0.003095	70.85	1034.1	1104.9	61.8	972.2	1043.1	0.1343	1.8382	1.9727
103.00	1.035	2.107	322.2	0.003104	70.96	1034.0	1104.9	61.9	972.1	1043.1	0.1345	1.8377	1.9724
104.00	1.066	2.171	313.3	0.003192	71.96	1033.4	1105.3	61.9	971.5	1043.4	0.1363	1.8341	1.9703
104.15	1.078	2.200	309.1	0.003232	72.10	1033.3	1105.5	61.9	971.2	1043.6	0.1370	1.8316	1.9686
105.00	1.098	2.236	304.7	0.003282	72.95	1032.8	1105.8	62.0	970.8	1043.8	0.1383	1.8292	1.9675
105.06	1.100	2.240	304.2	0.003287	73.01	1032.8	1105.8	62.0	970.8	1043.8	0.1384	1.8289	1.9673
105.55	1.127	2.300	300.1	0.003330	73.50	1032.5	1106.0	62.1	970.4	1044.0	0.1395	1.8269	1.9662
106.00	1.131	2.303	296.1	0.003374	73.95	1032.3	1106.2	62.1	970.1	1044.1	0.1404	1.8250	1.9654
107.00	1.165	2.372	288.3	0.003469	74.95	1031.7	1106.7	62.2	969.5	1044.1	0.1418	1.8208	1.9626
107.10	1.176	2.400	285.2	0.003507	75.05	1031.5	1106.8	62.3	969.2	1044.6	0.1425	1.8194	1.9618
108.00	1.199	2.443	280.5	0.003565	75.95	1031.2	1107.1	62.3	968.8	1044.8	0.1436	1.8169	1.9602
108.03	1.200	2.443	280.3	0.003568	75.98	1031.2	1107.2	62.4	968.8	1044.8	0.1437	1.8168	1.9602
108.80	1.225	2.500	274.1	0.003644	76.73	1030.7	1107.4	62.4	968.3	1045.0	0.1450	1.8133	1.9583
109.00	1.235	2.515	272.9	0.003664	76.94	1030.6	1107.5	62.4	968.2	1044.1	0.1454	1.8124	1.9578
110.00	1.271	2.589	265.5	0.003766	77.91	1030.0	1108.0	62.5	967.5	1045.4	0.1471	1.8082	1.9553
110.15	1.274	2.600	264.1	0.003782	78.09	1029.9	1108.1	62.5	967.4	1045.5	0.1474	1.8076	1.9550
110.79	1.300	2.647	259.1	0.003859	78.83	1029.6	1108.3	62.6	966.9	1045.7	0.1487	1.8046	1.9533

TABLE OF THE PROPERTIES OF DRY AND SATURATED STEAM AT PRESSURES BELOW THAT OF THE ATMOSPHERE.—By Lionel S. Marks.—(Continued)

Temp. Fahr.	Pressure, lbs. per sq. in.	Pressure, inches of Mercury (at 32° F.)	Sp. vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	External Work of Evaporation in B.t.u.	Internal energy of Evaporation	Internal energy of Steam	Entropy of Water	Entropy of Evaporation	Entropy of Steam
t	p		v	1/v	h	L	H		I	E	n	L/T	N
111.00	1.308	2.665	258.3	0.003871	78.94	1029.5	1108.4	62.6	966.8	1045.8	0.1489	1.8041	1.9530
111.47	1.323	2.700	255.2	0.003921	79.40	1029.2	1108.6	62.7	966.6	1045.9	0.1497	1.8022	1.9519
112.00	1.346	2.740	251.4	0.003978	79.93	1028.9	1108.8	62.7	966.2	1046.1	0.1506	1.8000	1.9506
112.73	1.372	2.800	246.5	0.004058	80.66	1028.5	1109.2	62.8	965.7	1046.3	0.1519	1.7970	1.9489
113.00	1.386	2.822	244.7	0.004087	80.93	1028.4	1109.3	62.9	965.5	1046.4	0.1524	1.7959	1.9483
113.35	1.400	2.850	242.4	0.004126	81.28	1028.2	1109.5	62.9	965.3	1046.5	0.1530	1.7944	1.9474
113.95	1.421	2.900	238.5	0.004192	81.88	1027.8	1109.7	63.0	964.8	1046.8	0.1540	1.7919	1.9459
114.00	1.426	2.904	238.2	0.004198	81.93	1027.8	1109.7	63.0	964.8	1046.8	0.1541	1.7917	1.9458
115.00	1.467	2.987	231.9	0.004312	82.92	1027.2	1110.2	63.0	964.2	1047.1	0.1559	1.7876	1.9435
115.15	1.470	3.000	239.0	0.004330	83.07	1027.1	1110.2	63.0	964.1	1047.1	0.1562	1.7872	1.9432
115.79	1.500	3.054	227.1	0.004404	83.71	1026.8	1110.5	63.1	963.5	1047.3	0.1572	1.7844	1.9417
116.00	1.509	3.073	225.8	0.004429	83.92	1026.7	1110.6	63.1	963.5	1047.4	0.1576	1.7836	1.9412
116.31	1.519	3.100	224.0	0.004466	84.23	1026.5	1110.7	63.2	963.3	1047.5	0.1582	1.7823	1.9405
117.00	1.553	3.161	219.9	0.004548	84.92	1026.1	1111.0	63.2	962.8	1047.8	0.1594	1.7795	1.9389
117.43	1.568	3.200	217.4	0.004601	85.35	1025.8	1111.1	63.3	962.5	1047.9	0.1601	1.7778	1.9379
118.00	1.597	3.252	214.1	0.004671	85.92	1025.5	1111.5	63.3	962.2	1048.1	0.1611	1.7755	1.9366
118.07	1.600	3.258	213.8	0.004680	85.99	1025.5	1111.5	63.4	962.2	1048.1	0.1613	1.7752	1.9365
118.52	1.617	3.300	211.2	0.004736	86.44	1025.2	1111.6	63.4	961.8	1048.3	0.1620	1.7734	1.9354
119.00	1.642	3.344	208.5	0.004796	86.91	1025.0	1111.9	63.5	961.5	1048.4	0.1628	1.7715	1.9343
119.60	1.665	3.400	205.3	0.004873	87.51	1024.6	1112.1	63.5	961.1	1048.6	0.1638	1.7690	1.9328
120.06	1.689	3.438	203.1	0.004924	87.91	1024.4	1112.3	63.6	960.8	1048.7	0.1645	1.7674	1.9319
120.23	1.700	3.461	201.9	0.004954	88.14	1024.3	1112.4	63.6	960.7	1048.8	0.1649	1.7665	1.9314
120.64	1.715	3.500	199.8	0.005007	88.55	1024.1	1112.6	63.7	960.4	1048.9	0.1656	1.7648	1.9304
121.00	1.736	3.535	197.9	0.005054	88.91	1023.9	1112.8	63.7	960.2	1049.1	0.1662	1.7634	1.9296
121.65	1.764	3.600	194.6	0.005140	89.56	1023.5	1113.1	63.8	959.7	1049.3	0.1673	1.7508	1.9281
122.00	1.785	3.635	192.8	0.005187	89.91	1023.3	1113.2	63.8	959.5	1049.4	0.1679	1.7594	1.9273
122.30	1.800	3.665	191.3	0.005228	90.21	1023.2	1113.4	63.9	959.3	1049.5	0.1684	1.7582	1.9266
122.65	1.812	3.700	189.7	0.005274	90.54	1023.0	1113.5	63.9	959.1	1049.6	0.1690	1.7569	1.9259
123.00	1.835	3.737	187.9	0.005323	90.90	1022.7	1113.6	63.9	958.8	1049.7	0.1696	1.7555	1.9251
123.61	1.862	3.800	185.0	0.005408	91.51	1022.4	1113.9	64.0	958.4	1049.9	0.1706	1.7531	1.9237
124.00	1.886	3.841	183.1	0.005462	91.90	1022.2	1114.1	64.0	958.2	1050.0	0.1713	1.7515	1.9228
124.25	1.900	3.868	181.9	0.005498	92.15	1022.1	1114.2	64.1	958.0	1050.1	0.1717	1.7505	1.9222
124.55	1.911	3.900	180.5	0.005541	92.45	1021.9	1114.3	64.1	957.8	1050.2	0.1722	1.7493	1.9215
125.00	1.938	3.948	178.4	0.005605	92.90	1021.6	1114.5	64.2	957.5	1050.4	0.1730	1.7475	1.9205
125.48	1.960	4.000	176.2	0.005675	93.38	1021.3	1114.7	64.2	957.2	1050.5	0.1738	1.7457	1.9195
126.00	1.992	4.057	173.9	0.005751	93.90	1021.1	1115.0	64.3	956.8	1050.7	0.1747	1.7436	1.9183
126.15	2.000	4.072	173.5	0.005760	94.00	1021.0	1115.0	64.3	956.7	1050.7	0.1748	1.7431	1.9180
126.39	2.009	4.100	172.3	0.005809	94.28	1020.9	1115.2	64.4	956.5	1050.8	0.1754	1.7421	1.9175
127.00	2.047	4.168	169.6	0.005900	94.89	1020.5	1115.4	64.4	956.1	1051.0	0.1764	1.7397	1.9161
127.28	2.058	4.200	168.2	0.005943	95.17	1020.3	1115.5	64.4	955.9	1051.1	0.1769	1.7384	1.9155
127.94	2.100	4.276	165.6	0.006043	95.83	1019.9	1115.7	64.4	955.5	1051.3	0.1780	1.7360	1.9140
128.00	2.103	4.282	165.3	0.006052	95.89	1019.9	1115.8	64.5	955.5	1051.3	0.1781	1.7358	1.9139
128.16	2.107	4.300	164.7	0.006077	96.05	1019.8	1115.9	64.5	955.4	1051.4	0.1784	1.7352	1.9136
129.00	2.160	4.399	161.1	0.006207	96.89	1019.4	1116.2	64.6	954.8	1051.7	0.1799	1.7318	1.9117
129.01	2.156	4.400	161.1	0.006209	96.90	1019.4	1116.2	64.6	954.8	1051.7	0.1799	1.7318	1.9117
129.68	2.200	4.479	158.4	0.006320	97.57	1019.0	1116.5	64.6	954.4	1051.9	0.1811	1.7291	1.9102
129.84	2.205	4.500	157.8	0.006344	97.73	1018.9	1116.6	64.7	954.2	1051.9	0.1813	1.7286	1.9099
130.00	2.219	4.520	157.1	0.006370	97.89	1018.8	1116.7	64.7	954.1	1052.0	0.1816	1.7279	1.9095
130.67	2.254	4.600	154.5	0.006480	98.56	1018.4	1117.0	64.8	953.6	1052.2	0.1827	1.7253	1.9080
131.00	2.279	4.640	153.2	0.006530	98.89	1018.2	1117.1	64.8	953.4	1052.2	0.1833	1.7240	1.9073
131.35	2.300	4.68	151.9	0.00659	99.24	1018.6	1117.2	64.8	953.2	1052.4	0.1839	1.7227	1.9066
131.50	2.303	4.70	151.3	0.00661	99.38	1018.9	1117.3	64.9	953.1	1052.2	0.1841	1.7221	1.9062
132.00	2.340	4.76	149.4	0.00669	99.88	1017.7	1117.5	64.9	952.8	1052.7	0.1849	1.7202	1.9051
132.31	2.352	4.80	148.3	0.00674	100.18	1017.5	1117.7	64.9	952.6	1052.8	0.1854	1.7191	1.9045
132.96	2.400	4.88	145.9	0.00685	100.84	1017.1	1117.9	65.0	952.1	1053.0	0.1865	1.7166	1.9031
133.00	2.403	4.89	145.8	0.00686	100.88	1017.1	1118.0	65.0	952.1	1053.0	0.1866	1.7164	1.9030
133.08	2.401	4.90	145.5	0.00687	100.96	1017.0	1118.0	65.0	952.0	1053.0	0.1867	1.7161	1.9028
133.85	2.450	5.00	142.7	0.00700	101.73	1016.6	1118.3	65.1	951.5	1053.3	0.1881	1.7130	1.9011
134.00	2.467	5.02	142.2	0.00703	101.88	1016.5	1118.4	65.1	951.4	1053.3	0.1883	1.7125	1.9008
134.49	2.500	5.09	140.5	0.00712	102.37	1016.3	1118.7	65.2	951.1	1053.4	0.1891	1.7106	1.8997
135.00	2.533	5.16	138.7	0.00721	102.88	1016.0	1118.8	65.2	950.8	1053.6	0.1900	1.7086	1.8986
136.00	2.600	5.29	135.4	0.00739	103.88	1015.4	1119.3	65.3	950.1	1054.0	0.1917	1.7048	1.8965
137.00	2.669	5.43	132.1	0.00757	104.87	1014.8	1119.7	65.4	949.4	1054.3	0.1933	1.7010	1.8943
137.44	2.700	5.50	130.6	0.00765	105.31	1014.6	1119.9	65.5	949.1	1054.4	0.1940	1.6993	1.8933
138.00	2.740	5.58	128.9	0.00776	105.87	1014.3	1120.1	65.5	948.8	1054.6	0.1950	1.6972	1.8922
138.83	2.800	5.70	126.3	0.00792	106.70	1013.8	1120.5	65.6	948.2	1054.9	0.1964	1.6941	1.8905
139.00	2.812	5.73	125.8	0.00795	106.87	1013.7	1120.6	65.6	948.1	1055.0	0.1967	1.6934	1.8901
140.00	2.885	5.88	122.8	0.00814	107.87	1013.1	1121.0	65.7	947.4	1055.3	0.1984	1.6896	1.8880
140.43	2.900	5.90	121.6	0.00821	108.30	1012.9	1121.2	65.8	947.1	1055.4	0.1991	1.6880	1.8871
140.80	2.940	6.00	120.5	0.00830	108.67	1012.7	1121.4	65.8	946.9	1055.5	0.1997	1.6864	1.8861
141.00	2.960	6.03	119.9	0.00834	108.87	1012.6	1121.4	65.8	946.8	1055.6	0.2000	1.6859	1.8859
141.52	3.000	6.11	118.5	0.00845	109.40	1012.3	1121.6	65.9	946.4	1055.8	0.2008	1.6840	1.8848
142.00	3.037	6.18	117.1	0.00854	109.87	1012.0	1121.8	65.9	946.1	1055.9	0.2017	1.6821	1.8838
142.81	3.100	6.31	114.8	0.00871	110.78	1011.5	1122.3	66.0	945.5	1056.2	0.2030	1.6791	1.8821
143.00	3.115	6.34	114.3	0.00875	110.87	1011.4	1122.3	66.0	945.4	1056.3	0.2033	1.6774	1.8817
144.00	3.195	6.51	111.6	0.00896	111.87	1010.8	1122.7	66.1	944.7	1056.6	0.2050	1.6746	1.8796
144.06	3.200	6.52	111.4	0.00909	111.93	1010.8	1122.7	66.1	944.7	1056.6	0.2051	1.6746	1.8797
145.00	3.277	6.67	109.0	0.00918	112.86	1010.3	1123.1	66.2	944.0	1056.9	0.2067	1.6709	1.8776
145.27	3.300	6.72	108.3	0.00924	113.13	1010.1	1123.2	66.3	943.8	1057.0	0.2071	1.6698	1.8769
146.00	3.361	6.84	106.5	0.00940	113.86	1009.7	1123.6	66.3	943.4	1057.2	0.2083	1.6672	1.8755
146.46	3.400	6.92	105.3	0.00950	114.32	1009.4	1123.7	66.3	943.1	1057.4	0.2091	1.6655	1.8746
146.89	3.430	7.00	104.3	0.00960	114.74	1009.2	1124.0	66.4	942.8	1057.6	0.2098	1.6639	1.8737
147.00	3.446	7.02	104.0	0.00962	114.86	1009.1	1124.0	66.4	942.7	1057.6	0.2100	1.6635	1.8735
147.62	3.500	7.12	102.5	0.00976	115.48	1008.8	1124.3	66.5	942.3	1057.8	0.2110	1.6606	1.8716
148.00	3.533	7.20	101.6	0.00985	115.86	1008.6	1124.4	66.6	942.0	1057.9	0.2116	1.6598	1.8714
148.75	3.600	7.33	99.8	0.01002	116.61	1008.2	1124.8	66.6	941.6	1058.1	0.2128	1.6571	1.8699
149.00	3.623	7.38	99.2	0.01008	116.86	1008.0	1124.8	66.6	941.4	1058.2	0.2132	1.6562	1.8694
149.85	3.700	7.53	97.2	0.01028	117.71	1007.5	1125.2	66.7	940.8	1058.5	0.2146	1.6531	1.8677
150.00	3.714	7.57	96.9	0.01032	117.86	1007.4	1125.3	66.7	940.7	1058.5	0.2149	1.6525	1.8674
150.91	3.800	7.73	94.9	0.01051	118.77	1006.9	1125.7	66.8	940.1	1058.8	0.2164	1.6492	1.8656
151.00	3.809	7.76	94.7	0.01056	118.86	1006.8	1125.7	66.8	940.0	1058.8	0.2165	1.6488	1.8653
151.79	3.900	7.93	93.0	0.01075	119.65	1006.3	1126.0	66.9	939.5	1059.1	0.2178	1.6460	1.8638
152.00	3.902	7.95	92.6	0.01080	119.86	1006.2	1126.1	66.9	939.3	1059.2	0.2182	1.6452	1.8634
152.27	3.920	8.00	92.0	0.01087	120.12	1006.1	1126.2	67.0	939.1	1059.3	0.2186	1.6443	1.8629
153.00	3.999	8.14	90.5	0.01105	120.86	1005.7	1126.5	67.1	938.6	1059.5	0.2198	1.6416	1.8614
153.01	4.000	8.14	90.5	0.01107	120.99	1005.7	1126.5	67.1	938.6	1059.5	0.2198	1.6416	1.8614
154.00	4.098	8.34	88.4	0.01131	121.86	1005.1	1127.0	67.1	938.0	1059.8	0.2214	1.6380	1.8594
154.02	4.100	8.35	88.4	0.01131	121.86	1005.1	1127.0	67.2	938.0	1059.8	0.2214	1.6380	1.8594
155.00	4.199	8.55	86.4	0.01157	122.86	1004.5	1127.4	67.2	937.3	1060.2	0.2231	1.6341	1.8575
155.02	4.200	8.55	86.4	0.01157	122.86	1004.5	1127.4	67.2	937.3	1060.2	0.2231	1.6344	1.8575

TABLE OF THE PROPERTIES OF DRY AND SATURATED STEAM AT PRESSURES BELOW THAT OF THE ATMOSPHERE—By Lionel S. Marks—(Concluded)

Temp. Fahr.	Pressure, lbs. per sq. in.	Pressure, inches of Mercury (at 32° F.)	Sp. vol., cu. ft. per lb.	Density, lbs. per cu. ft.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	External Work of Evaporation in B.t.u.	Internal energy of Evaporation	Internal energy of Steam	Entropy of Water	Entropy of Evaporation	Entropy of Steam
t	p		v	1/v	h	L	H		I	E	n	L/T	N
155.97	4.300	8.75	84.6	0.01183	123.83	1004.0	1127.8	67.3	936.6	1060.5	0.2246	1.6309	1.8556
156.00	4.303	8.76	84.5	0.01184	123.86	1003.9	1127.8	67.3	936.6	1060.5	0.2247	1.6308	1.8555
156.93	4.400	8.96	82.7	0.01209	124.79	1003.4	1128.2	67.4	936.0	1060.8	0.2262	1.6275	1.8537
157.00	4.408	8.98	82.6	0.01211	124.86	1003.4	1128.2	67.4	935.9	1060.8	0.2263	1.6272	1.8535
157.10	4.410	9.00	82.4	0.01213	124.95	1003.3	1128.2	67.5	935.8	1060.8	0.2264	1.6269	1.8533
157.86	4.500	9.16	81.0	0.01234	125.72	1002.8	1128.5	67.5	935.3	1061.1	0.2277	1.6241	1.8518
158.00	4.515	9.20	80.7	0.01239	125.86	1002.8	1128.6	67.5	935.3	1061.1	0.2279	1.6236	1.8515
158.77	4.600	9.36	79.3	0.01260	126.63	1002.4	1129.0	67.6	934.8	1061.4	0.2291	1.6209	1.8500
159.09	4.625	9.42	78.9	0.01267	126.86	1002.2	1129.1	67.6	934.6	1061.5	0.2295	1.6201	1.8496
159.67	4.700	9.57	77.8	0.01286	127.53	1001.9	1129.4	67.7	934.1	1061.7	0.2306	1.6177	1.8483
160.00	4.737	9.67	77.2	0.01296	127.86	1001.6	1129.5	67.7	933.9	1061.8	0.2311	1.6165	1.8476
160.55	4.800	9.77	76.3	0.01312	128.41	1001.3	1129.7	67.8	933.5	1061.96	0.2320	1.6146	1.8466
161.00	4.851	9.88	75.5	0.01325	128.86	1001.1	1129.9	67.8	933.3	1062.1	0.2327	1.6130	1.8457
161.42	4.900	10.00	74.8	0.01340	129.28	1000.8	1130.1	67.8	933.0	1062.2	0.2334	1.6115	1.8449
161.50	4.990	10.06	74.7	0.01340	129.36	1000.8	1130.2	67.9	933.1	1062.3	0.2335	1.6112	1.8447
162.00	4.967	10.12	73.8	0.01355	129.86	1000.5	1130.4	67.9	932.6	1062.4	0.2344	1.6094	1.8438
162.28	5.000	10.18	73.3	0.01364	130.10	1000.3	1130.5	68.0	932.4	1062.5	0.2348	1.6084	1.8432
163.00	5.086	10.36	72.2	0.01386	130.86	999.9	1130.8	68.1	931.9	1062.8	0.2360	1.6059	1.8419
164.00	5.208	10.61	70.6	0.01417	131.86	999.3	1131.2	68.1	931.2	1063.1	0.2376	1.6024	1.8400
165.00	5.333	10.86	69.1	0.01448	132.86	998.7	1131.6	68.2	930.5	1063.4	0.2391	1.5989	1.8380
165.54	5.390	11.00	68.3	0.01465	133.4	998.4	1131.8	68.3	930.1	1063.5	0.2400	1.5970	1.8370
166.00	5.460	11.12	67.6	0.01480	133.86	998.2	1132.0	68.4	929.8	1063.7	0.2407	1.5954	1.8361
167.00	5.589	11.38	66.1	0.01513	134.86	997.6	1132.4	68.5	929.1	1064.0	0.2423	1.5919	1.8342
168.00	5.721	11.65	64.7	0.01546	135.86	997.0	1132.8	68.6	928.4	1064.3	0.2439	1.5885	1.8324
169.00	5.855	11.92	63.3	0.01580	136.86	996.4	1133.3	68.6	927.8	1064.6	0.2455	1.5850	1.8305
169.29	5.880	12.00	62.9	0.01596	137.15	996.3	1133.4	68.7	927.6	1064.7	0.2459	1.5841	1.8300
170.00	5.992	12.20	62.0	0.01614	137.87	995.8	1133.7	68.7	927.1	1064.9	0.2470	1.5816	1.8286
170.06	6.000	12.22	61.9	0.01616	137.96	995.8	1133.7	68.8	927.0	1064.9	0.2471	1.5814	1.8285
171.00	6.131	12.48	60.7	0.01649	138.87	995.2	1134.1	68.8	926.4	1065.2	0.2486	1.5782	1.8268
172.00	6.273	12.77	59.4	0.01685	139.87	994.6	1134.5	68.9	925.7	1065.6	0.2502	1.5747	1.8249
172.67	6.370	13.00	58.5	0.01709	140.56	994.2	1134.8	69.0	925.2	1065.8	0.2513	1.5724	1.8237
173.00	6.417	13.07	58.1	0.01721	140.87	994.0	1134.9	69.0	925.0	1065.9	0.2518	1.5713	1.8231
174.00	6.564	13.37	56.9	0.01758	141.87	993.5	1135.3	69.1	924.3	1066.2	0.2534	1.5679	1.8213
175.00	6.714	13.67	55.7	0.01796	142.87	992.9	1135.7	69.2	923.6	1066.5	0.2550	1.5645	1.8195
176.00	6.867	13.98	54.5	0.01834	143.87	992.3	1136.2	69.3	923.0	1066.8	0.2565	1.5611	1.8176
176.06	6.860	14.09	54.4	0.01836	143.88	992.3	1136.2	69.3	923.0	1066.8	0.2566	1.5609	1.8175
176.85	7.000	14.25	53.6	0.01867	144.70	991.8	1136.5	69.4	922.4	1067.1	0.2579	1.5582	1.8161
177.09	7.023	14.30	53.4	0.01873	144.88	991.7	1136.6	69.4	922.3	1067.1	0.2581	1.5577	1.8158
178.09	7.182	14.62	52.3	0.01912	145.88	991.1	1137.0	69.5	921.6	1067.5	0.2597	1.5543	1.8140
179.09	7.344	14.95	51.2	0.01953	146.88	990.5	1137.4	69.6	920.9	1067.8	0.2612	1.5510	1.8122
179.15	7.350	15.00	51.0	0.01959	147.03	990.5	1137.5	69.7	920.8	1067.8	0.2614	1.5505	1.8119
180.00	7.51	15.29	50.2	0.01994	147.88	989.9	1137.8	69.7	920.2	1068.1	0.2628	1.5476	1.8104
181.00	7.68	15.63	49.1	0.02036	148.88	989.3	1138.2	69.8	919.5	1068.4	0.2644	1.5443	1.8087
182.00	7.85	15.98	48.1	0.02078	149.89	988.7	1138.6	69.9	918.8	1068.7	0.2659	1.5409	1.8068
182.05	7.84	16.00	48.1	0.02080	149.94	988.7	1138.6	69.9	918.8	1068.7	0.2660	1.5407	1.8067
182.86	8.00	16.29	47.3	0.02115	150.80	988.2	1139.0	70.0	918.2	1069.0	0.2673	1.5380	1.8054
183.00	8.02	16.34	47.2	0.02121	150.89	988.1	1139.0	70.0	918.1	1069.0	0.2675	1.5376	1.8051
184.00	8.20	16.70	46.2	0.02165	151.89	987.5	1139.4	70.1	917.4	1069.3	0.2690	1.5343	1.8033
184.80	8.33	17.00	45.4	0.02201	152.70	987.0	1139.7	70.2	916.8	1069.5	0.2703	1.5316	1.8019
185.00	8.38	17.07	45.3	0.02210	152.89	986.9	1139.8	70.2	916.7	1069.6	0.2706	1.5310	1.8016
186.00	8.57	17.45	44.3	0.02255	153.89	986.3	1140.2	70.3	916.0	1069.9	0.2721	1.5277	1.7998
187.00	8.76	17.83	43.5	0.02301	154.90	985.7	1140.6	70.4	915.3	1070.2	0.2737	1.5244	1.7981
187.44	8.82	18.00	43.1	0.02321	155.34	985.5	1140.8	70.5	915.0	1070.3	0.2744	1.5229	1.7973
188.00	8.95	18.22	42.6	0.02348	155.90	985.1	1141.0	70.5	914.6	1070.5	0.2752	1.5211	1.7963
188.27	9.00	18.32	42.4	0.02361	156.20	985.0	1141.1	70.6	914.4	1070.5	0.2756	1.5202	1.7958
189.00	9.14	18.61	41.74	0.02396	156.90	984.5	1141.4	70.6	913.9	1070.8	0.2767	1.5179	1.7946
189.95	9.31	19.00	40.94	0.02438	157.87	983.9	1141.8	70.6	913.3	1071.1	0.2782	1.5148	1.7930
190.00	9.34	19.02	40.91	0.02444	157.91	983.9	1141.8	70.7	913.2	1071.1	0.2783	1.5146	1.7929
191.00	9.54	19.43	40.10	0.02493	158.91	983.3	1142.2	70.8	912.5	1071.4	0.2798	1.5113	1.7911
192.00	9.74	19.83	39.31	0.02544	159.91	982.7	1142.6	70.9	911.8	1071.7	0.2813	1.5081	1.7894
192.39	9.80	20.00	39.01	0.02553	160.30	982.5	1142.8	71.0	911.5	1071.8	0.2819	1.5069	1.7888
193.00	9.95	20.27	38.54	0.02595	160.91	982.1	1143.0	71.0	911.1	1072.0	0.2829	1.5048	1.7877
193.22	10.00	20.36	38.38	0.02606	161.10	982.0	1143.1	71.1	910.9	1072.0	0.2832	1.5042	1.7874
194.00	10.17	20.71	37.78	0.02647	161.92	981.5	1143.4	71.1	910.4	1072.3	0.2845	1.5016	1.7861
194.66	10.29	21.00	37.29	0.02682	162.58	981.1	1143.7	71.2	909.8	1072.5	0.2856	1.4995	1.7850
195.00	10.39	21.15	37.04	0.02700	162.92	980.9	1143.8	71.2	909.7	1072.6	0.2860	1.4984	1.7844
196.00	10.61	21.60	36.32	0.02753	163.92	980.3	1144.2	71.3	909.0	1072.9	0.2876	1.4952	1.7828
196.89	10.78	22.00	35.70	0.02801	164.82	979.8	1144.6	71.4	908.4	1073.2	0.2889	1.4924	1.7813
197.00	10.83	22.05	35.62	0.02807	164.93	979.7	1144.6	71.4	908.3	1073.2	0.2891	1.4920	1.7811
197.75	11.00	22.40	35.10	0.02849	165.70	979.2	1144.9	71.4	907.8	1073.4	0.2902	1.4895	1.7797
198.00	11.06	22.52	34.93	0.02863	165.93	979.1	1145.0	71.5	907.6	1073.5	0.2906	1.4888	1.7794
199.00	11.26	22.99	34.26	0.02919	166.94	978.4	1145.4	71.5	906.9	1073.8	0.2921	1.4856	1.7777
199.02	11.27	23.00	34.25	0.02920	166.96	978.4	1145.4	71.6	906.9	1073.8	0.2921	1.4856	1.7777
200.00	11.52	23.47	33.60	0.02976	167.91	977.8	1145.8	71.6	906.2	1074.1	0.2937	1.4824	1.7761
201.00	11.76	23.95	32.96	0.03034	168.94	977.2	1146.2	71.7	905.5	1074.4	0.2952	1.4792	1.7744
201.10	11.76	24.00	32.90	0.03040	169.04	977.2	1146.2	71.7	905.4	1074.5	0.2954	1.4788	1.7742
201.96	12.00	24.43	32.36	0.03090	169.90	976.6	1146.5	71.8	904.8	1074.7	0.2967	1.4760	1.7727
202.00	12.01	24.45	32.33	0.03093	169.95	976.6	1146.6	71.9	904.8	1074.7	0.2967	1.4760	1.7727
203.00	12.26	24.96	31.72	0.03153	170.95	976.0	1146.9	72.0	904.1	1075.0	0.2982	1.4729	1.7711
203.08	12.25	25.00	31.67	0.03158	171.03	975.9	1146.9	72.0	903.9	1075.0	0.2983	1.4727	1.7710
204.00	12.51	25.48	31.12	0.03214	171.96	975.4	1147.3	72.1	903.3	1075.3	0.2997	1.4697	1.7694
205.00	12.77	26.00	30.53	0.03276	172.96	974.7	1147.7	72.2	902.6	1075.6	0.3012	1.4666	1.7678
205.87	13.00	26.47	30.03	0.03330	173.80	974.2	1148.0	72.2	902.0	1075.8	0.3025	1.4639	1.7664
206.00	13.03	26.53	29.95	0.03339	173.97	974.1	1148.1	72.3	901.9	1075.9	0.3027	1.4635	1.7662
206.85	13.23	27.00	29.47	0.03393	174.83	973.6	1148.4	72.3	901.3	1076.1	0.3040	1.4607	1.7647
207.00	13.30	27.08	29.39	0.03402	174.97	973.5	1148.5	72.3	901.2	1076.1	0.3042	1.4603	1.7645
208.00	13.57	27.63	28.85	0.03466	175.98	972.9	1148.9	72.4	900.4	1076.4	0.3057	1.4572	1.7629
208.65	13.72	28.00	28.50	0.03509	176.64	972.4	1149.0	72.5	899.9	1076.6	0.3066	1.4548	1.7614
209.00	13.85	28.19	28.32	0.03531	176.98	982.2	1149.2	72.5	899.7	1076.7	0.3072	1.4541	1.7613
209.55	14.00	28.50	28.02	0.03569	177.50	971.9	1149.3	72.6	899.3	1076.8	0.3081	1.4523	1.7604
210.00	14.13	28.76	27.80	0.03597	177.99	971.6	1149.6	72.6	899.0	1076.9	0.3087	1.4510	1.7597
210.42	14.21	29.00	27.59	0.03625	178.58	971.2	1149.8	72.7	898.7	1077.0	0.3094	1.4499	1.7593
211.00	14.41	29.33	27.29	0.03664	178.99	971.0	1150.0	72.8	898.3	1077.2	0.3103	1.4478	1.7581
212.00	14.70	29.92	26.79	0.03732	180.00	970.4	1150.4	72.8	897.6	1077.5	0.3118	1.4445	1.7563
212.13	14.70	30.00	26.78	0.03811	180.13	970.3	1150.4	72.8	897.7	1077.5	0.3120	1.4443	1.7563
213.00	14.99	30.52	26.30	0.03802	181.0	969.7	1150.7	72.9	896.8	1077.8	0.3133	1.4416	1.7549
213.03	15.00	30.54	26.27	0.03806	181.0	969.7	1150.7	72.9	896.8	1077.8	0.3133	1.4416	1.7549

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Catechism of Electricity

MANAGEMENT OF ELECTRIC STATIONS—II

1332. *Is electric-station work a dangerous occupation on account of the possibility of receiving shocks?*

It is dangerous work for careless men, especially where the current is generated at high voltage. Because a wire or busbar carrying current appears the same as it would were it not carrying current, the workers must rely entirely upon their knowledge of the circuit conditions. Ordinarily, it is not difficult to trace out a circuit before working upon it, to determine whether it is connected to a live source.

1333. *Is the human body a good conductor of electricity?*

It is a poor conductor; under the most favorable conditions it offers a resistance of about 2500 ohms. This is the resistance of 300 miles of ordinary telegraph wire. The chief resistance is that of the skin which, when dry, is about 100,000 ohms.

1334. *What voltage is considered fatal?*

Less than one ampere (about 0.8 ampere) of current flowing through the body will produce death, although less than this is often fatal. With the resistance of the body equal to 2500 ohms, the voltage corresponding to the fatal current is, according to Ohm's law,

$$0.8 \times 2500 = 2000 \text{ volts}$$

However, people are frequently killed by much less than this.

1335. *What voltage is considered dangerous?*

As before stated, the skin of the body offers a high resistance and this is particularly true of the hardened skin about the hands, where accidental contact is usually made. Because of this, shocks from pressures below 200 volts need be avoided only on account of the unpleasant sensations. With pressures above 200 volts, care should be exercised to guard against accidents and particularly so in the case of alternating current.

1336. *Have not persons lived after receiving 100,000 volts or more?*

Yes; voltages as high as 100,000, which are confined to alternating currents of high frequency, often produce what is known as a "skin effect," that is, the currents do not penetrate the body to any extent but pass over its surface. Therefore very high voltages are less frequently fatal than moderately high ones.

1337. *What is the best safeguard against dangerous or fatal shocks?*

The best safeguard is to make all connections before the current is passed through the circuit. This may be accomplished by introducing a switch in series with each of the main supply wires and the rest of the circuit; and leaving these main switches open until all the other connections are made.

1338. *If the safeguard just mentioned is not practicable, what are the next best precautions?*

When working on live circuits, never place the body so that it can come in series with the circuit or in contact with a part of the circuit at a potential different from that of the wire being held. To prevent this, stand on a dry piece of wood or other insulator and work with one hand as much as possible, keeping the other on some insulating support or in the pocket.

In the case of alternating currents having pressures above 1000 volts, too much care cannot be taken as the leakage is often sufficient to cause a dangerous shock, even though direct contact is not made with the charged parts. Insulated wire often fails to give protection, and it should not be touched unless one stands on a well insulated support and wears rubber gloves, rubber shoes, or both. It is always well to use tools having insulated handles.

1339. *What are the chances of successfully reviving victims of electrical shocks?*

Prompt and continued efforts along the proper lines have proved successful. In order that the task may not be undertaken in a half-hearted manner, it is well to bear in mind that accidental shocks seldom result in death unless the victim is left unaided for too long a time, or efforts at reviving are stopped too soon. In the majority of instances, owing to short and imperfect contact with the conductors, the shock merely temporarily paralyzes the nerves controlling the muscles of respiration.

1340. *What should first be done to revive a victim of electrical shock?*

The body must be removed at once from the circuit by breaking contact with the conductors. This may be done by using a dry stick of wood with which to roll the body over to one side, or to brush away the live wire. In the absence of a stick any dry piece of clothing may be used in seizing the victim. If in contact with the earth, the coat tails of the victim, or any loose piece of clothing may be seized to draw the body away from the conductors.

1341. *After the body has been removed from the circuit, what should be done?*

There are two methods in use for resuscitating the victim; the older and more generally known is the Silvester method. In this the body must be turned upon the back, the collar and clothing about the neck loosened and a coat rolled up and placed under the shoulders, so as to throw the head back. It is also well to feel in the victim's mouth and remove any foreign body, such as tobacco, false teeth, etc. Having done this, efforts should be made to force the victim to breathe, just as would be done in a case of drowning.

To accomplish this, the operator should kneel at the victim's head, facing him, as in Fig. 431, and seizing both arms, draw

FIG. 431

them to their full length over the head, holding them there for two or three seconds only. This is to expand the chest and allow air to enter the lungs. Then the operator should carry the arms down to the sides and front, firmly pressing them on the walls of the chest, as in Fig. 432, thus driving the air out of the lungs.

FIG. 432

1342. *How often should this arm movement be repeated?*

At least sixteen times per minute for an hour or more, or until natural respiration is established.

1343. *Can anything further be done to revive the victim?*

It is advisable to grasp the tongue with a handkerchief or piece of cloth, to prevent its slipping, and draw it forcibly out when the arms are being extended above

the head, as in Fig. 431, allowing it to recede again when the chest is being compressed.

1344. *What is the purpose of the tongue movement?*

It frees the throat so as to permit air to enter the lungs, and also, by irritation from contact of the under part of the tongue against the lower teeth, it often causes an involuntary effort to breathe.

1345. *What other method may be employed to resuscitate the victim?*

The Schaefer or "prone pressure" method has recently been recommended by a joint committee of the American Medical Association, the American Institute of Electrical Engineers and the National Electric Light Association.*

In this the subject, after being freed from contact with the conductor, is immediately placed upon his abdomen, as in Fig. 433, and pressure is applied rhythmically on the loins and lower ribs. To facilitate this the operator kneels,

FIG. 433

straddling the victim's thighs and rests his palms upon the latter's loins with the fingers spread over the lower ribs. With the arms held straight he swings forward slowly (Fig. 434) so that the weight of his body is gradually brought to bear upon the victim. This should take from two to three seconds and tends to com-

FIG. 434

press the abdomen, raise the diaphragm and drive out air from the lungs.

Next, he swings back so as to relieve the pressure and the lungs take in air. This should be repeated 12 to 15 times a minute until natural breathing has been started and then send for a physician.

1346. *Is the latter method better than the first?*

It has the advantage of being simpler and it is free from the trouble caused by the tongue falling back and blocking the air passages.

1347. *Would stimulants help in reviving the victim?*

They would have to be poured down the victim's throat and this would temporarily interfere with his breathing; hence they should not be given until the subject is fully conscious.

1348. *Who is responsible for the supervision and control of an electrical station?*

The superintendent or manager.

1349. *What qualifications must a station employee possess in order to succeed as the superintendent or manager?*

He must have an intimate knowledge of local conditions and requirements, have practical experience in the running of generators, motors, transformers, etc., and understand their principles of operation. He must also possess executive and business ability and have strict integrity.

1350. *What is one of the first problems that often confronts the new superintendent?*

He is frequently given an electrical station equipped with old apparatus and machinery and is confronted with the problem of increasing the net earnings without making any radical changes in the equipment.

1351. *How can he best accomplish this?*

Inasmuch as thousands of dollars have probably been invested in the installation, it is usually impracticable to scrap the old apparatus and machinery at the start and purchase an entirely new equipment. The new superintendent should therefore be content temporarily with the existing apparatus, but he should investigate the situation thoroughly so that when money is available he will be able to make the most necessary additions and improvements to the plant.

1352. *What rules should serve as a guide in making the additions and improvements?*

Make the station conform to modern practice so as to reduce as much as possible the cost of production.

1353. *How should the recommendations be put before the directors?*

They should be formed into a clear statement, covering the best kind of apparatus to install, the cost, and the probable profit resulting through their installation.

1354. *If the suggestions are approved, and, when put into effect, result in an increase of business, what should be the next move?*

The money resulting from the increased business should be applied to further alterations and extensions of the plant. Gradually, the transformation of the station will thus be brought about, and when satisfactory conditions are once secured, every effort should be made toward the maintenance.

1355. *What are the best methods of obtaining good men and retaining their services?*

Showing appreciation of their work by paying them what they are worth and advancing them to higher positions when vacancies occur or new situations open up.

1356. *How should the superintendent be governed in issuing orders to his subordinates?*

The orders must be worded plainly and should be of such a nature that they may be enforced to the letter. Under no conditions must favoritism be shown, and social or political influence should not be considered in the proper enforcement of the orders and regulations of the station.

1357. *What amount of technical training should a man have in order to become a competent station superintendent?*

He should preferably have three or four years' study and experimental work in electrical engineering at a technical school. If he has a persistent, studious nature and is naturally inclined toward the subject, he may obtain the theoretical training by home study.

1358. *What amount of practical training should a man have to become a competent station superintendent?*

The technical training should be supplemented with several years' practical training. A number of our large electrical manufacturing concerns have also regular channels through which one possessing a technical education and being fortunate enough to enter, can secure much valuable experience in the design, construction, testing and operation of the various classes of electrical machines.

1359. *How is the actual worth of a station superintendent to his company usually judged?*

From a profit-making basis.

1360. *Give some general rules for increasing the net profits.*

Reduce the operation and maintenance expenses to a minimum. After reducing the former to a kilowatt-hour basis compare them with corresponding expenses in similar stations, and determine in which direction efforts should be made toward reduction. To reduce the cost of maintenance, have frequent inspections made of all machines and appliances, and whenever a defect is first noted have it attended to at once.

1361. *What privileges should be given the station superintendent by his employers?*

Inasmuch as the superintendent is held directly responsible for the results, he should be given the privilege of putting into effect whatever methods he deems advisable. His methods should not be disputed or confidence lost in his ability until positive proof of failure is at hand. In the meanwhile, they should endow him with full power so that his subordinates will regard him as the one man in authority whose orders are not to be questioned.

*The National Electric Light Association has recently got out a pamphlet and chart covering this method, which it intends to circulate largely.

Care and Operation of Transformers*

By W. M. McConahey

Formerly it was customary to ship transformers packed separately from their tanks. When thus shipped it was practically impossible to prevent the insulation from absorbing moisture, even when special methods of packing were resorted to. Such schemes as packing in tin-lined cases or placing calcium-chloride inside of packing cases never proved very satisfactory, so that before putting a transformer into service it was nearly always necessary to dry it out. This required time and care and a certain degree of expert knowledge which made it very desirable that some method of shipment be devised which would make drying out before installing unnecessary. This is now accomplished by bracing the transformer securely in its tank, filling the tank with oil and shipping in this manner. The only limit imposed upon this method is the inability of the railroads to handle very large sizes because of insufficient clearances. This has been overcome to a considerable extent, however, by the use of a special form of car with the floor much lower than that of an ordinary flat car.

When a transformer is shipped in its own tank with the oil it should arrive at its destination with the insulation and oil practically dry and free from moisture. To make sure of this, however, it is best to draw a few samples of oil from the bottom of the tank and test them. The best way to make the test is to use some standard form of testing cup and a variable-ratio testing transformer by means of which the voltage can be raised gradually. The cup is filled with a sample of the oil, the gap is adjusted to a fixed length with a micrometer screw and the voltage is raised until a spark jumps across the gap through the oil. It is customary to set the gap at about 150 mils, and with such a setting good oil will show a breakdown at not less than 30,000 volts. If no means of varying the voltage is at hand, so that it is necessary to use a fixed voltage for making the test, the gap should be set to give 200 volts for every mil of separation and breakdown should not occur at this setting. However, it is not advisable to use a testing voltage below 6600 volts.

It is important for the life of the transformer, particularly if it be of high voltage, that the tank be closed tightly so as entirely to exclude dust and dirt. This is done by packing tightly around all outlet bushings and placing gaskets under the cover, and also under handhole and manhole covers.

After the transformer has been installed and put in operation, it should not be expected to operate continuously without further attention. A periodical inspection should be made about once every six months. This should include a test of a sample of the oil drawn from the bottom of the tank and an examination of the interior. If the oil shows the presence of too much moisture, or an inspection shows a deposit of dirt or slime in the transformer, the oil should be drawn off and filtered and dehydrated, and the transformer and the interior of the tank should be thoroughly cleaned with dry clean waste. The ventilating ducts can generally be cleaned by squirting dry oil through them. If the transformer is water-cooled the cooling coils should be examined to find out whether any deposit has been formed on them by the oil. Some oils will form a coating on the cooling coils which acts as a heat insulator and causes the transformer to run hot. This deposit can be removed only by scraping and rubbing the coils until they are clean.

If the oil in a transformer is in a dirty condition or forms a deposit, it should be filtered, and if it contains moisture, steps should be taken at once to remove this moisture. The filtering process can be performed easily by passing the oil through two or three thicknesses of bolting cloth.

The removal of moisture is not such an easy task. Various makeshift schemes have been tried from time to time but all of them are far from satisfactory. Among these schemes are placing bags of dry lime in the oil, passing hot air through the oil, or heating the oil so as to vaporize the moisture and drive it off. To remove the moisture completely and leave the oil in first-class condition, it should be passd through some standard form of dehydrator. Probably the best form of this class of apparatus is the filter-press type, in which the oil is pumped through several thicknesses of a specially prepared paper. The paper has the faculty of allowing the oil to pass through freely, while at the same time retaining the moisture until it becomes saturated. When the paper approaches saturation it should be removed and replaced by fresh dry paper.

A simple and satisfactory way of keeping the oil clean and dry is to so connect a small dehydrator outfit to a transformer tank that the oil can be pumped from the bottom of the tank through the dehydrator and back again into the tank at the top. The connection at the bottom can be made through the oil valve provided with all transformers. The necessary connections can be made and the dehydrator put into service without in any way interfering with the operation of the transformer.

Assuming that the necessary care is taken to keep the transformer clean and dry, it is equally important that it be not overloaded to such an extent that the insulation is damaged by excessive heating of the coils. Insulation will not stand prolonged heating at a temperature much above 100 deg. C. without permanent injury.

It is practically out of the question to determine the actual temperature of the copper, which is the hottest part, so that about the only guide the operating man has is the temperature of the oil. It is customary for manufacturers to guarantee that the temperature rise of the windings of power transformers at normal load will not exceed 40 deg. C. The standard temperature guarantee for power transformers when operating continuously at 25 per cent. overload is 55 deg. C.

If the windings are well ventilated the maximum temperature will not exceed the average by more than a small amount, but if they are poorly ventilated there may be a wide difference between the maximum and average temperatures. As the load is increased the temperature drop from the copper to the oil increases rapidly, so that for heavy overloads the temperature of the copper is much above that of the oil. For this reason, if it is necessary to put a heavy overload on a transformer for several hours, some judgment must be exercised in trying to determine from the temperature of the oil what is the temperature of the windings. If this temperature (not temperature rise) is allowed to get much beyond 100 deg. C., there is danger of the insulation being permanently injured.

Self-cooling transformers can now be built in sizes up to about 2000 or 3000 kw., depending upon the voltage and frequency. The output for which water-cooled transformers can be built is limited only by the weight and the mechanical dimensions which make handling or transportation difficult. Ordinarily the water rate is ¼ gal. per minute per kilowatt-loss, which gives a temperature rise of 15 deg. C. from the inlet to the outlet of the cooling coil.

Some water will form a deposit on the inside of the cooling coil which will act more or less as a heat insulator, and may in extreme cases close up the coils entirely. The presence of this deposit can be detected by a decreased flow and an increased temperature rise of the water or by an increase in the temperature of the transformer. It can generally be removed by passing a solution of hydrochloric acid through the cooling coils.

A water-cooled transformer should not be operated except for a very short time without the circulation of water through the cooling coils. The transformer will heat up very rapidly and will soon reach a dangerous temperature. It cannot even be operated in this manner for any length of time without load, because the iron loss in itself is sufficient to raise the temperature above the danger point. To guarantee against overheating due to the

*From a paper delivered before the Seattle convention of the National Electric Light Association, June 10-14.

stopping of the flow of water, it is customary to equip water-cooled transformers with alarm contact thermometers, to which alarm bells can be wired to give warning if the temperature should rise too high.

With air-blast transformers about 150 cu.ft. of air per minute is required per kilowatt of loss. The air pressure required varies from about ½ to 1 oz., depending upon the size of the transformer. If the air pressure is shut off from an air-blast transformer it can carry full load for only a few minutes before reaching a dangerous temperature. This will be understood when it is remembered that there is no oil to store heat, and since there is little radiation, practically all the heat generated in the windings has to be stored in the copper. The thermal capacity of copper is very low, and therefore it heats up rapidly.

Starting cold and throwing on full load, with the air blast going, an air-blast transformer will reach steady temperature conditions in about two hours, while an oil-cooled transformer would require probably six to ten times as long. Because of the rapid heating up of the windings, an air-blast transformer cannot safely carry a heavy overload for more than a few minutes. The temperature rise of the windings under normal operation varies approximately in the same ratio as the copper loss.

CORRESPONDENCE

Circuit Breaker Alarm

Considerable trouble was experienced through the circuit-breakers coming out and as there was much noise around the plant the station attendant was often unaware that a breaker had gone out until the telephone rang and someone would inquire what the trouble was with the power. After being called up several times he concluded to attach an alarm, so accordingly wired up a small bell and a couple of dry batteries as per diagram. The circuit-breakers were each on a separate railway circuit and took their current from the same busbar.

The attendant took off the two clips at *A*, between which the arms of the breakers rest when out, and insulated one side by putting a piece of mica between the clip and the body of the breaker. He also drilled out the holes and bushed them, so that one clip was insulated from the other and when the arm of the breaker came out it rested between the two clips and closed the circuit so the bell would ring. As there was only one bell he thought by connecting it as shown in the sketch it would be all right and would ring when either breaker came out.

The first time breaker No. 1 came out the bell rang all right. The attendant pulled the switch below the breaker and then put the breaker in. Before he closed the switch, however, there was a report and smoke began to roll up back of the board. Upon investigation he found that the wire from the left side of No. 2 breaker was burned off at *X*. It developed that when closing the breaker he also closed the circuit, permitting the current from breaker No. 2 to flow through the batteries and out on the line through No. 1 breaker, but as the wire was not large enough to carry the current, it promptly melted. After procuring another bell and connecting the breakers separately he had an alarm on both breakers and they worked all right.

LEON L. POLLARD.

Fairfield, Maine.

Burnt Commutator Bars

In answer to Mr. Kramer's inquiry in the May 21 issue, I would say that burnt spots on the commutator are caused by loose leads, poor connections between brushes and studs, poor contact of the brushes on the commutator and high mica. Although he says the mica is not high, still from his having repeatedly to sandpaper the burnt spots it would appear that the mica is high, although not all over, perhaps. If not due to the causes mentioned, I would suggest that Mr. Kramer try cutting the mica down ⅛ in., using for the purpose a sharp, narrow chisel.

HERBERT HILL.

Middle Falls, N. Y.

It appears that Mr. Kramer's trouble is caused by a short-circuit or ground due to oil soaking into the mica and carbonizing it or to arcing due to carbon and copper dust. The arc will eat away the mica and pit the bars on the under side, heating them enough to change their color.

I have had practically the same experience with three different machines and in each case found that the discolored bars were grounded. After putting new mica insulation between the bars and also under them the ground disappeared and we had no further trouble with the bars becoming discolored.

WILLIAM T. GARLITZ.

McKees Rocks, Penn.

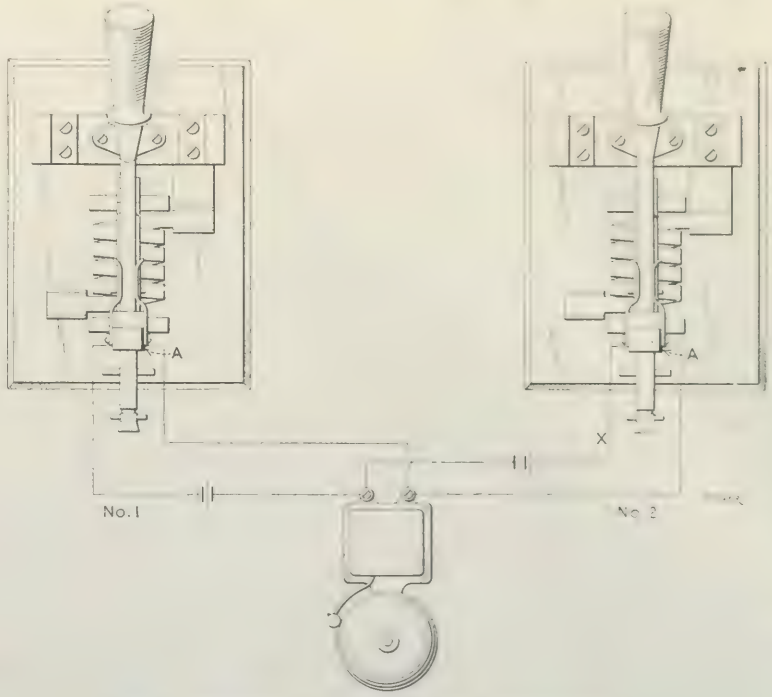
Referring to the inquiry by Mr. Kramer in the May 21 issue, I have a small 125-volt machine which caused considerable trouble by the commutator becoming black and appearing to be burnt. I cleaned it with sandpaper and in five minutes it would be as black as ever. Therefore, I concluded that the brushes had become soft, so put on a new set but without any better results and in three weeks' time they were all gone.

Upon looking over the machine I found the pulley out of round about ⅜ in. and also a high spot on the commutator about ⅜₂ in. I then turned down the commutator and put on a new pulley, also a new set of brushes and have since run the machine 3½ months without a stop. The commutator has not been sandpapered and is as bright as when new. The brushes are also in excellent condition.

I never use any kind of oil on the commutator, but instead use paraffin because it is dry and no dust will collect on the brushes.

G. LAWTON.

Bantam, Conn.



BELL CONNECTED TO BOTH CIRCUIT BREAKERS

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

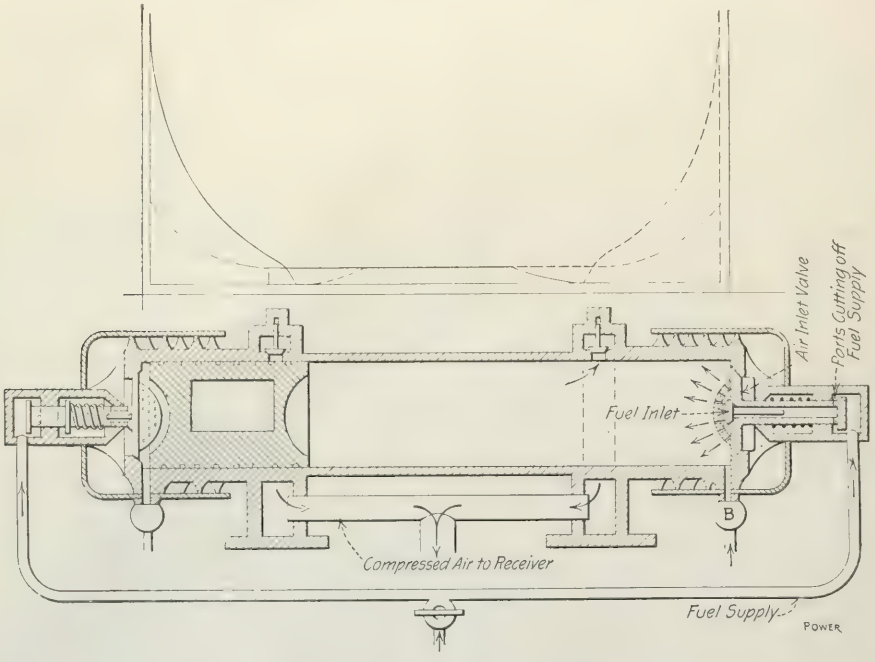
Flying Piston Gas Engine for Air Compression

Among the attempts to make the explosion process of the gas engine act directly (without connection of piston and crank) upon the driven machine, the Humphrey water pump has already reached some practical importance. The idea, therefore, arises whether it is not possible to employ this general principle in air compression, that is, to build a compressor in which the energy produced by combustion of a gas-air mixture shall be used directly for air compression without interposing a crank drive.

In the accompanying illustration, which is reproduced from *Engineering*, is shown such a device ascribed to Guiseppe Matricardi, of Pallanza, Italy. The machine has already been constructed and experimented with. The operation is as follows: A flying piston is driven by the explosion of the gas charge from one end of the cylinder to the other, while its motion compresses the air before it and forces it through a delivery valve into the receiver. The piston must, for such a process, be large enough and of proper weight, its mass being so great that the kinetic energy stored up in it during the expansion of the exploding mixture, suffices to perform the compression work. The air is forced through a passage arranged some distance from the end of the stroke. As soon as the piston has passed over this port, the escape of air is cut off and the air still entrapped in the cylinder, to which the necessary combustible mixture has already been added, is compressed; then occurs the ignition. The piston is then forced back on the return stroke, and the same process is repeated.

The operation can best be followed on an indicator diagram, as shown. At each end of the cylinder is provided an automatic valve through which the air to be compressed enters the cylinder. A cap drawn over each cylinder end compels the air, before entering, to brush the walls for the purpose of cooling them. The spindle of the air inlet valve is hollow and ends in an enlarged chamber. In the cover of this hollow spindle are a large number of fine holes through which the fuel is injected. The injection of the fuel is possible, however, only when the air-inlet valve is closed, since if the valve be open the feed of fuel is cut off. Compressed air is used for starting, the connection being at *B*.

The field of use of such a machine is restricted since it cannot handle pure air, the air delivered being mixed with the products of combustion.



SECTION THROUGH CYLINDER

Operating Features of the Diesel Engine

Much has been printed from time to time in the technical press regarding the design, efficiency and general characteristics of the Diesel engine, but little has been said from the operator's viewpoint. For information of this character we are indebted to William Sowter, who, in a recent paper read before the Dublin Section of the Institution of Electrical Engineers, set forth a number of points gained from several years' experience in the operation of Diesel-engine plants.

He is of the opinion that the life of a Diesel engine is equally as long as that of a well built steam engine, providing, of course, it receives first-class attention. In this respect he points out that a steam plant, if not properly maintained, will become less efficient as time goes on; yet it often continues to run as though everything were in order, but with an increased fuel consumption. On the other hand, a Diesel engine must be kept in good running condition; otherwise there will be difficulty in getting it to run at all and its behavior will at once indicate that something is out of order.

Regarding lubrication, he says, it is important that a pure mineral oil having a high flash point be used; also, in spite of the claim that the cost of lubrication in a Diesel-engine plant is no greater than in a steam plant, his experience has pointed to this item of cost being about 50 per cent. greater in the Diesel plant. However, while efficient lubrication is necessary an excessive quantity of oil should be avoided. The piston and gudgeon pin are lubricated by a pump which forces the oil through a hole in the liner and piston to the pin and also through an annular space between the liner and the cylinder casting. It is the usual practice to have about six small holes through the cylinder at this point and through these the oil is forced to meet the piston. If the oil becomes dirty or contains carbon these holes are liable to become clogged.

Oil from the crank case usually contains carbon and should be pumped into a settling tank, heated by a steam coil or some other means. After settling, the oil should be drawn off from the top and passed through a good filter. It is practically impossible to remove all the carbon from the oil and consequently if the oil is used repeatedly, more and more carbon will be fed into the cylinders and

other parts until sufficient accumulates to completely stop up the passages. For this reason it is recommended that the piston be removed every six months and the lubricating passages cleaned. The carbonization of oil may also cause trouble in the air-compressor cylinders. Before the author realized this possibility, filtered oil was used in the compressor. In the course of time the high-pressure air pipe leading from the compressor to the intercooler blew off at the joint, when it was discovered that the pipe was completely blocked with solid carbon.

On account of the high compression the valves must be kept tight; otherwise leakages will occur, reducing the compression and making it difficult to start the engine; moreover, when started proper combustion will be impeded. Mr. Sowter believes that each valve should be examined and ground at stated intervals. He has found that, under the conditions in which he is working, the valves will run without any attention for the following periods: exhaust, 600 hr.; air, 2000 hr.; fuel, 600 hr., and starting, one year.

The fuel valve is so adjusted that the fuel is sprayed into the cylinder a few degrees before the crank reaches the top center on the compression stroke. It is important that this valve be set properly and be tight; if there is any leakage, fuel will be admitted at the wrong time, resulting in irregular running of the engine. The atomizer is liable to become choked if the fuel is dirty and insufficiently filtered; hence it should be examined and cleaned at the same time as the needle valve.

More or less vapor is carried over into the compressor, and when the compressed air is cooled in the intercooler, this vapor is condensed and unless the drain on the intercooler is blown down frequently, water is carried over into the receivers. This tends to cause rusting which travels from the receivers and pipe to the starting valve where it will accumulate and cause the latter to stick. If this occurs, the contents of the bottle may be lost before the valve can be shut. To avoid this the intercooler drain should be blown down at regular intervals, say every half hour. The air-storage receivers are provided with siphons and drains, and should be drained every day. The high-pressure suction and delivery valves of the compressor require grinding about every 600 running hours and the low-pressure valves about every 2000 hr.

Difficulty in starting is usually due to low compression or incorrect adjustment of the fuel air or exhaust valves. Low compression generally results from a bad condition of the valves and is an indication that they require grinding. Furthermore, the engine may be overheated, due to defective water circulation or overloading, and the valve casings may be distorted, resulting in leakage. If no leakage is apparent from examination, the indicator will show if the compression is low.

The setting of the needle valve may readily be tested by barring round the engine until the crank is near the firing center; compressed air should then be admitted to the blast pipe, the indicator cock opened, and the engine slowly barred around until air commences to issue from the indicator cock. This is the point at which the fuel valve commences to open, and should correspond with the maker's setting marks. The air and exhaust valves should commence to open at the proper time and in order to insure this, most makers stamp on the engine the proper clearance between the cams and rollers at a marked point; these clearances should be checked periodically, and adjusted if necessary.

The exhaust from a Diesel engine should be smokeless, although black smoke may be emitted if the blast pressure is too low, or if the engine is overloaded. The exhaust gases escape at a pressure of about 40 lb. per square inch and a proper form of muffler should be fitted. The author recommended, where possible, to lead the exhaust to the base of an existing chimney.

Regarding cooling water, he contended that the amount required by a Diesel plant is much less than that required by a suction-gas producer or steam plant. His experience had indicated that, with an inlet temperature of 50 deg. F. and an outlet temperature of 140 deg. F., about 3 gal. per brake horsepower are required per hour.

CORRESPONDENCE

Operating with Dirty Gas

We used natural gas for many years and at its failure installed a bituminous producer plant rated at 2000 hp. for power and furnace work. It was claimed that the gas would be free from tar to an extent that no trouble would be experienced from this source. We found it very different, however, for after two hours' run on the manufactured gas the engine had to be shut down to prevent injury to the governor valve.

To relieve conditions a filter (see Fig. 1) was placed on the gas-supply line. The gas enters the filter at the top and the upper layers of filtering material catch most of the tar. These usually have to be changed weekly, although the lower layers will last as long as three weeks if the gas is not too bad. About 8 lb. of waste are required to charge the filter. It would not make much difference whether the gas came in at the top or bottom of the filter; only it permits the removal of the layers which have become so foul as to restrict the flow of gas, without disturbing the remainder.

Once a week the governor valve is removed and cleaned, and every three weeks we clean the gas passages between the governor valve and the cylinders and grind the valves.

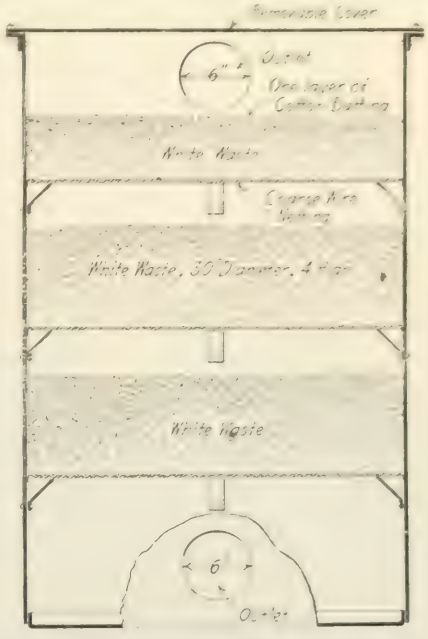


FIG. 1. SECTION THROUGH FILTER

After an engine has been in service some time, ridges will form around the seat and, as the valve stems and the guides become worn, there is a likelihood of the valves seating on these ridges, causing them to leak and affecting the satisfactory operation of the engine.

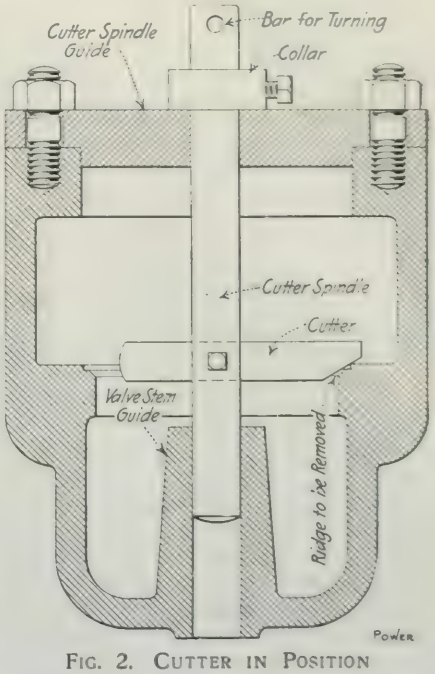


FIG. 2. CUTTER IN POSITION

Fig. 2 represents the tool used in removing the shoulders from the valve seats. The feed is gaged by letting the cutter rest on the ridge, then raising the collar the thickness of a piece of paper.

J. O. BENEFIEL.

Anderson, Ind.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Good Economy in a Gas Power Ice Making Plant

By C. A. Tupper

At Toledo, Ohio, the Citizens' Ice Co. operates an ice-making plant with a capacity of 70 tons of raw water daily. In its design the management was highly successful in securing high operating economy: (1) by the use of natural gas as fuel; (2) by its utilization in the cylinders of a gas engine rather than for raising steam in boilers, and (3) by reducing the distillation of water to a minimum.

The ice machine installed is of the "Great Lakes" design, having two vertical single-acting ammonia compressors, with 18x30-in. cylinders. The ammonia condenser is of the double-pipe style, having eleven 1¼- and 2-in. pipe coils 12 pipes high, and measuring 19 ft. over all. The ammonia receiver is 24 in. in diameter by 10 ft. long.

The freezing tanks contain the usual 1¼-in. ammonia evaporating coils provided with 696 four hundred-pound cans; the tanks are operated on the flooded system. The accumulator has one ammonia expansion valve; and a coil in contact with the humid suction gas from the tanks on its way to the machine serves to precool it.

Gas power being used for driving the ice machine, there is no exhaust steam available to make the usual distilled water ice, but ice of excellent quality is produced mainly from raw water.

The cans are filled with filtered raw water, and air is introduced into the center of each can through a ¼-in. pipe dropped into the can from the top. Ten cans are handled as a unit; that is, a 1-in. air pipe crosses over the top of each row having 10 branch pipes, one for each can. The air is supplied by a Sturtevant pressure blower driven from the main engine.

Before the ice block is entirely congealed or when there still remains a hole several inches wide, the air pipe is withdrawn; and the water, with the impurities forced out by freezing, is eliminated by introducing a piece of pipe connected to a suction pump. The center of the block is then filled with distilled water. The ice block thus obtained is a raw-water block with distilled-water core.

The ice machine is driven from a 300-hp. Allis-Chalmers horizontal tandem, four-cylinder, double-acting gas engine, 18x24 in., running on natural gas.

The gas and air are controlled separately by throttling valves located in each cylinder. This permits varying the compression to suit different loads. The main inlet valve is of the single-beat poppet type, eccentric operated. The gas valve is in one piece with the main valve and opens somewhat later than the latter, to insure scavenging the cylinder before the mixture is admitted. The exhaust gear is similar to the main inlet gear and is operated through a series of rolling levers from the main eccentric. All exhaust is muffled.

The engine produces its own electric igniting current, a small generator being belted from the main shaft.

For starting the engine, air is stored in tanks at 200 lb. pressure by a small belted compressor driven from the engine. The starting device consists of a small poppet inlet air valve at each end of each cylinder, operated by the layshaft. Air is admitted to each cylinder in turn at what would be the working stroke.

All wearing surfaces, including the main bearings, slides, crank and crosshead pins, are arranged for a continuous gravity and forced-feed oiling system, and the cylinders are lubricated by Richardson sight-feed oil pumps. The cylinders and exhaust valves are water-jacketed, and the piston rod and piston are hollow and water cooled. The consumption of gas was smaller than anticipated, but more water was used for cooling than expected.

To further increase the economy of the plant beyond that gained by using

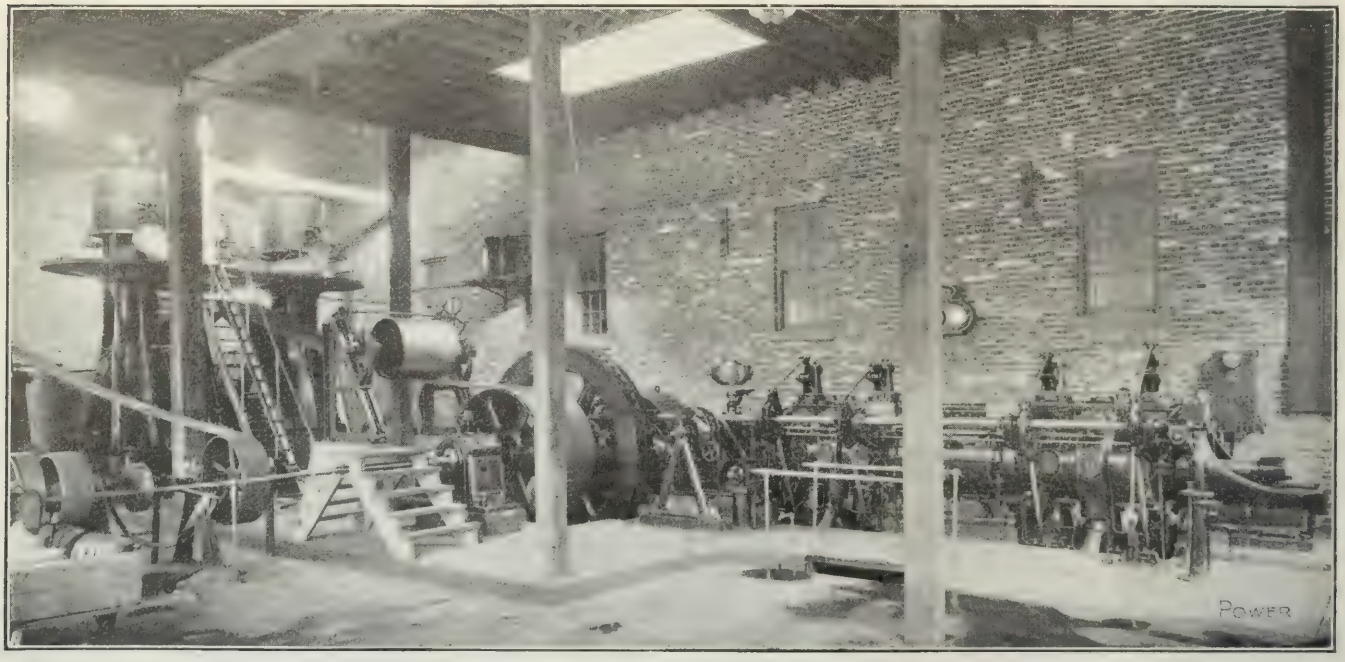


Fig. 1. Ice Making Plant of Citizens Ice Co., Toledo, O.

raw water and gas power, the distilled water for the cores is furnished by Griscom-Spencer multiple-effect evaporators. These produce several times the weight in distillate of each pound of steam supplied to them. The steam is furnished by a 20-hp. Erie City return-tubular boiler, fired by natural gas.

Two air-hoist ice cranes remove the ice to the dumps, air being furnished by the belted compressor mentioned, which was made by the Union Steam Pump Co.

The Downie water-circulating pump is a double-acting deep-well pump. The well, about 600 ft. deep, supplies an abundance of water at 53 deg. F.

The ice machine is belt-connected to the gas engine without any clutch. Full-size bypass connections on the ice machine enable the engineer to start the gas engine without load. The auxiliaries, including the water pump, pressure blower and air compressor, are driven from a line shaft belted to the shaft of the gas engine and operated through a clutch coupling. Should the gas engine be shut down for any reason, this clutch coupling is disengaged and the auxiliaries are driven by a 25-hp. motor, taking its current from the mains of the local electric company.

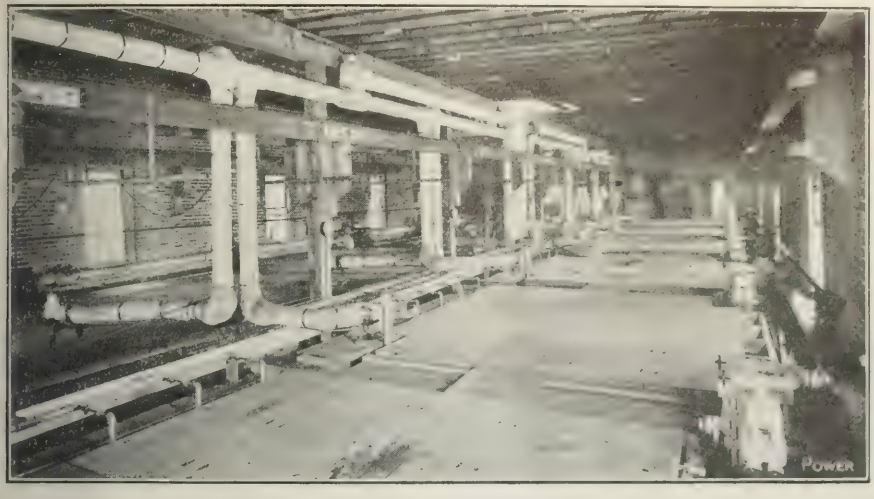


FIG. 2. FREEZING TANKS WHICH ARE OPERATED ON THE FLOODED SYSTEM

The machinery was shut down from October, 1911, until March, 1912, without overhauling, and J. M. Murphy, secretary and general manager of the company, to whom the writer is indebted for these data, states that when started up it ran perfectly.

Sweating Causes Disintegration of Pipe Covering

In the plant of a correspondent, the sweating of a refrigerating pipe has caused the partial disintegration of the 3-in. cork covering. He inquires whether wrought-iron pipe painted or galvanized pipe will remedy the trouble. The answer may perhaps be of general interest.

Unless the pipe has actually rusted through, the moisture encountered does not come from the inside; there is no such thing as sweating through a pipe. The so called sweating of cold surfaces is due to the cooling of the surrounding air by contact with the cold surface.

Air always contains a certain amount of latent humidity; in other words, there is always mixed with air a certain amount of water vapor. The capacity of the air to carry this vapor depends directly on the temperature. The warmer the air the more vapor it will hold, the colder the less. When air is saturated with vapor—that is, when it has absorbed all the vapor it can at a given temperature—the slightest drop in temperature will produce precipitation or a throwing down of a part of the moisture. If the temperature, on the other hand, rises, the air will no longer be saturated and a capacity for absorbing more moisture will have been created.

The temperature at which air becomes saturated is known as the "dew-point." When drops of water appear on a cold brine pipe or even a pitcher of ice water, it is because the layer of air immediately surrounding the cold surface has been chilled below the dew-point and a part of the moisture has no longer been able to remain in the air. In the case of a cold pipe insulated with cork or other kinds of covering, the liability of the air to be cooled to the dew-point is greatly reduced since the temperature of the outside of the covering is not nearly so low as that of the pipe.

Nevertheless, a condition of atmospheric humidity will sooner or later exist when contact with a surface only a degree or two colder than the air will produce precipitation. If there is even the smallest opening through the waterproofing on the outside of the covering, air will enter, and since the farther it passes into the covering the lower the temperature encountered, the more likelihood of precipitation. When the moisture has once been precipitated in the small openings through the waterproofing it has limited exposure to the air for reëvaporation, and unless there is a rise in temperature to increase the absorptive power of the immediately surrounding atmosphere it will remain there.

If the temperature of the insulation finally falls below 32 deg., the moisture is frozen and in so doing expands, cracking the insulation still further, and into these minute cracks the moisture flows when a successive rise in temperature melts the ice. A re-occurrence of the freezing operation still further tends toward the decomposition of the insulation and the nearer the moisture penetrates to the low-temperature pipe and the more frequent the variations in temperature, the more rapid will be the deterioration.

The place for the waterproofing or, more accurately, air-proofing of all kinds of cold insulation, is where it comes in contact with the air, and not at a place remote from the point of attack as in the case of the application of waterproof paper, paint, etc., next to the surface of cold pipes. The only possible function of such waterproofing would be to prevent the pipe from rusting after the insulation has been penetrated and rendered useless by moisture.

If the present pipe covering is badly disintegrated and shows signs of being frozen, it should be removed and replaced by a new covering. If it only shows deterioration in places, it can possibly be dried out. If there is any time of year when the pipes are not in service for a considerable length of time, they may be disconnected and the drying operation accomplished by passing steam through them. When the insulation has been thoroughly dried the exterior should be given a good coat of rubber sealing compound, supplied by the manufacturers, or several coats of paint.

Charging Ammonia into Compression System

The mechanical details of different compression refrigerating plants vary somewhat so that the accompanying diagram is only a sketch of the system. The ammonia liquefied in the condenser flows to the liquid receiver in all compression systems. From the receiver the liquid flows through a stop valve *S* to the neighborhood of the cold-storage compartment to be refrigerated or to the ice-making tank, as the case may be, where it passes through an expansion valve *E* into the expansion coils.

Somewhere between the valves *S* and *E* or the stop valve on the receiver and the expansion valve on the expansion coils, is usually located an outlet closed by a charging valve *C* to which the ship-

ping drums of ammonia charged into the system are attached. The procedure is as follows:

The shipping drum *SD*, full of ammonia, is placed in a convenient position near the outlet *C*, the valve end of the drum resting on the floor, the other end being elevated on a block 4 or 5 in. high. Connection between the valve on the drum and the charging valve can best be made through a flexible pipe connection made up with four ells and a box union something like that shown in the sketch; this connection will allow the position of the charging drum to be shifted slightly. Connection having been made, the stop valve on the charging drum may be slightly opened to allow sufficient ammonia to pass out into the connecting piping to test it for leaks.

If the piping is found to be tight, the valve *S* on the receiver can be closed and the valve *C* opened, in which case, providing the stop valve on the tank is wide open, valve *C* will act as an expansion valve. The liquid ammonia from the shipping drum is forced up hill into the system because of the pressure of the ammonia vapor in the shipping drum above the liquid.

Water from the bottom of a boiler under pressure would be forced up hill in the same way because of the pressure of the steam above the water, the only difference being that whereas it requires the high temperature of the firebox to raise steam pressure, ordinary atmospheric temperatures are sufficient to keep up the pressure in vessels containing anhydrous ammonia.

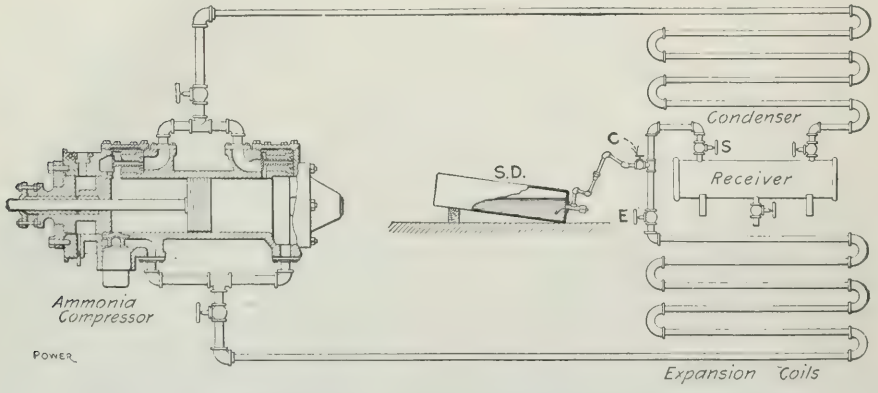


Diagram of Compression System, Showing Charging Connections

The machine may be operated under its usual working back pressure, expanding the anhydrous ammonia from the shipping drum into the system instead of from the ammonia receiver until the former is emptied, and then the expansion through the regular expansion valve may be resumed. The emptying of the drum will be indicated by the appearance of frost on the piping connecting the drum and the valve *C*. This connection will not frost so long as there is a column of warm liquid ammonia passing.

As soon as the liquid has been exhausted so that there is no longer sufficient evaporation to maintain a high pressure in the drum, the temperature of the remaining liquid ammonia drops with the decreasing pressure until at 47 lb. the temperature of the liquid reaches 32 deg., which is sufficiently low to precipitate and freeze moisture from the atmosphere. This appears as frost on the outside of the charging pipe.

The disappearance of this frost indicates that no cold liquid is passing or that the drum is empty. After closing the drum stop valve and valve *C*, the piping can be disconnected and another drum substituted.

If care is exercised in regulating the expansion through valve *C*, the operation of the machine need not be interfered with except in case of machines so large that the expansion of the ammonia from a single drum will not maintain the accustomed back pressure, in which case the machine should be slowed down until the charging process is completed.

CORRESPONDENCE

Boil-over in Absorption Machine

Boil-overs are more prevalent with vertical than with horizontal generators. The action is very similar to that of priming in boilers. The aqua in the generator suddenly disappears, passing into the condenser and filling the receiver. The extra amount of heat brought into the condenser by the hot aqua will heat the water in the condenser and thus hold up the head pressure; very often it will raise the pressure. This puts the machine out of commission as far as freezing is concerned.

When a boil-over occurs, it is best to cut the steam off of the generator, and leave it so, as long as the head pressure is great enough to drive liquor from the generator to the absorber. Plenty of cooling water should be used on the absorber and the foul gas burnt off; in other words, proceed at once to get a vacuum on the low-pressure side of the machine. The expansion valve, or dump if there is one on the machine, should be opened and thus let the liquid in the receiver work its way back to the absorber. This may take some time and often requires a high vacuum on the absorber. As soon as the liquid begins to show in the absorber gage-glass, turn a little steam on the generator and start the ammonia pump. It is well to start the machine very slowly and be careful that the condenser and dehydrator are not too cold. It may be necessary to turn off the cooling water until the machine is well started.

The fundamental cause of a boil-over is a sudden reducing of pressure in some part of the high-pressure side outside of the generator. This can happen in several ways. High water in the boilers giving wet steam reduces the amount of gas generated, thus lessening the quantity of the gas to the condenser, so that its temperature drops and likewise the pressure. This reduction in pressure does not show on the high-pressure gage, since the gage is attached to the generator. The same condition is brought about when the steam trap on the generator fails to work continuously. Very often the dehydrator will get too cold and act as a condenser, bringing about a reduction of pressure in the condenser, as in the previous cases. The above conditions are more apt to occur, where the aqua is weak or very dirty (soapy).

If the operator will be careful to regulate the steam to the generator, cooling water to the condenser and dehydrator, the expansion valve, and the speed of the ammonia pump, all to suit the load required of the machine, there will be very little trouble from boil-overs.

Gay A. Robertson.

Louisville, Ky.

Summer Course in Refrigeration

Of special interest to refrigeration engineers and students of mechanical refrigeration is the summer session of the College of Mechanics and Engineering of the University of Wisconsin.

The school opened June 22 and includes the study of heat engines, gas producers, power plants, refrigerating machinery, principles and methods of producing low temperatures, etc. It closes Aug. 30. Full information is given in a bulletin issued by the university and obtainable by addressing F. E. Turneaure, dean of the College of Engineering, University of Wisconsin, Madison, Wis.

Running a plant along the line of least attention means least pay for the engineer.—*Modern Engineer.*

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POWER

Tendencies in American Engineering Practice

As a rule, American engineers bear a reputation for being progressive, in contrast to the ultra-conservatism of the English. While this may apply in general, still one is forced to admit two tendencies prevalent in American practice regarding the adoption of anything new in the power-generation field—either skepticism totally excludes or delays the adoption of really meritorious apparatus, long after it has been in practical use elsewhere, or the enthusiasm for a new thing, highly efficient in itself, carries it into fields for which it is not adapted.

A few years ago the steam turbine attained great popularity, and many predicted that the knell of the reciprocating engine had been sounded; yet for certain classes of service the reciprocating engine is still holding its own. This is not to the detriment of the turbine, which for large units under steady conditions of load and speed, has come to stay, but it merely goes to prove that no type of prime mover, however efficient in itself, is suitable to meet all conditions of service.

Similarly, not long ago great predictions were made for the producer plant, which many believed would displace the steam plant. Upon this wave of popularity many plants of this type were installed under conditions for which they were not adapted. When the expected results failed to materialize, the producer installations were blamed, whereas the blame should have rested upon the engineer who installed the plants and who was not far-sighted enough to analyze and meet the existing conditions.

While the producer plant is thermally more efficient than the steam plant, its initial cost is greater and its life less; also certain local and operating conditions must be considered. Nominally the field for the producer is in the utilization of low-grade coals.

Still another case is that of burning oil fuel under boilers. This is used almost exclusively on the Pacific coast where coal is expensive and oil comparatively cheap, whereas on the Atlantic coast the conditions are just the reverse. In practically all cases the problem is economic and is affected by local conditions; unless worked out on that basis, good results cannot be expected.

Become a Designing Engineer

Should a steam engineer be content to remain confined between the four walls of his engine room, or should he so develop his knowledge and broaden his scope of usefulness that he will be capable of accomplishing more important work than that of starting and stopping his engines? In other words, should he confine his ability to operating a steam plant successfully, or should he fit himself to meet possible demand for extensive alterations, or the building of a new power plant; in fact, become a designing engineer? If not, where is the dividing line between the operating and consulting engineer?

The trouble with most engineers is that they are content to let someone else think for them, plan for them and supervise the carrying out of such plans.

A competent engineer should not hesitate to assume the responsibility of designing a power plant if one is required and he is given the opportunity to do so. It is true he may make mistakes; so do consulting engineers. There might be features about the finished installation open to criticism, but whoever inspected a power plant where some feature could not be improved?

Nothing venture, nothing gain, is a truism. Nothing ever would have been accomplished in the mechanical or scientific worlds, had only those who thoroughly understood the problems worked them out; there would have been no problems.

Take a 1000-hp. plant. Most engineers have a general idea of the equipment that should be installed. Sizing up the general conditions and demands upon the plant will be necessary. The well read

man could draw a general layout of the plant, and knowing the available space, operating conditions and money available could design a station that would meet the requirements and operate economically.

How? By using his eyes, profiting by his own experience and that of others and obtaining data from the operation of other successful plants.

Because it is not the custom for operating engineers to design power plants, it is no reason why he should not begin, broaden out in his work and make himself a more valuable man.

If one makes a practice of visiting power plants he will be surprised, perhaps, to find that many have been designed and erected by the chief engineer. He will find them well laid out, operated efficiently and with credit to the men who had the courage to tackle the larger end of power-plant engineering.

Furthermore, a man who can design a plant is more capable of successfully operating it than the man who is content to start and stop the engine.

Effect of Stiffness of Head on Braces

On page 97 a correspondent takes exception to our editorial in the issue of June 11 upon the above subject.

The instructions given by the Ohio Board of Boiler Rules implied that the ability of the head to withstand pressure is its unbraced capacity *plus* the capacity added by bracing.

An unbraced head will bulge under even a little pressure. The Nichols formula determines the pressure, which allows it to bulge only a permissible amount. Any such amount of bulge would, however, produce an excessive strain in the braces.

The pressure allowable upon an unbraced head by letting it deflect an amount which will not unduly strain the material, and that provided for by holding the head rigidly in place by braces, depend upon incompatible conditions, and the safety of the head can never be proportional to the sum of the two.

The formula given by the board allows an unreasonably small number of braces, as the example given in the editorial criticized shows.

If the rule which these instructions modify imposes excessive bracing, it should not have been adopted. If the bracing which it requires is necessary for safety it should be enforced. If it is advisable to continue in service temporarily, boilers which have not sufficient bracing to comply with this rule, an exception should be frankly made in their cases for the length of time desired, instead of resorting to a subterfuge, which involves a mechanical fallacy.

The Mechanical Engineer

As in other places, the line of succession keeps up in the power plant, and it is a school that, after due service, sends forth its graduates. The man in the chief position has an assistant qualified to succeed him, or steadily becoming so. This way of fitting for promotion is direct in the marine service, and in the stationary branch not markedly indirect.

In the great industrial field there are many vocations, but it is doubtful if any offers a greater incentive for studying and mastering command of the essentials of the calling than that of the mechanical engineer. In a peculiar way it has a prestige of its own; it represents power, and the greater the engineer's fitness, the more substantial are the rewards. If the occupation has risen to the dignity of a profession, the credit belongs to painstaking workers in its ranks.

In his "History of Our Own Times," Justin McCarthy states that Stephenson's locomotive ascended its throne in its great test trial at Ramhall, England, in 1832, a test in which John Ericsson figured with his own outfit. Stephenson won.

Of mechanical achievements, McCarthy remarks the hostile sentiment evinced by a certain influential class against the notice and the honor and rewards in prospect for such toilers as Watt and Stephenson. It was a vulgar intrusion of mere mechanics; it was a surprise from a humble source. But logical events stunned and subdued this class, and began the era of a new profession wherein not the designers of machines alone, but the skillful operator who handles them competently has won an enviable title.

An effort in the direction of supplying technical information to practical men, analogous to that contemplated by the Institute of Operating Engineers, is being made at Alton, Ill., under the inspiration and direction of Wilbur R. Smith. An organization of woodworking machinery operators has been effected and the plan is to organize operators of other machinery, including power-plant engineers, to secure the adoption of an industrial course in the public schools, organize classes, lectures, discussions, etc. The woodworkers are working under the auspices of the Educational Committee of the Y. M. C. A. with a membership of over one hundred and an average attendance at the classes of over fifty.

Know Your Employees

It was once believed that personal contact with his employees detracted from the dignity of the typical old-time manager. The present-day head of a concern knows that by personal contact he can awaken an interest in many of his men who hitherto seemed indifferent to all else but the pay envelope. He realizes that his own position is greatly dependent upon securing the working force's maximum efficiency. If he stands aloof, they show no interest; if he goes among them and shows a personal interest in their comfort and their work, he gets an added output and willing service.

In most industrial organizations the business methods have been reduced to an exact science, the machinery is the last word in efficiency, but the power plant is often treated as a necessary evil and about all that is known of it is that it is a big expense item on the books. One manager said recently that he had not entered the plant for six months. Such inattention discourages the engineer who, under other conditions, would take pride in his work, perform his labors willingly and study his employer's interests.

Once the "absentee" manager shows himself to the engineering force, inquires into its ways of work, is solicitous for its comfort and studies its needs, he will find that if his employees have shown indifference it is due to his own neglect, and that they are treating him just as they have been treated.

It is only natural for men to be stimulated to better endeavor when they can claim to know their employer and say that he knows them and what they are contributing to the success of his business.

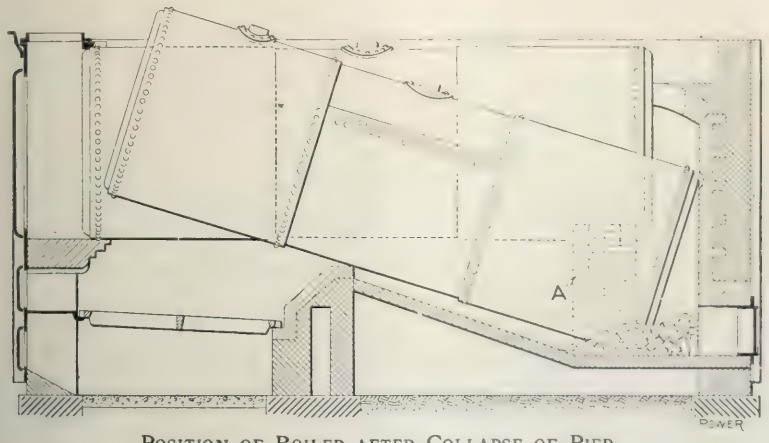
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Boiler Drops under High Pressure

On the morning of June 18, at the plant of the Windsor Print Works, North Adams, Mass., a serious boiler accident occurred to a 60-in. by 15-ft. return-tubular boiler, the setting of which is herewith illustrated. There was no loss of life but the fires had to be drawn in the 20 boilers of the plant, causing interruption of service.

All the boilers are of the return-tubular type, 60 in. by 15 ft., and two shells are supported over one furnace by a wall at the front and a brick pier and saddle at the rear. There are no center walls in the furnaces. The pier *A* supporting one boiler crumbled and fell, allowing the shell to fall on the bridge-wall, as shown in the full lines, or, as the fireman aptly expressed it, "She stood on her hind legs."



POSITION OF BOILER AFTER COLLAPSE OF PIER

Although the "twin" boiler was not disturbed in its setting, many of the steam and water connections were broken. Great excitement prevailed in the boiler room until the steam pressure was reduced and the fires in all boilers drawn.

A. W. GRISWOLD.

Cheshire, Mass.

Cleaning a Flywheel Pit

The following ingenious trick I saw worked by an engineer surprised me with its simplicity.

In our plant it became necessary to clean out a flywheel pit of an engine as it was partly filled with small stones and mud, due to some mason work done around it. The engineer took a stout rope and wound it around part of the rim of the wheel. The rope was put on thick enough to fill the space between the wheel and the bottom of the pit. The wheel was then turned a revolution and all the dirt was thrown out on the engine-room floor. This may be an old kink among power-plant men, but it was a new one to me.

FREDERICK HANSA.

New York City.

A Twisted Valve Stem

One day when making my rounds of the plant, I heard a continuous rattle in the long condenser piping. From previous experience I knew that one of the engines in my charge had broken something, so I started in their direction. When nearly there, one of the oilers notified me that something was wrong with his engine. The engine was revolving at about half speed under the load and steam was blowing through the low-pressure cylinder. The engine was shut down immediately and we started to investigate.

I first took the head off the high-pressure cylinder, thinking that possibly some of the follower bolts had become detached and got into the low-pressure cylinder, but the high-pressure piston was in good condition. The low-pressure cylinder head was also taken off and everything in the cylinder was found all right. All the snap rings were examined and showed no damage. I next took off the caps of the valves and discovered that the admission valve on the low-pressure cylinder had evidently got a bump which twisted the stem. The valve was far enough away so that the piston could not have done it, and the only conclusion I could come to was that at some previous time this engine had been wrecked and the broken parts had remained in the receiver or piping, and at this particular moment had got into the low-pressure admission valve, twisting the stem.

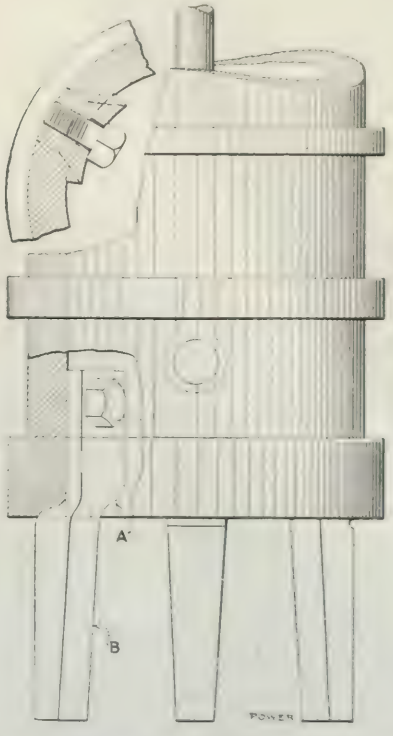
Fortunately I had a spare stem so I put it in place, set the valve and then took it down and cut a keyseat in it. This all took time, however, and it meant a six-hour shutdown before we got started again.

A. RAUCH.

Pittsburgh, Penn.

Repair to a Governor Valve

The valve of our engine governor lost one of its legs, which prevented it from sliding freely in the bore of the valve chamber. Because of this and the valve being worn, it was apt to fall a little to one side and bind against the chamber. Although it would throttle the steam and regulate the speed satisfactorily, I decided to avoid trouble by repairing the valve.



SHOWING NEW LEG ATTACHED TO GOVERNOR VALVE

It was a cast-iron valve, as shown, and I remedied the fault by making and attaching a new leg. It happened that the leg broke off flush with the valve, so I filed it smooth and from a piece of ⅜-in. square iron made a new leg. Where

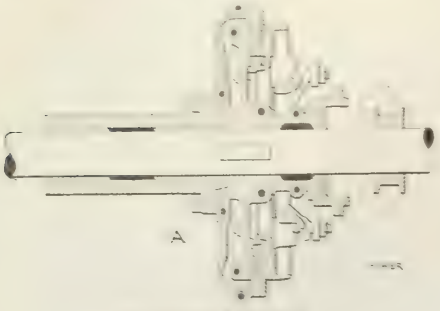
the new leg had to go, I filed a small slot *A* and into it the leg was fitted. The top of the leg, where it was fastened to the valve, was heated and drawn to give more surface as I wished to fasten it with a ⅝-in. bolt. The shape of the leg is shown at *B*. There was ample room for the head of the bolt in the recess of the valve, and it did not interfere with the action of the governor spindle. After the bolt was screwed in tight, I upset the end of it to prevent it from coming loose. The leg was filed a true circle with the opposite one and the job proved very satisfactory.

H. W. HANKIN.

Troy, N. Y.

A Faulty Friction Clutch

Recently I erected a large friction clutch upon a 3 15/16-in. shaft. But upon running the clutch I found that the stationary friction or driving plate *A*, shown in the illustration, did not run true with the shaft. I was under the impression



FRICTION CLUTCH

that the base of the plate was not true, but examination showed that the key which holds the plate did not bottom at one end on the keyseat in the shaft, thus leaving that point high, with the result that the part of the plate with the keyseat could not seat solid on the shaft. After the key was made to fit its seat, the plate ran true.

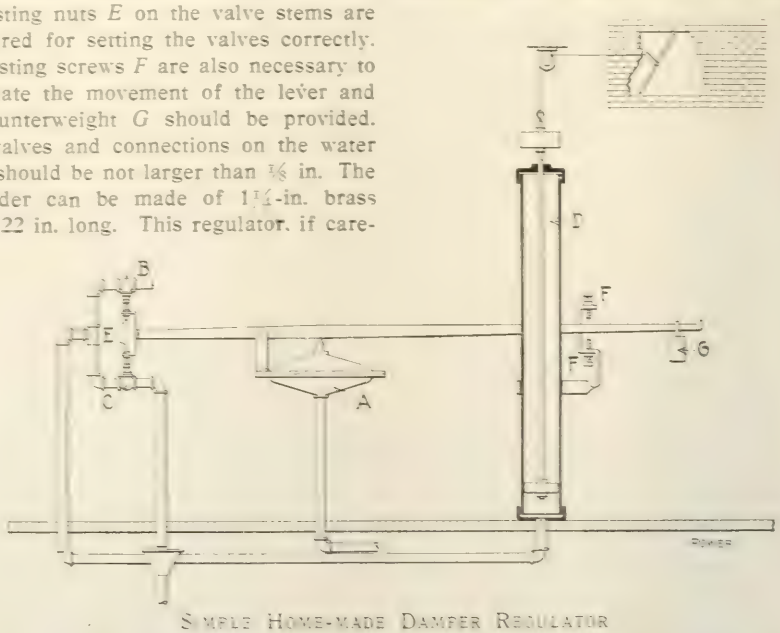
H. A. JAHNKE.

Milwaukee, Wis.

A Simple Damper Regulator

Here is a simple home-made damper regulator I made from things mostly gathered from the scrap pile. An old diaphragm from a reducing valve or something of that sort is most essential. Connect the diaphragm *A* to a steam line from the boiler. Procure two ⅛-in. globe valves and file off the threads so they will work easily, and connect as shown. As the steam pressure rises, the valve *B* opens, and valve *C* closes. Water is then admitted to the cylinder, raising the piston, which is connected to the damper by a cord, and closing the damper. As the pressure lowers, the lever drops, closing valve *B* and opens *C*, draining the cylinder *D*, allowing the weighted piston to open the damper. The valves should be connected so that the water pressure is on top of the disk, for then the lever will act much more quickly.

At 60 lb. water pressure, or over, a very small opening is required. Small adjusting nuts *E* on the valve stems are required for setting the valves correctly. Adjusting screws *F* are also necessary to regulate the movement of the lever and a counterweight *G* should be provided. All valves and connections on the water line should be not larger than ¾ in. The cylinder can be made of 1¼-in. brass pipe 22 in. long. This regulator, if carefully made with knife-edge bearings, will operate within 2 or 3 lb. change of pressure.



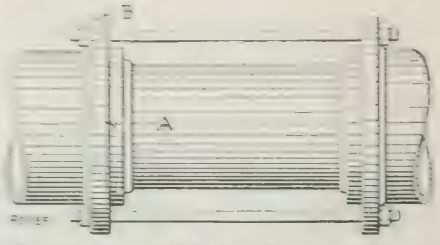
SIMPLE HOME-MADE DAMPER REGULATOR

WILSON RACE.

Jamesville, N. Y.

Emergency Steam Main Repair

While employed at a large plant within 50 miles of Boston, we undertook one week-end to renew some parts of a 15-in. steam main that was in bad condition. The measurements had been taken and new parts ordered. The pipe was flanged and we did not discover until we had



TWELVE-INCH FLANGED PIPE ON 15-IN. STEAM LINE

hoisted the pipe into place that one of the pieces was 2 ft. 2 in. short and another about 2 ft. 6 in. short. Here was a serious condition of affairs, and the nearest place to get pipe of this size was nearly 50 miles away. The telephone was kept busy for about half an hour with the result that we found we could not get a filling-in piece before Monday night, but that at a place about five miles away we could get 12-in. pipe the next morning, which was Sunday.

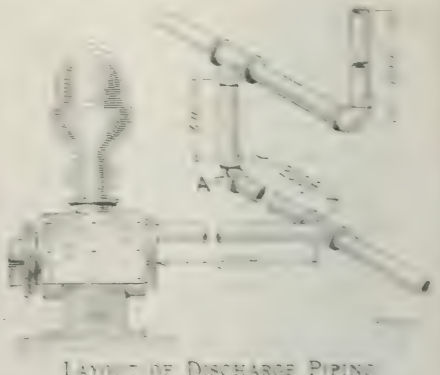
We finally got around the difficulty by making the connection illustrated. A 12-in. flange *A* just fitted inside the bolt holes of the 15-in. flange *B*, so by taking short pieces of 12-in. pipe with a flange on each end, using ⅞-in. rods with a nut on each end to connect the 15-in. flanges, we were able to start on time and run until we got the proper pieces to finish the job.

JAMES ROONEY.

Providence, R. I.

What Broke the Discharge Pipe?

A steam pump has a discharge pipe connected as illustrated. The pressure on this pipe never exceeds 40 lb. and it is wrought-iron with extra-heavy cast-iron screwed fittings. Twice it has broken at *A*, each time with a clean break, showing no flaws in the fitting. The pump



LAYOUT OF DISCHARGE PIPING

makes only 40 strokes per minute and there is no air to cause water-hammer. I know that it would be better if the 60-ft. riser went directly up instead of having the tee connected "bullhead," as shown, but, although this connection is wrong, I do not think it is the cause of the break. Perhaps some interested reader can suggest the real cause.

JOHN STRONG.

London Junction, Ont.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Effect of Stiffness of Head on Braces

In an editorial under this heading in the June 11 issue, the Ohio Board of Boiler Rules is rather severely criticized as to its ruling relating to strength of flat heads. The Board is under no misapprehension as to the strength of flat surfaces and the writer of the editorial would have been fairer if he had stated the conditions under which the instructions quoted were issued.

Under the Ohio rules, all boilers installed after Jan. 1, 1912, must be fully braced in accordance with the best engineering practice, viz: that the braces shall carry all the load; but when the new law was put into effect a large number of horizontal tubular and flue boilers were already installed that were insured by recognized insurance companies and inspected by their experts. These were considered safe for the pressures at which they were operating, and yet, under the Ohio rules, these boilers would require a number of additional braces to be fitted.

In most cases, these boilers had been operated for several years in the condition under which they were found and had shown no signs of weakness or distress in this particular, so were considered sufficiently safe by inspectors for the pressure carried. Moreover, the insurance companies had large financial interest in their safety. To avoid unnecessary hardship or injustice toward the owners of these boilers, it was decided to make some allowance for the strength of the heads, and as Nichols' rule is an accepted one, it was adopted, but with a factor of safety of 8 instead of 4, which is common.

It is generally conceded that the flat head of a boiler is safer when braced as in standard practice, yet it must be admitted that there are many boilers in this country operated by reputable concerns, and insured by the largest insurance companies, which are devoid of any bracing, and have been operated at pressures of from 80 to 100 lb.

I am a firm believer in the proper construction of boilers, but it is difficult to show an owner why his boiler, which has successfully operated for years at a certain pressure, without showing any distress, should suddenly have become unsafe and must be replaced because a new rule has been enforced in the state. This might be satisfactory to the repair man, but it would be unjust to the owner.

Most boiler explosions of late years have shown the initial rupture in some portion of the shell and not in the head, which would indicate that the old method of bracing flat heads was not altogether a failure.

In making new laws and rules pertaining to the construction and inspection of boilers, it is well to make haste slowly and to consider the fairness and justifiability of the rules as applying to plants already in existence, and which are considered by competent men as being reasonably safe. In new construction, every care should be taken to see that in all details of workmanship, design and construction, the boiler conforms to every requirement of possible safety.

JOHN NERBERG.

South Norwood, Ohio.

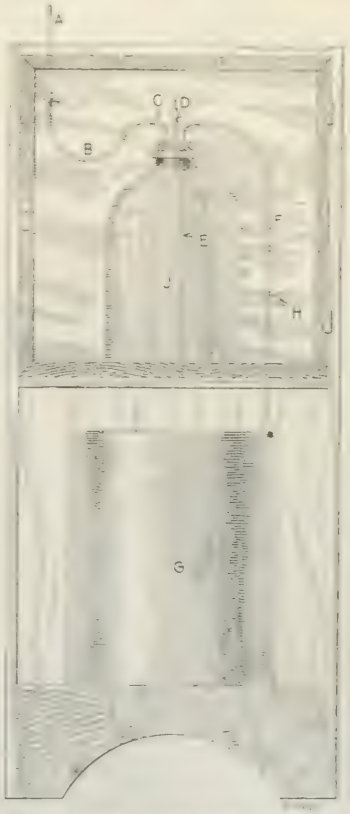
[See editorial on page 94]

Improved Flue Gas Sampler

Mr. Nelson's flue-gas sampler described in the June 4 issue is excellent, but does not possess the advantage of the one herewith described. Any engineer can make it and it is one with which cheating is impossible as the sampling bottle is locked in a case, as shown. The case consists of two sides, a top and two shelves. The upper shelf is closed in the back and has a glass door which may be locked. The lower shelf holds the waste-water can.

The sampling bottle is an ordinary 5-gal. water bottle with a cork drilled to receive three ⅛-in. brass tubes *C*, *D* and *E*. The tube *E* extends to the bottom of the bottle and acts as a siphon. The entire top of the cork should be covered with sealing wax to make it air-tight to get satisfactory results.

To take a sample, proceed as follows: Lift the can *G*, which should be full of water and of greater capacity than the bottle, to the top of the case. Now open the pinch cock *D* and allow the water in *G* to siphon through the rubber tube *F* into the bottle *J*. The rubber tube *F* is portable as it is only dropped through a hole in the upper shelf when the sampler is in operation. Allow the bottle to fill until the water flows from the cock *D*. This drives all the air from the bottle. The can *G* is now returned to the bottom shelf, the rubber tube *F* put through the hole and the sampler is ready to operate. The flue gas comes through the ⅛-in. pipe *A*, which is connected to the aspirating tube in the flue through the rubber tube *B* and the brass tube *C* into the bottle *J*. The water in *J* siphons out through the brass tube *E* and rubber tube *F* into the waste can *G*. The flow is regulated



FLUE GAS SAMPLING OUTFIT

at the cock *H*. After this is regulated satisfactorily the case is locked until it is desired to take a sample.

A. POHLMAN.

Brooklyn, N. Y.

B. S. Nelson's gas-sampling apparatus, described in the June 4 issue, is simple and, while answering some purposes, it is not as accurate as the one herewith described.

If it be only desired to determine the amount of CO_2 in the flue gases, Mr. Nelson's method will work admirably; but if, as in many cases, it is desired to make a full analysis, his method is not accurate. In a complete analysis, hydrogen and hydrocarbons are determined; gas collected over water will show an appreciably greater percentage of moisture

and hydrogen than is present, and thus give an erroneous idea of the efficiency of the furnace.

Let *A*, Fig. 1, represent a sampling

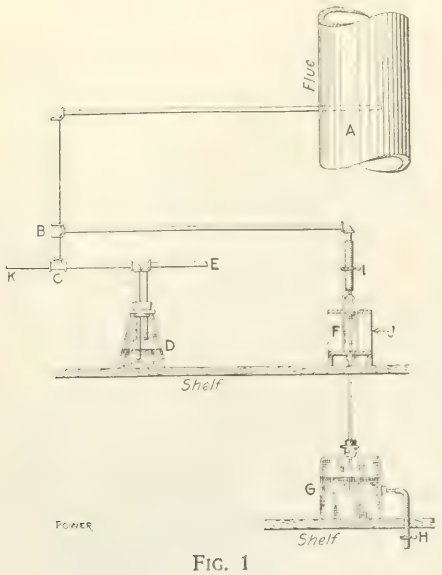


FIG. 1
FLUE GAS SAMPLING OUTFIT READY FOR OPERATION

tube placed in the flue. This should be of ¼-in. pipe extending nearly across the flue, open at the end, and with a number

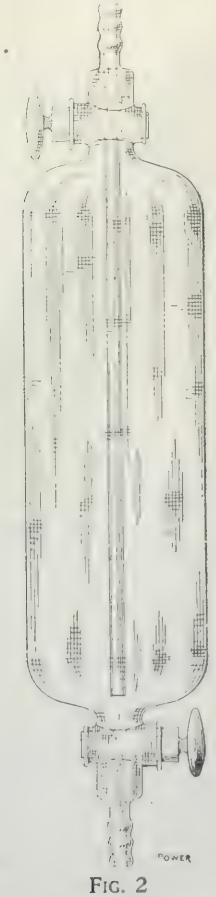



FIG. 2
SAMPLING BOTTLE

of holes drilled in it in a plane at right angles to the flow of the gases. *B* is a tee connected to the sample bottle *F* and to a steam line *K* through another tee *C*. The flask *D* is filled with water, to indicate the aspirating effect in the flue. After the steam has passed through *D* it exhausts at *E*.

The sample bottle *F*, constructed as shown in Fig. 2, is filled with mercury, which is displaced by the gas and collected in the bottle *G*. A fresh sample bottle may be filled with mercury from *G* through a rubber tube fitted with a pinch-cock *H*. The wooden stand *J* supports the sampling bottle.

The action of the apparatus is as follows: Steam passing through *K* causes an aspirating effect in the sample tube *A*, thus drawing gas from the flue. Some of this passes from *B* to the sample bottle *F*, while some is drawn down through the aspirator *D* and forced out at *E*. The sample being collected over the mercury is a true one and shows no greater percentage of moisture or hydrogen than is actually present.

A pinch-cock *L* should also be applied on the rubber tube connecting the sampling line with the bottle *F*. Sample bottles similar to the one shown in Fig. 2 may be obtained from any dealer in chemical apparatus.

CHARLES M. REED.

Annapolis, Md.

Calculating Horsepower on the Slide Rule

I read with interest the rules given by G. W. Richardson and F. R. Low for calculating horsepower on the slide rule and beg to be allowed to criticize them and offer something better.

In the first place the rules as given by Mr. Richardson are much too long to be remembered. Secondly, he uses a special mark on his slide rule which is bad, for if a mark were to be made on the slide rule for every constant one uses it would be covered with marks and the regular graduation lines be hard to read. The rule given by Mr. Low is better, but the constant used is hard to remember because it is an abstract number with no meaning attached. In addition it leaves the decimal point to be estimated, which is all right as long as familiar sizes are dealt with but not when one calculates an example that deals with sizes so small or so large that the result cannot be guessed at.

The best way of solving this particular problem is to *Multiply the m.e.p. by the piston speed and divide by* 100. *This gives the horsepower for a* 20½*-in. cylinder.*

It is then easy to find the horsepower for any other cylinder by merely changing this result in the ratio of the squares of the diameters. Thus for a 10¼-in. cylinder divide by 4, for a 41-in. cylinder multiply by 4. This can be expressed by the proportion

$$20.5^2 : \frac{Pp}{100} :: d^2 : hp.$$

which would be set on the rule as follows:

$$A \text{ scale, } 20.5^2 \qquad d^2$$

$$B \text{ scale, } \frac{Pp}{100} \qquad hp.$$

where

P = Piston speed in feet per minute;
p = M.e.p. in pounds per square inch;
d = Diameter of cylinder in inches;
$hp.$ = Horsepower.

The squares of 20.5 and of d will be found on the A scale above those quantities themselves on the D scale, and one can work with the quantities themselves instead of their squares, using the line on the runner to carry them across from the D to the B scale. The setting then becomes

$$B \text{ scale, } \frac{Pp}{100} \qquad hp.$$

$$D \text{ scale, } 20.5 \qquad d$$

EXAMPLE: What horsepower is developed by a 12x24-in. engine running at 130 r.p.m. with a mean effective pressure of 42 lb.?

First operation: calculate the piston speed.

$$P = 130 \times 4 = 520 \textit{ ft. per min.}$$

The multiplication by 4 is because the piston travels 2 × 24 in. = 4 ft. at each revolution.

Second operation,

$$\frac{Pp}{100} = \frac{520 \times 42}{100} = 218$$

A 20.5-in. cylinder will then develop 218 hp. under the above conditions.

Third operation,

$$20.5^2 : 218 : : d^2 : hp.$$

B scale, 218; 74.8 *answer*
D scale, 20.5; 12

The 218 is to be located in the second or right-hand half of scale B.

In performing this operation a value is assigned to each line on scale D, corresponding to the line which has the value 20.5. Thus the left-hand index will be 10 and the right-hand index will be 100. The line 12 on this scale will then have the value 12. This is transferred to scale B, where we find the answer (748); but the 218 has fixed the value of scale B to be 10 — 100 — 1000, that is to say, the left-hand index of scale B has the value 10; the center index has the value 100 and the right-hand index has the value 1000, and since the answer, 748, lies in the first half of scale B or between 10 and 100 it must be 74.8 and there is no need of pointing off 7 decimal places or counting how many times the right-hand end or the left-hand end of the slide projected.

This formula is based on the fact that the area of a 20.5 circle is 330 sq.in., which multiplied by the above constant of 100 gives 33,000, the equivalent in foot-pounds per minute of one horsepower.

F. F. NICKEL.

East Orange, N. J.

Turbine or Engine

I like your idea as to what constitutes a rotary and turbine engine in the editorial under the above, in the June 4 issue. I quote the latter part of the second paragraph of the article referring to the Spiro motor in the Feb. 6 issue in substantiation of my belief that it is a rotary engine, to wit: "When the ends of the grooves in which the action occurs pass the line of contact so that they are no longer closed by the teeth of the opposite gear, the fluid would be exhausted, as indicated at the end of the gear in Fig. 2," which is conclusive proof that this pressure is held upon moving surfaces until it finds an opening for exhaust, while in the turbine the angle of the vanes imparts a zig-zag motion to the entering steam which in no wise particularly affects the constancy of the flow after starting up.

Turbines, as I understand them, admit of using only one rotor altogether from the above fact in their working, while a rotary engine, being operated by a static pressure, requires some form of "head"* in order that the steam may gain some kind of a "toe hold" to be effective upon the surrounding surfaces. As will be seen from observing the characteristics of the different types, this head may vary in form greatly, and as I see it this extra rotor in the Spiro is only a modification of some existing types.

LLOYD V. BEETS.

Nashville, Tenn.

Breakage of Separator Glasses

Many readers seem interested in the breakage of separator gage-glasses, but no one has as yet advanced a tenable theory as to the cause.

It is my belief that changes in the temperature of the steam in the separator body due to the variation in pressure, are responsible for this trouble. If a steam-engine indicator be connected to the separator, it will show that the pressure fluctuates, sometimes as much as 12 or 15 lb. with each stroke of the engine, which induces severe temperature variation. This sudden change of temperature of the steam inside the glass, though small, occurs twice for each revolution of the engine and soon affects the molecular structure of the glass, resulting in crystallization. This is proved by the fact that when one of these glasses fails, it does not split or show a clean break, as a boiler gage-glass often does, but breaks up into minute particles which are thrown out in a veritable shower.

When the glass is set out from the separator body, the inside of the connecting pipes is covered with a thin film of water, which has a tendency to check any sudden variation in temperature in the pipe or glass by reëvaporating when the temperature falls, or adding condensation when it rises. This equalizes or counteracts the temperature changes and protects the glass.

F. P. READ.

Memphis, Tenn.

*Abutment [Ed.]

Saving the Heat Units

J. C. Hawkins in May 21 issue under the heading, "Saving the Heat Units," shows by his deductions and figures that a saving of 74 tons of coal a year is being effected. This looks good at first reading, but I do not think Mr. Hawkins is saving all that he thinks he is.

Before he changed the air supply it was taken in at 75 deg. F. and heated in the furnaces to the temperature of the flue gases, which requires an expenditure of many B.t.u. The air over the boiler, at a temperature of from 140 to 175 deg., he states he uses under the grate, thereby saving the heat required to raise the temperature from 75 to 175 deg., but for mechanical reasons he keeps the temperature down to 120 deg. Before the change, the air over the boilers was of a high temperature, and the temperature difference between it and the boilers, headers and piping was low and necessarily the rate of heat flow was low.

After the change a current of cold air is continually moving across the boilers. The temperature difference will be greater and therefore the heat will flow from the boilers, headers and piping much faster. It is therefore evident that Mr. Hawkins is not getting his hot air for nothing.

The difference between 75 and 175 deg. must be made up somewhere, so the heat is being supplied by radiation from the boiler.

H. R. BLESSING.

Philadelphia, Penn.

Efficiency of Joints in Boiler Shells

The article under this heading by H. L. Strong in the May 14 issue gives the impression that the only satisfactory longitudinal seam is a quadruple-riveted double-butt strap seam. After a complete discussion for calculating the efficiency of a single-riveted lap joint, at the bottom of the first column on page 704, the author states, "Such a joint is unsatisfactory." The riveting is weaker than the net section between the rivet holes but this does not necessarily make the joint improper. He argues that the efficiency of the joint is low, but this is no reason for condemning the joint.

In turn he condemns each joint for these same reasons and not once mentions the pressure he wishes to carry, upon which all of his calculations should be based. Then he infers that a quadruple-riveted butt-strap joint is the ideal seam. As a matter of fact this seam is seldom used, even on horizontal return tubular boilers where a seam of high efficiency is especially desired. Instead, triple-riveted butt joints are common. Their efficiency is fairly high and they are much cheaper to manufacture than the quadruple butt-strap joint.

The use of a quadruple or even a triple-riveted seam would be the height of folly in some cases; for example, on the drum of a Stirling boiler. A part of the shell plate in this boiler is cut out for the tubes and the efficiency at this section through the tube holes is only 45.5 per cent. Aside from the inherent objection to lap joints, which the author did not mention, the first seam would be perfectly satisfactory as long as the working pressure is kept below 85.8 lb.; for example, on a boiler generating low-pressure steam for heating purposes. Each successive joint is satisfactory as long as used on a boiler working below the corresponding pressure for the joint.

If the author had refrained from saying after each investigation that "The joint is unsatisfactory" and merely called attention to the fact that with more elaborate joints the efficiency is higher, the reader would not be left with the impression that no joint but the quadruple-riveted butt strap is worth anything, when in reality each of the other joints has its field of usefulness.

CARL E. WOLFSTYN.

Ann Arbor, Mich.

Mr. Wolfstyn's criticism of my article on "Efficiency of Joints in Boiler Shells" is appreciated, but I think he has overlooked the fact that the article was not intended to deal with the merits or demerits of any type of joint or boiler, but simply to demonstrate the methods of calculating the efficiency of the several types.

I am aware that all of the several types of joints have their fields of usefulness and that none of them are to be absolutely condemned. What was intended in saying that several of the joints under consideration were unsatisfactory was that the design or layout of those particular joints was unsatisfactory because there was too great a difference between the strength of the rivets and net section of plate between the rivet holes. It is commonly understood that the rivets in a unit length of joint should be as strong as the net section of plate between the rivet holes on the outer row.

If the layout of a joint of any type does not meet this condition it is unsatisfactory. The efficiency of the joint in such a case is not as high as it might be, and maximum efficiency can only be obtained by redesigning the joint and altering the pitch sufficiently to bring about a balance between the two factors mentioned. The unsatisfactory joints were not of standard design and the object in not making them so was given.

H. L. STRONG.

Portland, Maine.

Inquiries of General Interest

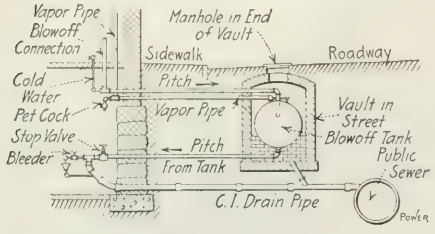
Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Underground Blow-off Tank

How can a boiler blowoff tank be placed under a private street with connections for receiving blowoff water from boilers, and be provided with suitable connections for discharging into a public sewer which is 12 ft. below the surface? How large should such a tank be in proportion to size of boilers?

L. G. S.

The object being to provide means for cooling the blowoff water before it is discharged to the sewer, the tank should be large enough to hold the largest quantity of water blown off in any 24 hours, unless cooling water is used. In the latter case the tank needs to be only large enough to receive a comparatively small



UNDERGROUND BLOWOFF TANK

charge of blowoff water, to be cooled down by addition of about twice the quantity of cold water, and the mixture can be immediately discharged to the sewer. The annexed sketch explains the main features of the connections. Under any circumstances the tank should be strong enough to safely carry the full working pressure of the boilers, with liberal allowance for deterioration from corrosion.

Notched Weir

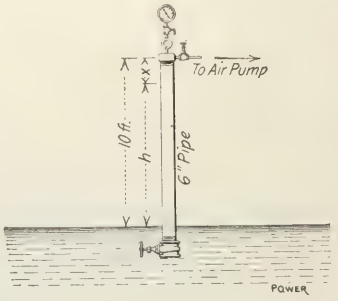
With the same head above the bottom of the weir, will the average velocity of flow per square inch of sectional area be the same over a triangular notched weir as over a rectangular weir? What is the formula for flow over a right-angled triangular weir in thin plate?

D. P. B.

The average velocity will be different. According to Prof. Thomson's experiments the flow over a symmetrical right-angle weir in thin plate is, Q in cubic feet per second $= 2.54 \times \sqrt{h^5}$, h being the depth, in feet, of the apex of the notch below the surface of still water in the pond.

Rise of Water in Vacuum Pipe

A vertical 6-in. pipe, as shown by the sketch, is provided at its lower end with a stop valve and dips below the surface of water of a pond. After closing the lower valve, water stands in the pipe at the level of the pond. The upper end of the pipe is capped and is provided with a vacuum gage and small stop valve,



RISE OF WATER IN VACUUM PIPE

which is connected to an air pump. Starting at zero on the vacuum gage, air is exhausted until the gage indicates 28 in. vacuum. If the air valve is then closed and the lower valve is opened, how high will the water rise in the pipe and how many inches of vacuum will the gage indicate?

J. W. D.

Neglecting the relatively small volume of the connections to the gage and the air valve, water would rise in the pipe a distance h until its weight, combined with the compression of air acting down on the column of water, balances the pressure of the atmosphere, i.e., 14.7 lb. per square inch absolute. Twenty-eight inches of vacuum would equal

$$(30 - 28) \times 0.491 = 0.982 \text{ lb.}$$

pressure per square inch, absolute. Assuming that temperatures remain constant and calling x the volume occupied by the air after the water is admitted, and p the absolute pressure, in pounds per square inch, then the pipe being a uniform cylinder

$$p \times x = 0.982 \times 10$$

that is,

$$p = \frac{9.82}{x}$$

The pressure per square inch due to a column of water 1 ft. high being 0.433 lb. per square inch, a water column h feet high would create

$$h \times 0.433 \text{ lb.}$$

pressure per square inch at its base. Therefore,

$$\frac{9.82}{x} + 0.433h = 14.7$$

and as $h = 10 - x$, x is found to be 0.91 ft., $h = 9.09$ ft. and

$$p = \frac{9.82}{0.91} = 10.79 \text{ lb.}$$

per square inch, absolute.

Reduced to inches of mercury, this pressure would be

$$\frac{10.79}{0.491} = 21.97 \text{ in.}$$

which in inches of vacuum would be

$$30 - 21.97 = 8.03$$

Setting Valves of Duplex Pump

What is the method of setting the steam valves of a duplex pump?

W. J. C.

Remove all cylinder heads and push the piston rod of one side toward the steam cylinder until the piston would be far enough over to strike the head end of the cylinder and make a mark on the rod at the end of the stuffing-box follower. Then move the rod toward the water cylinder until the water piston would strike the cylinder head of the water end and make another mark on the rod at the end of the same follower as before. Locate a mark midway between the two marks and move the rod back until this central mark comes to the end of the same follower. The rod is then in central position.

Uncover the steam chest, move the valve to its central position and adjust the length of the valve rod so that the nut or setscrews which move the valve will be central with the lugs on the valve when the valve-rod link pin is in place. Then set the steam valve of the other side of the pump in the same manner.

Running Turbine Non-Condensing

A 250-kw. turbo-generator, at present running condensing and driving full load, is supplied with steam at 150 lb. boiler pressure from a 78-in. by 18-ft. boiler, which has 164 three-inch tubes; the grates are 6x6½ ft. and supplied with natural draft. Can sufficient steam be generated under the same boiler conditions for running the turbine noncondensing when driving the same load?

J. G. D.

Assume that a turbine of this size when running condensing uses 20 lb. of steam per kilowatt-hour, and running noncondensing would use practically twice as

much, making the steam consumption 250 × 40 = 10,000 lb. per hour, then allowing an evaporation of as much as 10 lb. of water per pound of coal, 1000 lb. of coal would have to be burned per hour on a grate of 39 sq.ft., that is, 25.6 lb. of coal per square foot of grate per hour, which could not be done with ordinary natural draft.

Study Questions

This Week's Questions
Last Week's Answers

(46) Over what yard area can a portable electric drill be used that is attached by a 35-ft. cable to one corner of a building 30 ft. long by 20 ft. wide?

(47) A steam engine, a steam turbine and a gas engine were installed at different times. The steam engine's years of service equal those of the other two combined. In 5 years more the gas engine will be half as old as the turbine and in 10 years half as old as the steam engine. How long has each been in service?

(48) What is the equivalent in units of work to the heat energy required to bring 1 gal. of water at 62 deg. F. to the boiling point?

(49) Calculate the weight of a 12-ft. length of 6-in. extra-heavy pipe, given the outside diameter 6.625 in., inside diameter 5.75 in. and weight per cubic foot 480 lb.

(50) Neglecting air resistance, how far will a body fall from rest in 10 sec., and what velocity will it acquire?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(*Answers*)

(41) Let

a = Capacity of the tank;
b = Amount of water drawn out per hour.

Then $a + 8b$ = amount pumped in 8 hr. at 60 strokes per minute.

$$\frac{a+8b}{8\times 60\times 60}=\frac{a+8b}{28{,}800}=\textit{amount pumped per stroke} \quad (1)$$

and $a + 4b$ = amount pumped in 4 hr. at 72 strokes per minute.

$$\frac{a+4b}{4\times 60\times 72}=\frac{a+4b}{17{,}280}=\textit{amount pumped per stroke} \quad (2)$$

Equating (1) and (2)

$$\frac{a+8b}{28{,}800}=\frac{a+4b}{17{,}280}$$

$$17{,}280\,a + 138{,}240\,b = 28{,}800\,a + 115{,}200\,b$$

$$11{,}520\,a = 23{,}040\,b$$

$$a = 2b \quad (3)$$

Therefore with pumping stopped the tank would empty in 2 hr.

(42) Let

x = Number of hours pump must run at 84 strokes per minute.

Then

$$a + xb = \textit{amount of water to be pumped} \quad (4)$$

From (3) in previous answer $a = 2b$; $b = \frac{a}{2}$.

Substituting for b in (4)

$$a + \frac{xa}{2} = \textit{amount to be pumped}$$

Substituting for b in either (1) or (2) of last answer, it is found that

$$\frac{a}{5760} = \textit{amount pumped per stroke}$$

Then

$$\frac{a}{5760}\times 84\times 60=\frac{7a}{8}=\textit{amount pumped per hour at 84 strokes per minute}$$

Therefore

$$x=\frac{a+\frac{xa}{2}}{\frac{7a}{8}}$$

$$\frac{7ax}{8}=a+\frac{ax}{2}$$

$$7ax = 8a + 4ax$$

$$3ax = 8a$$

$$x=\frac{8}{3}=2\tfrac{2}{3}\ hr.$$

(43) Denote the resistance of No. 1 wire by R_1 and No. 2 by R_2. Similarly let I_1 be the current in No. 1 and I_2 that in No. 2. Since both wires are joined in multiple, the drop in potential is the same through each; let this be E. Then

$$E = I_1R_1 = I_2R_2$$

and

$$I_1=\frac{I_2R_2}{R_1}=\frac{I_2\times 7}{5}$$

But the total current is 38 amp., from which

$$I_1 + I_2 = 38$$

Substituting,

$$I_2+\frac{I_2\times 7}{5}=38$$

$$\frac{12\,I_2}{5}=38$$

$$I_2=\frac{38\times 5}{12}=15.83\ amp.$$

and

$$I_1 = 38 - 15.83 = 22.17\ amp.$$

(44) The products of the specific heats, weights and changes of temperature for the two substances, copper and water, must equal each other.

$$cw\ (t - T) = c'w'\ (T - t')$$

where

c = Specific heat of copper = 0.0951;
w = Weight of copper = 1 oz.;
t = Initial temperature of copper = 212 deg.;
T = Final temperature of copper = value to be found;
c' = Specific heat of water = 1;
w' = Weight of water = 1 lb. = 16 oz.;
t' = Initial temperature of water = 55 deg.

Then

$$0.0951\times 1(212 - T) = 1\times 16(T - 55)$$

$$0.0951\ (212 - T) = 16(T - 55)$$

$$20.1612 - 0.0951\,T = 16\,T - 880$$

$$16.0951\,T = 900.1612$$

$$T=\frac{900.1612}{16.0951}=559276.$$

Therefore the rise in temperature is 0.9276, or nearly 1 deg.

(45) The formula for the modulus of elasticity is

$$E=\frac{P\,t}{A\,e}$$

where

P = Total force in lb. = 50;
t = Length in in. = 100 × 12 = 1200 in.;
A = Cross-section in in. = ½ × 0.02 = 0.01 sq.in.;
e = Deformation or stretch in ft. = value to be found;

$$30{,}000{,}000=\frac{50\times 1200}{0.01e}$$

$$e=\frac{60{,}000}{30{,}000{,}000\times 0.01}=\tfrac{1}{5}\ in.$$

Adamat Rubber Matting

A composition rubber matting, manufactured by the Flintkote Manufacturing Co., 66 Beaver St., New York City, is known as Adamat flooring. It is composed of a felt mat base with gum surfaces, making both sides usable. It has no odor, is sanitary and has the advantage of a nonslipping surface without being corrugated. It will not harden or oxidize and can be had up to 90-ft. lengths.

Our National Chamber of Commerce

A National Chamber of Commerce has been organized at Washington, D. C., and a central office established at that city. The object is to provide a national clearing house for the development and consideration of business opinion and to secure united action upon questions affecting the commercial interests of the United States.

Members of the National Chamber of Commerce will be elected from commercial or manufacturers' associations and annual conventions will be held at representative cities. Reports will be published from time to time giving the results of the discussion of the various questions taken up before the Chamber of Commerce at the conventions.

The Shear-Klean Grate

The peculiar combined rocking and sidewise motion imparted by manipulating the lever *A*, Fig. 1, is the feature of the new grate manufactured by the Shear-Klean Grate Co., Monadnock Block, Chicago. Alternate teeth of the steel pin *J*, Fig. 2, cored into the bottom part of each grate bar, projects through the cam slots *I*, Fig. 1, in the cam operator *F*. These slots are cut at an angle of approximately 60 deg. to the center line of the cam operator. Thus, when the cam operator is swung from side to side it has the effect of pushing the

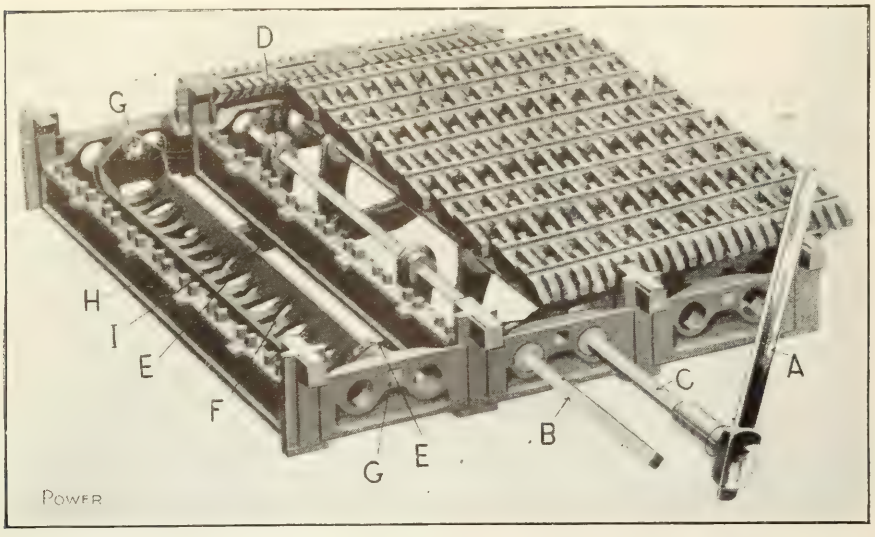


FIG. 1. PARTLY ASSEMBLED VIEW OF THE SHEAR-KLEAN GRATE

grate bars are long and the intermediate ones short, and the longer teeth of adjacent bars interlock when the grate is assembled.

In Fig. 1 the bars on the left half of the grate have been removed to show the operating mechanism. One-half of the grate is operated by the shaft *C* and pins *J* out of the vertical position and causing the grate bars to rock back and forth.

Every second slot is longer than the intermediate ones so that the pins on the intermediate bars strike the ends of their slots when the operator has been rotated a certain amount. Further rotation of

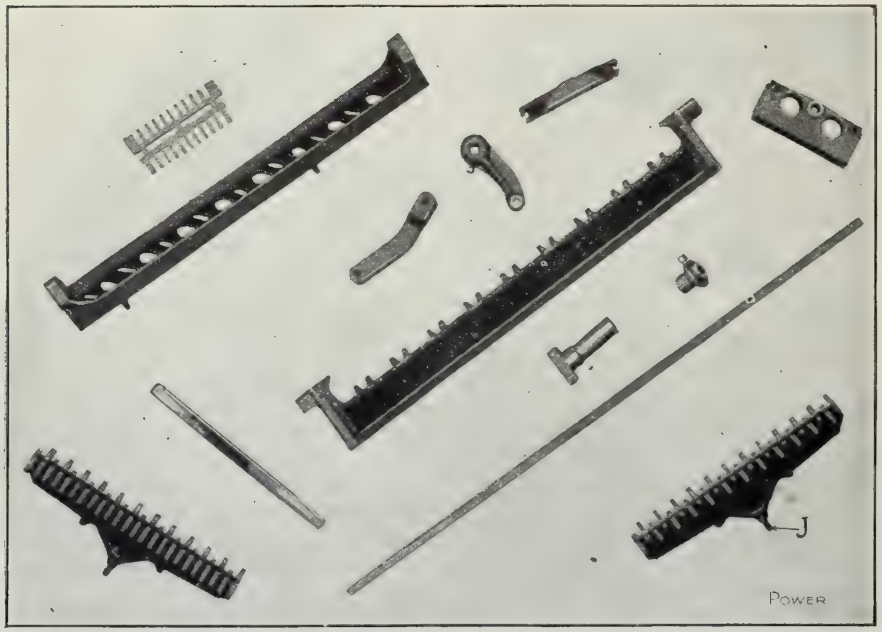


FIG. 2. THE PARTS OF THE SHEAR-KLEAN GRATE

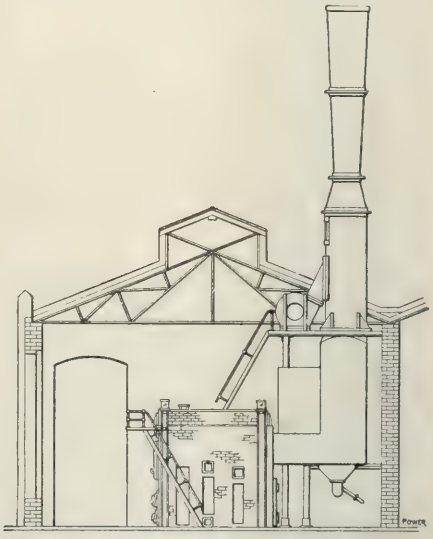
the other half by shaft *B*. When shaft *B* is rotated the levers *D* and links *E* transmit the motion to the cam operator *F*, which swings on the pins *G*. The grate bars rest on the two bearing frames between guides *H*. A long hardened the operator causes it to push these pins sidewise and hence every other bar is shoved along sidewise in its bearings until its teeth strike the teeth of the adjacent bars. Reversing the motion of the operator causes all the grate bars to tilt in the opposite direction, the teeth of one bar scraping against those of its neighbor. Thus the teeth are self-cleaning and any slag or clinker is scraped off and falls into the ashpit.

It is claimed by the manufacturers that because of this double action of the grate the ash and clinker are cut off from the bottom of the fire in horizontal layers and no unburned coal is wasted by falling through between the bars.

Air-Jet Chimney Ventilator

In this system the draft is created by an air-jet chimney ventilator, the operating air being produced by a rotary blower which creates a suction or negative pressure in the furnace. The blower consumes but ½ of 1 per cent. (at the heaviest load of the grate 1 per cent.) of the fuel burned.

As the combustion gases do not pass through the blower expensive repairs and



AIR JET CHIMNEY VENTILATOR

replacements are eliminated. This system is said to considerably increase the capacity, efficiency and advantages of the superheater and economizer, as small quantities of gases of high temperature are utilized in them. With a varying steam consumption one boiler may be forced for a time without putting extra boilers in operation, and it is possible to use low-grade fuels like lignite, brown coal, etc., in the furnace.

This ventilator is manufactured by Schutte & Koerting Co., Thompson and Twelfth Sts., Philadelphia, Penn.

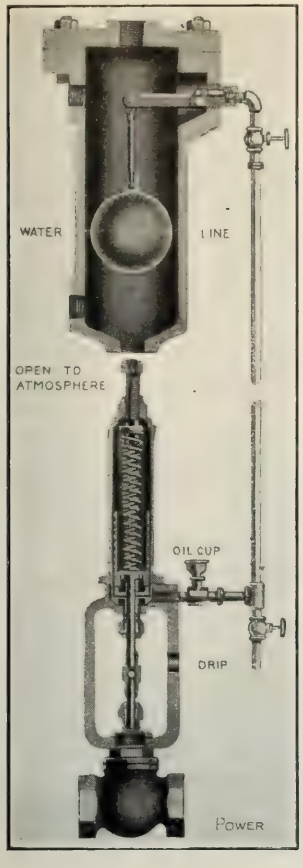
Grip-Tite Belt Dressing

This preparation is applied to belts to fill up cracks, give a velvety surface and prevent slipping. It is also said to remove moisture and lubricating oil, and to prolong the life of the belt. The dressing is applied in a liquid state. The Grip-Tite Co., New Brunswick, N. J., is the manufacturer.

Murray Boiler Feed Regulator

The Murray automatic boiler-feed regulator, illustrated herewith, is designed to automatically hold the water in the boiler at a predetermined point.

When the water in the boiler falls below its normal level. the float in the water



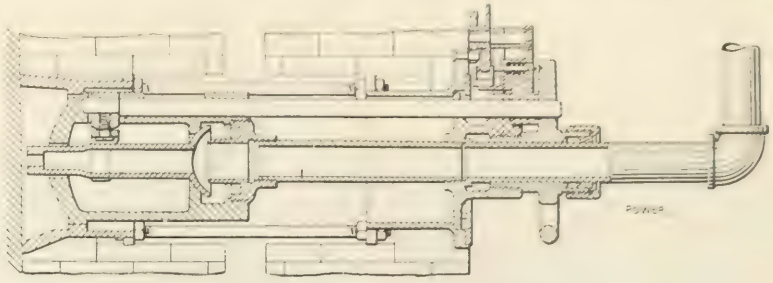
MURRAY REGULATOR

column opens the needle valve and admits steam through the ¼-in. pipe into the lower end of the cylinder, thereby forcing the piston up against the tension of the spring. This action causes the piston rod to open the regulating valve. Water is then admitted through the feed pipe into the boiler until the normal level has been restored, when the needle valve will again close as the float arises. This shuts off steam from the ¼-in. pipe and allows the spring to force the piston down, closing the regulating valve. This regulator is manufactured by the Murray Specialty Manufacturing Co., Detroit, Mich.

Improved Monarch Flue Blower

The improvement of this blower provides an automatic means for directing the steam-jet nozzles, so that the operator cannot neglect any tube in a boiler when using the machine. With the new design, it is now only necessary to turn on the steam and rotate a handwheel three or four times to clean all of the boiler tubes. The steam nozzle delivers two or more jets of steam into the tubes and registers with each tube by an automatic arrangement on the handwheel.

The device is located in the rear wall and blows the soot out in the direction of the natural draft. The movement of the nozzle is secured by means of a dog and ratchet on the handwheel, which is alternately self-engaging and self-releasing. This tube blower is manufactured by the Monarch Steam Blower Co., 187 River St., Troy, N. Y.



IMPROVED MONARCH TUBE BLOWER

Sterling Double-Oiling System Pump

The Sterling double-oiling system pump, Fig. 1, is designed to handle clean and dirty oil at the same time. The main casting contains a steel rack plunger

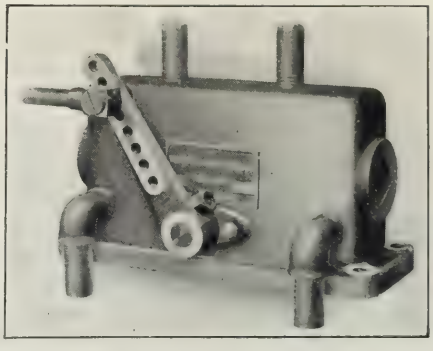


FIG. 1. STERLING DOUBLE OILING SYSTEM PUMP

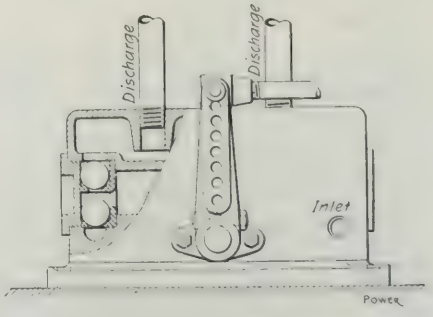


FIG. 2. VALVE ARRANGEMENT

which is reciprocated by a gear segment fastened to a shaft extending through the pump and having at its outer end a lever connected to some moving part of the engine. The operating lever has eight adjusting holes to control the amount of oil pumped, and it is also adjustable on the shaft which it drives so that it may be set at any angle. It is, therefore, possible to arrange the movement of the main pumping plugner so that it will travel from its extreme position at the end of the stroke inwardly much or little, so that the pump may just comfortably handle the amount of dirty oil which it received without sucking large quantities of air through the receiving well of the dirty-oil system.

The driving rod connecting from the engine to the pump is ⅜-in. pipe.

The pump casting has a hollow portion in the top acting as an air cushion or dome for the discharge of both ends and the steel ball valves, Fig. 2, at either end are accessible without disturbing any piping by removing their seats. The pump is manufactured by the Sterling Machine Co., Norwich, Conn.

Horsepower for Blowers

In a rule for estimating the approximate power required to displace a given amount of air at a given pressure under ordinary conditions, a certain blower manufacturer says: "Multiply the number of cubic feet delivered per minute by the pressure in ounces per square inch at the blower and the product by 0.003; divide by 11 and add 25 per cent. to the quotient."

Instead of multiplying by 0.003, then dividing by 11 and adding 25 per cent., why not multiply at once by

$$0.003 \times 1.25 \div 11 = 0.000341$$

Boiler Furnace Efficiency

On Tuesday evening, July 2, W. A. T. Willink read a paper on this subject before the Institute of Operating Engineers at its quarters, 29 West Thirty-ninth St., New York City. Its purpose was to stimulate the institute to greater action in preventing efficiency losses in boiler practice. The question, the speaker said, should not be left for solution to the consulting engineer as the operating engineer was qualified and in favorable position to make the necessary tests and correct combustion troubles.

After dwelling on the elements of combustion, Mr. Willink emphasized the im-

portance of ascertaining the volume of escaping flue gases as well as their quality. It is important because the temperature of the gases depends upon the volume. It was pointed out that "every time the air supply is doubled, the percentage of CO_2 drops from 16 to 10 per cent., and the loss in fuel will be 4 per cent., while the loss in fuel due to a drop from 10 to 4 per cent. of CO_2 is 18 per cent. Furnace conditions grow worse in geometrical ratio of progress as we go down in the scale of carbon dioxide percentages."

Mr. Willink next spoke of gas-analyzing apparatus, saying that the Orsat has the disadvantage of not being a recording apparatus. The analysis made by the Orsat is volumetric, and to get it by weight, multiply the percentages by volume by the molecular weight of the gas and divide by the sum of all products. An illustrated description of a simple and accurate recording flue-gas apparatus of German make which furnishes 40 analyses an hour, was very interesting, inasmuch as the apparatus consists chiefly of a number of intercommunicating siphons, which may be made by the engineer from pipe and pipe fittings. The suggestion was approved that blueprints be made of an apparatus patterned after the one described and distributed among the members that they might construct for themselves one devoid of troublesome complications.

After dwelling on the importance of recording the temperature and draft as well as the CO_2, Mr. Willink showed a drawing of a simple draft-recording instrument. To get the best results in combustion by draft there should be one gage to give the pressure at the bottom of the chimney where the flue enters; one to show the difference between this and the draft pressure above the grates, and a third to give the difference of pressure above and below the grate. A study of these pressures determines three prolific sources of waste in boiler practice, improper cleaning of the fires, obstructed flues and leaks in the masonry of the furnace.

Interesting discussions followed Mr. Willink's paper, in which all favored analyzing flue gases, but condemned, at least for the present, the awarding of bonuses to firemen for high percentages of CO_2. It was unanimously affirmed that bonus systems led to "jockeying" of the dampers and fires by the firemen, which, while producing high percentage of CO_2, lowered the furnace efficiency, increased the coal consumption and caused serious drops in steam pressure.

Mr. Willink responded by saying that if recording draft and temperature instruments in conjunction with the CO_2 recorder were used, this evil would cease, as those guilty would be immediately detected.

New Electric Syndicate

According to the press reports, the Westinghouse Electric & Manufacturing Co. and other interests identified with the Electric Properties Co. have, with Stone & Webster, formed a syndicate to take over the Electric Properties Co.

This last named company was formed in 1906 to acquire, finance and develop properties in which electricity is the motive force. It acquired the entire capital stock of Westinghouse, Church, Kerr & Co., and through that firm transacted a general engineering and construction business. The Electric Properties Co. had been controlled by the Westinghouse Machine Co.

Edison Co. Celebrates

As its contribution to New York City's "safe and sane Fourth," the Edison Co. furnished almost 100,000 lights, apportioned among the smaller parks. Private contractors did the wiring at the expense of the city. At City Hall alone there were 6000 lights. At Chelsea Park, where the trees are young, the enclosure only was outlined with lights. At Mount Morris Park, the southeast corner was illuminated, but the effect of a practical canopy of Japanese lanterns was striking. On Riverside Drive, thirteen consecutive trees were hung with lanterns in recognition of the original thirteen colonies.

Memphis Inspection Laws

The Board of Commissioners of the city of Memphis, Tenn., has adopted an ordinance to regulate the construction and inspection of steam boilers and generators and their appurtenances and connections, etc., which provides for internal inspection at least once a year and is to be enforced by a city boiler inspector and the police department. The rules attending to and made part of the ordinance appear to be those of the Massachusetts Board. F. H. Camble is boiler inspector.

Cannot Sell Current below Cost

Permission for the Groton, Mass., municipal light board to sell current at 10.8 cents per kilowatt-hour has been refused by the Massachusetts Gas & Electric Light Commission. The town wanted to sell current at 12 cents per kilowatt-hour for commercial lighting with a 10 per cent. discount. Groton supplies 69 private customers and its public buildings.

The commission states that as it has obviously cost the town more than 12 cents to supply current, the private customers must bear their full share of the burden which the town assumes. A supply for less than cost compels the public "to pay for the special advantages enjoyed by a few."

Explosion at Ruppert Plant

By the blowing out of a compressor head and the bursting of the big ammonia pipes in the refrigerating plant of the Jacob Ruppert brewery, New York City, on June 28, Max Taboer, the assistant engineer, was instantly killed, four other employees seriously injured and the plant slightly damaged. Forty employees whose work takes them into the plant escaped.

The plant had five large ammonia tanks and 15 miles of circulating pipe through all parts of the brewery.

The fire department had great difficulty in closing the feed pipes as the men had to travel through 125 ft. of the accumulating fumes, but in 20 min. after their arrival they succeeded in turning off six valves and returned to the open in safety.

The damage to the brewery itself was comparatively slight. Some of the piping in the power plant was loosened, but the five dynamos were uninjured and were set running a half hour after the accident. The cause of the explosion is unknown.

Cyclone Shuts Down Plant

Meager press reports from Regina, Saskatchewan, state that on June 30 a cyclone destroyed eight blocks in that city, putting the power plant out of commission. The wires are all down and the town is without light or power. Everything is being done to repair the damage. There are about 200 persons dead and injured, and it is estimated that the property loss will be nearly $2,000,000.

Canadian A. S. E. Convention

On July 23, 24 and 25, at Belleville, Ont., the twenty-third annual convention of the Canadian Association of Stationary Engineers will be held. The headquarters will be at the Hotel Quinte. Belleville Lodge No. 34, in conjunction with the national officers, has worked very hard for the success of the convention, and as the city is centrally located, it is expected that the attendance of delegates this year will be unusually large.

The local body has sent invitations to engineers, manufacturers, superintendents of plants and others interested to visit the mechanical exhibit, which will be held in connection with the convention. An elaborate program of entertainment has also been arranged.

The association has issued a handsome 100-page souvenir setting forth the many advantages of Belleville as a convention city and giving a history of the organization. It is well illustrated with portraits of the association officials and views of the many places of interest in and about Belleville.

Iowa N. A. S. E. Convention

The ninth annual convention of the Iowa State Association of the National Association of Stationary Engineers was held at Sioux City, on June 27, 28 and 29. The exhibit hall in the Auditorium was formally opened on Thursday afternoon, June 27, at 2 o'clock. At 8 p.m., G. B. Van Tassel gave an illustrated lecture on "Combustion," after which an informal smoker was attended by delegates and guests.

On Friday morning, Chairman F. E. Newkirk opened the first business session, at 9 a.m. Following the invocation, by the Rev. Smith, Mayor A. A. Smith welcomed the association, National Secretary Fred Raven responding. Harry Jones lectured on "The Indicator" and addresses were made by the officers and invited speakers and committees were appointed on credentials, ways and means, constitution and auditing.

At the afternoon session, W. A. Converse, chemical director of the Dearborn Drug & Chemical Works, gave an illustrated lecture on "Water for Boiler Use." Later, the delegates and visitors witnessed a baseball game between the engineers and supply men, at Riverside Park; the former won. In the evening a banquet was served at the pavilion.

At the Saturday morning session, the reports of the officers and committees were received. The constitution committee recommended changing the section relating to the calling of the annual state convention to read: "The convention shall be convened at the discretion of the state officers, the date to be announced in January of each year." The recommendation was ratified.

The special license-law committee recommended that the new committee be instructed to draft a license law and submit it to the national license-law committee for approval and support. This law, after approval by the national committee, is to be submitted to the state legislature.

C. W. Naylor, of Chicago, gave a lecture on "Station Costs and Operation," followed by brief talks by National-Secretary Raven, F. E. Ransley, president Central States Exhibitors Association; Abner Davis, F. E. Newkirk, W. L. Osborne, R. E. Hills, W. G. Graham, H. M. Jones and F. M. Williamson.

At the afternoon session the following officers were elected: President, A. E. Powell, Burlington; vice-president, John Devine, Des Moines; secretary, Abner Davis, Cedar Rapids, reëlected; treasurer, Robert Mullen, Marshalltown; conductor, C. E. Goodwin, Burlington; doorkeeper, A. R. Mutton, Waterloo; state deputy, James A. Coulson, Burlington. Secretary Raven installed the officers. On the recommendation of A. E. Powell, Burlington was chosen for the next state convention.

The exhibit hall was unusually well adapted to exhibition purposes and the various booths of the members of the Central States Exhibitors Association, which had charge of the hall, were well arranged and fully deserved the attention which the delegates and visitors bestowed upon them. The following exhibited:

The Exhibitors

American Steam Gauge & Valve Manufacturing Co., V. D. Anderson Co., Crandall Packing Co., Dearbon Drug & Chemical Works, Garlock Packing Co., Greene, Tweed & Co., Hawk-Eye Compound Co., Hills-McCanna Co., Home Rubber Co., Jenkins Brothers, H. W. Johns-Manville Co., Keystone Lubricating Co., Lunkenheimer Co., Osborne Valve & Joint Co., William Powell Co., Power, Schaeffer & Budenberg Manufacturing Co., C. E. Squires Co., United States Graphite Co., Murray Iron Works Co., Anchor Packing Co., the Under-Feed Stoker Co. of America, Valvoline Oil Co., Sioux City Air Compressor Co., Western Boiler Compound Co., George B. Carpenter Co., Quaker City Rubber Co., Interstate Electric & Manufacturing Co., Commercial Lubricating Co., Western Valve Co.

SOCIETY NOTE

The German Masonic Engineers Association of New York, with a membership of 75, will apply for a charter at the coming convention of the Universal Craftsmen at St. Louis. There are about 500 German-speaking engineers connected with the fraternity in New York City.

OBITUARY

Andrew E. Stevens

On June 23, Andrew E. Stevens, manager of the Consumers Power Co., Minot, N. D., was so badly injured in an automobile accident that he died within three hours after reaching the hospital.

He was born in Rushford, Minn., Feb. 7, 1869, and was educated in the Winona public schools and in the University of Minnesota.

Mr. Stevens had been in the employ of H. M. Byllesby & Co. since January, 1911. He had several years' experience in shop and street-railway work in Minneapolis and St. Paul, and had managed several telephone exchanges for the Wisconsin Telephone Co. For several years prior to engaging with Byllesby & Co. he was on the sales force of the Fort Wayne Electric Works.

Mr. Stevens was a member of the Beta Theta Pi fraternity. In Minot he was an active member of the Commercial, Elks and Minot Gun clubs.

Mr. Stevens is survived by his widow, a son, mother, two brothers and a sister. The funeral was held Tuesday afternoon, June 25, from the home of Mr. Stevens' mother, 905 West Franklin Ave., Minneapolis.

PERSONAL

Charles A. Harmony, municipal electrician at Centralia, Wash., has been appointed district superintendent for the Washington-Oregon Corporation.

Frank B. Goss, who has been with the B. F. Sturtevant Co. for the past seven years, is now sales manager of the Blaisdell-Canady Co., New York City.

Howard Lancaster has resigned his position as city electrician of Paris, Ky., to become superintendent of the telephone service of the Lexington & Eastern R.R., Lexington, Ky.

C. M. Rogers, who was recently appointed combustion engineer of the Empire District Electric Co., Joplin, Mo., is just recovering from a six weeks' attack of typhoid fever.

E. L. Murphy has been appointed to succeed F. A. Buss as the Western representative of the Ohio Brass Co., Mansfield, Ohio, with offices in the Fisher Building, Chicago.

Robert R. Keith, formerly with the Sterling Machine Co., Norwich, Conn., has accepted the position of assistant superintendent with the Sheffield Car Co., Three Rivers, Mich.

Reuben Bowen, connected with the International Steam Pump Co. for several years, latterly as district and department sales manager, has resigned to accept the position of general sales manager of the Epping-Carpenter Co., Pittsburgh, Penn.

Oscar M. Frykman has been appointed city electrician of Minneapolis, Minn., succeeding F. G. Dustin, who resigned to enter the commercial field as manager of the Northern Electrical Co., Minneapolis. Mr. Frykman is an electrical man of experience and has served as first assistant in the city department during the last three years.

Charles B. Moore, formerly vice-president of the American Arch Co., of New York City, and inventor of a number of locomotive devices, has been elected a vice-president of the Jacobs-Shupert U. S. Firebox Co., of Coatesville, Penn. Mr. Moore will have charge of the Western offices of the company, in the Railway Exchange Building, Chicago, Ill.

E. J. Wehrly and George R. Lyon, respectively president and vice-president and treasurer of the newly organized Power Equipment Co., which succeeds the power-plant department of the Arbuckle-Ryan Co., Toledo, Ohio, have been associated with the latter concern for several years. All unfinished power-plant construction contracts of the Arbuckle-Ryan Co. have been turned over to the Power Equipment Co., which will continue as contracting engineers for the installation of complete power plants, and as sales agents for the Stirling water-tube boiler and the C. & G. Cooper Co. Corliss and uniflow types of steam engines. The offices of the new company are at 305-307 Nasby Building.

Moments with the Ad. Editor

It is only occasionally when we stop to think that we realize what a lot of genuine fun we can get out of our work.

Of course every one can readily see the pleasure artists of all kinds take in their pursuits —the pleasure a painter gets from his picture, the sculptor from his statue, the composer when the band crashes out the stirring music that is the fruit of his toil.

But what we are more directly concerned with right here is the pleasure the every day man can get out of his work—the farmer from tilling the soil, the business man in closing a deal, you—the engineer—in keeping your plant at top-notch efficiency.

When the philosopher said, "So much as you put into your work so much it will return you," he was not trying to put a lot of meaningless high brow stuff over on us.

Any man who has done a day's honest work knows the remarkable simplicity and the vital truth of that saying.

The better you do your work, the better you like it—the better satisfied you are with your position and with yourself.

Doing our work with interest, working heartily, cheerfully, steadfastly and giving it the best that is in us robs our toil of a great deal of the laboriousness it may contain.

There is also a very practical side to this question. Take your own profession.

If an engineer has but little interest in his plant, if he does only what is absolutely required of him and no more, if he "holds down" his job and that is about all, it's a foregone conclusion that he can't advance very rapidly nor far.

On the other hand if he be alert to seize every chance for improving his plant, for adding to its efficiency in some particular, for eliminating any error that causes waste, a certainty it is the man higher up will notice it.

The employer who sees his engineer taking pleasure in his work will be far more ready to grant increased wages than would the employer who sees his man do his work merely as a matter of duty.

Advertising has directly helped many a man to strengthen his position, to qualify himself for advancement and to increase his interest in his job.

If you see an article advertised and know that it can improve your plant, if you root for its installation, and finally see it on duty in your engine room giving the kind of service you expected, you'll get the same kind of pleasure out of the experience that any good workman, no matter what his work, from banking to shoveling, gets out of a job well done.

That is one reason why it pays to keep in constant touch with the advertising pages of "Power." Here are advertised devices and appliances designed solely for the purpose of cutting-down the cost of operating the power plant and increasing its efficiency.

These pages, rightly used, offer the engineer an opportunity to improve the plant that is in his charge, to advance in his profession and to add vastly to the pleasure he can get out of his work.

Get the habit of studying them every week and you'll soon appreciate the truth of this.

POWER

Vol. 36 NEW YORK, JULY 23, 1912 No. 4

THE $50 PRIZE

for the best article on

The Operation of a Steam Turbine

has been awarded to

PAUL HOFFMAN

of Easton, Pa.

The Judges:

H. G. STOTT,
Sup't Motive Power, Interborough Rapid Transit Ry.

J. D. ANDREW,
Sup't Power Stations, Boston Elevated Ry. Co.

OLAFE OLESON
Chief Engineer Commonwealth Edison Co., Chicago

have unanimously decided that of the 46 articles submitted his was the best.

They also agree that that submitted by F. R. Brosius, of Columbus, Ohio, is second in merit.

Mr. Hoffman's article will appear in full in an early issue. The others will be used more or less extensively, or in so far as they contain different ideas and suggestions, and all the authors, including the prize winner, will be compensated in proportion to the use made of their material.

Old English Power Plants

By John S. Leese

Some very old engines and boilers which have passed the century mark are described. A few are still in operation, using coal which is a byproduct of the clay which they hoist.

While a description of what is undoubtedly the oldest operating steam plant in the world cannot fail to interest power-plant men, it will surprise many present-day engineers to learn that its engines and boilers are still performing their everyday service satisfactorily and reliably.

The writer visited this plant of the Madeley Wood Co., at Iron Bridge, Shropshire, England. The bridge from which the town takes its name is itself of considerable historical interest to engineers. It spans the River Severn, and was constructed in 1779 by Abraham Darby, of Coalbrookdale Works. It is the first iron bridge of record and has a single arch of 120-ft. span. Describing the bridge, Robert Stephenson said: "If we consider that the manipulation of cast iron was then in its infancy, a bridge of such dimensions was doubtless a bold as well as an original undertaking, and the efficiency of its details is worthy of the boldness of its conception."

Fig. 1 shows the plant of the Madeley Wood Co. This spot is known as the Wharf Pit. The engine is used solely for pumping, and three boilers, two of the haystack and one of the egg-ended type, supply it with steam. It was impossible to obtain a picture of the engine itself, owing to the darkness and lack of space inside the engine house. This engine has an open-top steam cylinder, 26 in. in diameter, and has an 8-ft. stroke. The beam is of cast iron and has a working length of 20 ft., the "main gudgeon" being 10 ft. from each end.

At the water end of the engine are two working barrels, each of 9¾-in. bore, working from a depth of 300 ft., the lift accomplished by each barrel being 150 ft. The two haystack boilers are 16x14 ft. and the egg-ended boiler is 6 ft. in diameter by 28 ft. long. This installation works at a steam pressure of from 8 to 10 lb. The boilers are constructed of ⅜-in. cast-iron plates with single-riveted lap seams; the pitch of the rivets is 1¾ in. center to center. The boilers have 4-in. bore, mushroom-type safety valves. The coal consumption of the pumping set is 1½ tons of slack coal per day, working continuously, which is not high. The equipment is approximately 115 years old.

Beyond the company's office are three blast furnaces, the lining of one of which is 27 years old and is still in good condition. Beyond the furnaces are the engines shown in Figs. 2 and 3. The engine in Fig. 2 was installed in 1779; its working life terminated in 1889. This engine was used for clay-pit winding work and has an oak crank web. The man with the shovel was the fireman at this installation during the latter years of its useful life.

Fig. 3 is a view of the Blests Hill Clay Pit winding engine. It is a "compound" engine having open-top "hot" and "cold" cylinders of 35 and 24 in. bore respectively. The "cold" cylinder is shown at the right of the picture. The beam is 14 ft. long from the end to the connecting-rod journal. The flywheel is 13 ft. in diameter, and the crankshaft is geared to a second-motion shaft with a speed reduction of 4 to 1. The winding sheaves are mounted on the second-motion shaft. The chains, clearly shown, are built of wrought-iron links with wooden packing or distance pieces. They sustain loads of between 18 and 20 cwt., and the lift now is 300 ft.

A cast-iron haystack boiler, seen in the background, is 10x12 ft. and has ⅜-in. plates and single-riveted lap seams. Between 8 and 10 lb. is the pressure carried. The safety valve is of the mushroom type and has a 4-in. bore. This engine consumes 16 cwt. of slack coal per day and was installed 105 years ago.

Further on, is the Shaws Pit winding installation, Fig. 4, which includes the winding gear, engine house, boilers and smokestack. This engine lifts 18 to 25 cwt. 660 ft. The engine has a single 36-in. cylinder with a 4-ft. stroke, and the working length of the beam is 14 ft. 5 in. The 15-ft. flywheel is driven with a gear reduction of 4 to 1 from the crankshaft to the second-motion shaft. Steam is supplied at 4 to 6 lb. pressure by a 10x12-ft. haystack boiler and a 5x20-ft. egg-ended boiler, both equipped with mushroom-type safety valves. The coal consumption of this engine is unknown, the coal used being that raised with the clay. This installation dates back about a century. The driver, shown in the photograph, is 72 years old and succeeded his father as the driver of this engine.

In Fig. 5, a view through the engine-room door of the Shaws Pit engine house, may be seen the connecting-rod, crank web, pressure gage, pump rod and valve-gear.

Fig. 6 was taken from the platform shown in Fig. 5 and gives a clearer view of the valve-gear. The valve rod and roller tappet are plainly visible, as are the controlling lever and the seat where the engine driver sits in full view of the pit mouth.

Fig. 1. Plant of the Madeley Wood Co.

Fig. 2. An Installation of 1779

Both the Blests Hill Pit and the Shaws Pit engines have overwinding alarm bells in addition to the dial indicators inside the houses. The alarm bell, seen in Fig. 4 through one of the spaces between the flywheel arms, is operated by a knocker arm which is rotated and moved axially by the scond-motion shaft so as to act at a determined height and depth. Reaching diagonally upward from the base of the alarm-bell pedestal is the rod operating the winding indicator inside the engine house. In Fig. 3 the bell can be seen through the arm of the second-motion shaft pinion, also the indicator rod which passes through the wood wall of the engine house.

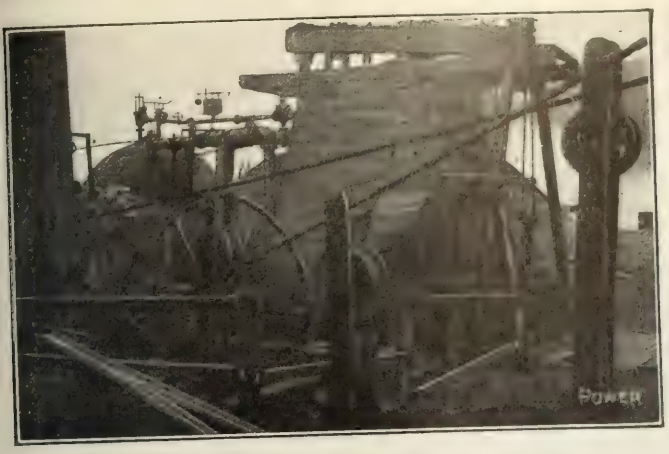


FIG. 3. AN ANCIENT WINDING ENGINE

FIG. 4. SHAWSPIT WINDING INSTALLATION

All these engines are of Newcomen type, water being injected into the cylinder to condense the steam below the piston at the end of each upstroke. The pistons are packed with rope and have a water-seal maintained 2 in. deep upon their upper surfaces. The engines run at from 8 to 16 strokes per minute and operate very smoothly. The beams, shown in Figs. 2, 3 and 4, and the supporting gantries are of oak.

Who built the engines is not known, but the Wharf Pit engine is of the genuine Newcomen type. The Blests Hill and Shaws Pit engines are locally known as "Adams" Newcomen engines.

The Newcomen engine in the South Kensington Museum is supposed to be fairly old, but the Madeley Wood Co.'s installations are said to be older still; moreover, they are still running.

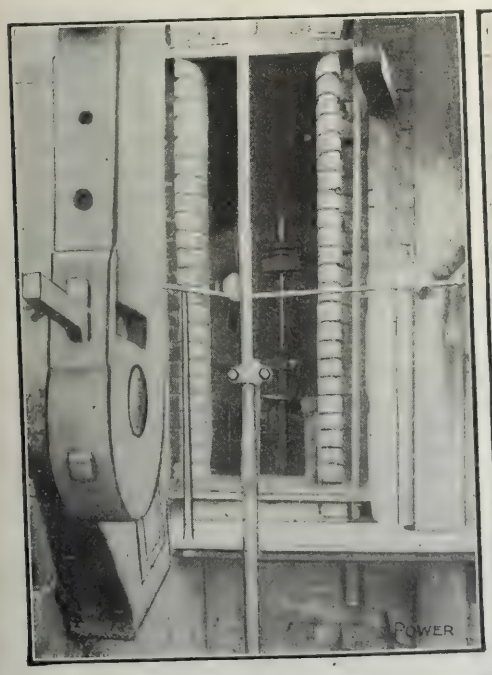


FIG. 5. SHOWING CONNECTING ROD OF THE SHAWSPIT ENGINE

FIG. 6. SHOWING CONTROLLING LEVER OF THE SHAWSPIT ENGINE

The writer is indebted to the chief engineer and his staff for the foregoing details of this article.

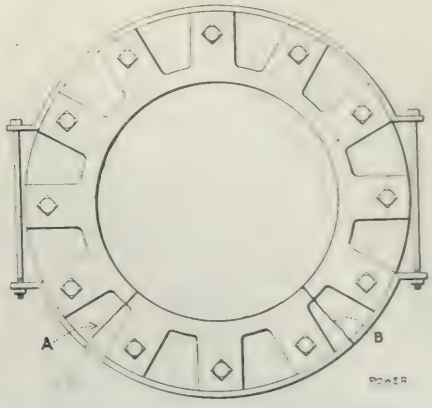
Nearly 32,000 tons of graphite is imported by Germany annually. The average price of American graphite is $135.66 a ton at Hamburg.

Reinforced Broken Flange

By W. H. Wakeman

One of the flanges on a 12-in. pipe line began to leak steam. An investigation showed that a section had broken out at *A* and *B*, but was held in place by three flange bolts. The pressure on this line is usually about 4 lb.; it is never more than 8 lb. After the break was discovered the pressure was reduced to 2 lb.

The external diameter of the flange is 19 in. Two ⅝x1½-in. iron band clamps were made to fit around it as shown, so made that when put in place the ends were 6 in. apart. Holes 1⅛ in. in diameter were bored though these projecting ends, and when ⅝x9-in. bolts were inserted in them they cleared the flange about ¼ in.



HOW THE BROKEN FLANGE WAS REINFORCED

When the clamps were put in place a piece of sheet packing was put between the lower clamp and the broken piece. After the packing was thoroughly heated the nuts were again tightened and once again the next morning.

Nothing more was done to the flange, and it is now perfectly tight. The clamps cost less than $3, making a very cheap repair job and yet one that is entirely satisfactory.

Duchesne's Experiments on Superheat

V. Dwelshauvers-Dery*

For saturated steam the author finds a substantial agreement between the temperatures of the steam tables, those indicated by mercurial thermometers and those shown by the very sensitive thermo-couple which he uses. For superheated steam, however, there is a wide difference, increasing with the degree of superheat, the mercurial thermometers reading in some cases 70 or 80 degrees below the thermo-couple. In a mass of superheated steam there may be strata of different temperatures and even moisture. This calls into question all the determinations upon superheat in which mercurial thermometers have been used. The variation of the specific heat of superheated steam is determined by the author and it is shown that as the pressure increases all its properties become independent of the pressure and vary only as a function of the temperature.

*Professor emeritus, University of Liége.

The observant experimenter frequently finds out things in the course of an investigation for which he had no thought of looking. The work of the clever investigator, Armand Duchesne, instructor at the University of Liége, Belgium, affords a new example. He started out to study the variation of the temperature of the steam and the metal of the containing cylinder of a steam engine during the period of compression.

As during this period the steam is generally superheated, and a poor conductor, while the temperature varies at each instant, it was necessary that he should have a thermometer, so to speak, without calorific capacity. This necessity resulted in his invention of what he calls a "hyperthermometer," an electric thermocouple of which the platinum and silver wires have a diameter of $^1/_{100}$ of a millimeter (about 0.0004 of an inch) and present numerous junctions disseminated throughout the space of which one wishes to determine the temperature in such a way as to give the average temperature instantaneously or nearly so.

It is not all to create a thermometer. It was necessary still to calibrate it with all the exactitude possible; the more so as the author wished also to verify the law

$$\frac{pv}{T} = constant$$

for the superheated vapor of water, by operating with high compression in the cylinder of a steam engine. This would be the second time that an apparatus of the size of a steam engine should serve to determine physical laws, with as much as, if not more, precision than the diminutive apparatus of the ordinary laboratory. The first time was when Hirn attempted to determine the mechanical equivalent of heat by means of the Logelbach engine, with a capacity of 150 hp.

It was in the calibration of his hyperthermometer that Monsieur Duchesne first encountered the unexpected. Comparison with a mercury thermometer in the same current of steam was the only available method. Steam was made to flow slowly through a large pipe furnished with a thermometer close to the boiler, called the auxiliary thermometer t_a, passing through the metal of the pipe by means of a stuffing-box and plunged in the current of steam. On the same pipe was a second mercury thermometer t_v placed in a thermometer cup screwed into the pipe and filled with oil. Finally came the hyperthermometer, or thermocouple of silver and platinum, arranged as shown in Fig. 1.

Saturated steam was used, the only medium which can furnish a given temperature constant and uniform at all its points. The pressure was shown by a large mercury column, and the temperature by the steam tables which was that of the hot junctions. The wires of platinum and of silver communicated respectively with the large wires of the same metal, which issued by insulated stuffing-boxes from the steam space in which the instrument was used. They were connected to conductors which led to the galvanometer, and submerged in vessels containing melting ice. All of the apparatus was protected so far as possible against loss of heat by radiation.

Monsieur Duchesne gives in his treatise the results of six groups of experiments, of which the average temperatures, as given by the mercurial thermometer, t_v, were about 115, 129, 139, 150, 156 and 162 deg. C., differing from the tabular values of the temperatures of the saturated steam in which the thermometer was immersed by only a fraction of a degree.

When one operates with saturated steam the thermometer t_v at the end of a certain time indicates, therefore, a temperature of saturation close to the temperature marked by the auxiliary thermometer t_a, or determined by means of the pressure and the steam tables, and the hyperthermometer is in accord or nearly so. But when one operates with superheated steam this is no longer true. The thermometer t_v no longer accords with the auxiliary thermometer and the greater the superheat, the greater the lack of agreement.

Here is the explanation of this fact: Saturated steam gives up its heat freely, and consequently the heat of the mass of saturated steam passes in quantity across the thermometer cup and the oil to the thermometer t_v, and the internal surface of the oil cup is not cooled. But if the steam which surrounds the oil cup is superheated, it is only the film of steam adjacent to the exterior surfaces of the oil cup which participates in this exchange of heat, on account of the low conductivity of superheated steam. The thermometer t_v will, therefore, indicate a temperature inferior to that of the auxiliary thermometer.

Applying the same reasoning to the auxiliary thermometer, which is plunged directly in the steam, one is justified in concluding that it will not give exactly the temperature of the environment in which it is plunged if surrounded by superheated steam, and that it will be necessary, in order to determine this temperature exactly, to have a thermometer without calorific capacity.

The hyperthermometer of Duchesne responds practically to this condition, if it is properly calibrated. The calibration of the thermometer has been effected experimentally with the greatest care, the details of which are carefully set forth in the treatise.

It is necessary now to compare the indications of the simple mercury thermometer with those of the hyperthermometer of Duchesne. The experiments were made under varying pressures as follows: 17,084, 27,024, 36,228, 48,759, 56,762, 66,930, kilograms per square meter (from about 25 to about 95 lb. per square inch) and under each pressure with degrees of superheat varying from zero (saturation) to 173, 185, 117 and 98 deg. C. The differences between the indications of the hyperthermometer and the mercury thermometer increased in each case the more the superheat was augmented, and amount to 85, 79, 54, 55, 46 and 43 deg. for the foregoing pressures. The diagrams giving these differences are curious to study and show that they are governed by very definite laws.

Repeating these experiments with the apparatus more perfectly protected against heat radiation, there was obtained a notable attenuation of these differences as follows:

17,000	27,000	48,000	59,000	68,000	kg./m.²
8.1	30.6	49	39	33	deg.

These results were certainly unexpected.

Monsieur Duchesne, wishing to push to the utmost his tests on this important point, which may call into question all the experiments which have been made on superheated steam with mercury thermometers, plunged the mercury thermometer into the center of the mass of superheated vapor, passing it through the wall of the containing pipe by means of a stuffing-box. With this arrangement the differences diminish as follows:

18,000	28,700	38,600	49,200	59,500	70,100
7.6	6.1	3.5	3.3	3.1	2.0

These differences are thus reduced to the order of possible errors in the evaluation of the temperatures by the hyperthermometer, but they are always of the same sign, which justifies the belief that it is the latter instrument which gives the true temperature.

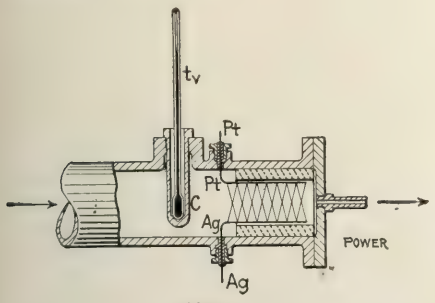


FIG. 1

The explanation given by Monsieur Duchesne of the difference between the temperatures indicated by the hyperthermometer and the thermometer t_v involves the hypothesis that in the body of the same mass of superheated steam are to be found contiguous layers or strata of steam of very different temperatures. Hirn had already remarked this. Monsieur Duchesne has demonstrated it experimentally; he has shown even that there may exist strata of saturated steam.

In brief, then, one first point is established, that with superheated steam all observations of temperature may be false if they are not made with a thermometer practically without mass.

In possession of an apparatus adapted to determine exactly the temperature of superheated steam, Monsieur Duchesne had the ambition to investigate experimentally the law of the compressibility of the superheated vapor of water; that is to say, the law which connects the pressure p, the absolute temperature T and the volume v' of a kilogram of that substance. Many authors have given formulas, a résumé of which may be found in the remarkable work, "La Vapeur d'Eau Surchauffée," by Monsieur Marchis, professor at the Sorbonne, of Paris. Other works are those of Zeuner, of Tumlirz, of Weyrauch and of Callendar, but none is directly based on experiment.

In the laboratory founded by the writer at the University of Liége, numerous diagrams had been taken by the indicator on the steam engine, operating with a high degree of compression. The greatest precaution had been taken to obtain a precision as absolute as possible in the measurement of the pressures and the volumes of steam during compression. The temperature at each instant had not been determined. But there is one quantity of the highest importance which experiment does not give directly; that is, the weight of steam in action during the compression, and consequently that which had been inclosed in the cylinder at the moment of the closure of the exhaust valve.

One could calculate the weight of pure steam inclosed at that moment, since the diagram gave its volume and its pressure, from which could be deduced the weight per cubic meter by means of the steam tables. But might not this steam be accompanied by a weight of condensed water deposited as dew on the metallic walls of the cylinder?

Hirn had assumed that this water did not exist, that the walls were dry and the steam saturated, without, however, furnishing any positive proof; but he had added that the question would not find its definite solution except in the study of very long compression. In fact, to George Duchesne, the older brother of Armand, is due the credit of having decided this point. By an intelligent analysis of the diagrams of compression he has shown that the hypothesis of humidity in the cylinder at the commencement of compression is untenable, because at that moment *the temperature of the metal is considerably higher than that of the steam;* that not only must the steam be dry and saturated, but that it is probably superheated; that without taking into account the slight superheating, the weight of vapor present can be regarded as equal to the product of the volume occupied in cubic meters by the weight of a cubic meter of saturated steam, as given in the tables at the pressure measured on the indicator diagram.

The diagram of the indicator, therefore, shows directly at each point of the compression curve the pressure p and the volume occupied by the steam. Indirectly, it permits the calculation of the weight of steam m acting during the compression, and consequently the volume v' per kilogram for each point. There remained only the temperature T to determine to establish the function

$$\frac{pv'}{T} = C$$

Monsieur Duchesne traced a diagram of this from the indications of his hyperthermometer, and he worked with unusual compression, up to nine-tenths of the stroke. From this collection of data he drew his conclusion with as much positiveness and genius as science. Such is the work resumed in his thesis, which has received the favorable recognition of the eminent professors of the Sorbonne at Paris, Messieurs Appel, Koenigs and Marchis.

The method adopted by Monsieur Duchesne for calculating the weight of fluid operating during the compression is particularly remarkable and worthy of consideration. He traces the curve of experimental temperatures during the compression. Upon it he superposes the curve of temperatures of saturation from the commencement of the compression. These curves at first are identical. Then they separate, the experimental curve mounting more rapidly than the other. It is at the point of separation of these two curves that the steam ceases to be saturated, and becomes superheated.

This point may, therefore, be taken as that of the coördinates of the curve which will furnish the element for the

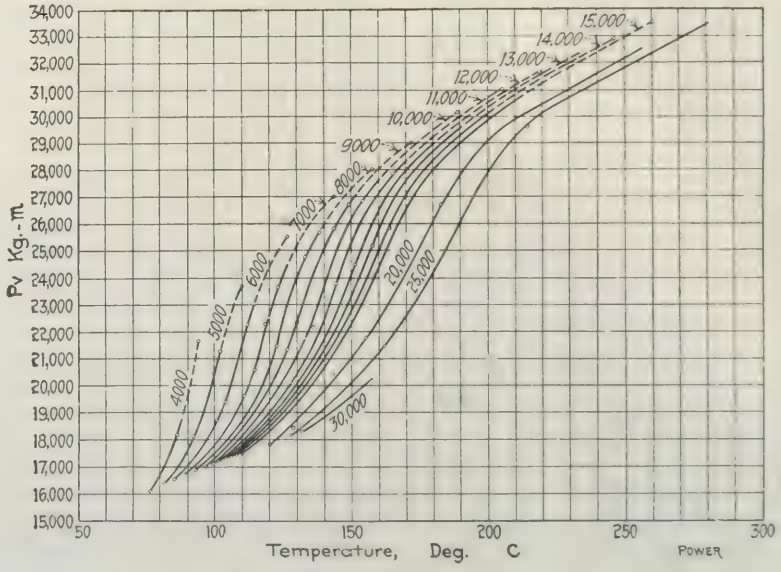


FIG. 2

calculation of the weight m of the steam operating during the compression; that is, the volume occupied in cubic meters and the weight of a cubic meter of saturated steam at the indicated pressure. If

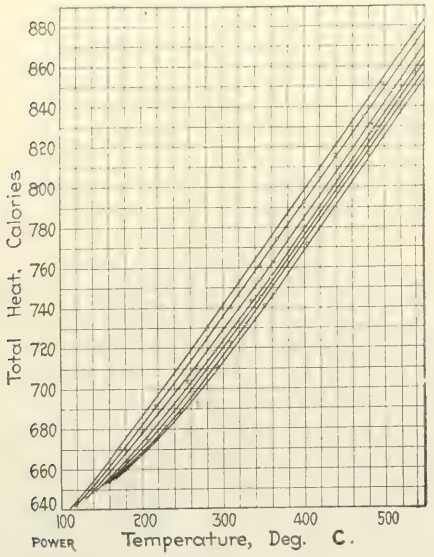


FIG. 3. CURVES OF TOTAL HEAT

this method does not give an absolutely certain result, it may in any case be said that it assures an approximation as great as may be hoped for in practice.

One may now trace, according to the table of experimental results, diagrams in which pv' and p are plotted as functions of the temperature. These tables and diagrams are included in the memoirs of Monsieur Armand Duchesne on pages 40 to 48. Here is reproduced only the curve deduced from the preceding, and giving for a constant pressure indicated on the curve itself as abscissas the temperatures t, and as ordinates the values of pv'. These curves illustrate, therefore, the equation

$$pv' = f(t)$$

each one for one value of p, Fig. 2.

An examination of this group of curves will show that each of them finishes by taking a rectilinear course, according with the increase of temperature and commencing with a certain value of t. And further, if the rectilinear portion of either of these lines is continued toward the left in a straight line, it will intersect the axis of temperatures at the point which marks the absolute zero; that is to say, at

$$p = -273 \text{ deg. C.}$$

which leads Monsieur Duchesne to the two following conclusions:

FIRST PROPOSITION: *For a given pressure, the superheated vapor of water obeys the law of Gay-Lussac beyond a certain limit of temperature; that is to say, that at constant pressure the product* pv' *is proportional to the absolute temperature* T. *Below this limit the law of compressibility is represented by a curve, which has a point of inflection, and of which the form corresponds fairly well to an equation of the third degree.*

SECOND PROPOSITION: *If the superheated vapor of water is at a temperature sufficiently above that of its point of saturation, its properties approach those of a perfect gas, and the greater its pressure the more this is so.*

Monsieur Duchesne has not given his results without having submitted them to thorough verification, both mathematical and experimental, which, however, would be too long to develop here. Among the properties of superheated steam he wished to determine with his habitual exactitude the total heat λ' of a kilogram of steam saturated at t and superheated to t'. He made superheated steam to flow for an hour at a temperature t' into a surface condenser, where a continuous current of water condensed and cooled it to a temperature $t'f$, taking from it the quantity of heat necessary to elevate its own temperature from ti to tf.

The tables give the heats of a kilogram of water at various temperatures, and notably $q'f$ for $t'f$, qi for ti, qf for tf. If during the hour there has flown from the boiler to the condenser a weight M_a of steam, which has been superheated from t to t', the quantity of heat carried to the condenser is $M_a\lambda'$. This heat has raised the temperature from ti to $t''f$, of the weight M_c of circulating water in the condenser which carries off, therefore, $M_c(q''f - qi)$. On the other hand, at the outlet of the condenser the condensate, in weight M_a, has preserved a temperature $t'f$ and a heat value $M_a q'f$. One has, therefore,

$$M_a\lambda' = M_c(q''f - qi) + M_a q'f + \text{the losses of all sorts.}$$

Whence

$$\lambda' = \frac{M_c}{M_a}(q''f - qi) + q'f + \frac{losses}{M_a}$$

In order to appreciate the method by which Monsieur Duchesne determined the losses, it will be necessary to read the memoir itself. The smaller quantity, M_a, the water coming from the condensation of the steam, was collected in a gaged receptacle provided with a thermometer. The gage gave the volume and, knowing the temperature, the weight per cubic meter could be easily determined.

The weight of the circulating water was determined by causing it to pass over a weir, accurately calibrated in advance. At short and equal intervals of time the height of the water level above the sill of the weir was observed, and the average was taken. The table belonging to the weir gives the volume of water which has passed over it per second or per minute, or per hour. A thermometer placed near the weir gives the average temperature of this water. From the tables can be deduced the weight per cubic meter, and consequently the weight of the volume past M_2. No experiment was commenced until uniform conditions had been thoroughly established, so that there should be no difference between the initial and final states.

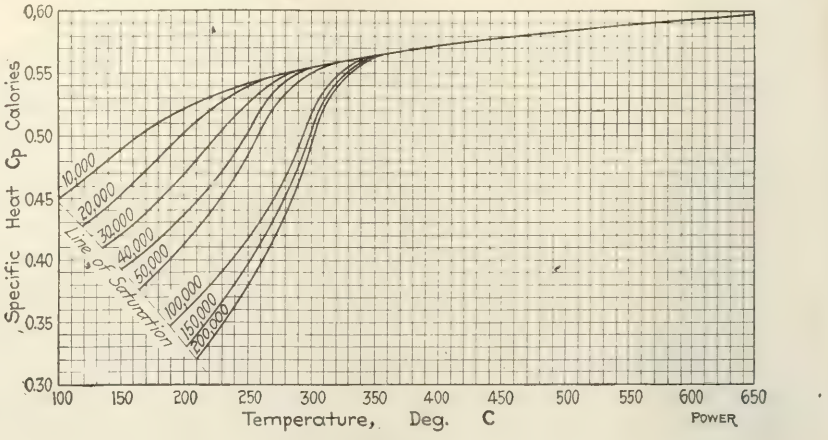


FIG. 4

The above described method was applied as follows: Several tests were made maintaining the pressure constant and varying the temperature of superheat, which gives for the various pressures the variations of the total heat of the superheated steam as a function of the temperature. As it was impossible to maintain the same pressure rigorously throughout all the tests of the same series the pressure has been calculated, assuming that whole series was made at the average pressure.

At the limit, the properties of slightly superheated steam must accord with those of saturated steam. For each curve there is obtained, therefore, one more point in calculating the total heat of vaporization, and the temperatures of saturation corresponding to the pressures. The mean pressures chosen were about 10,000 kilograms per square meter apart.

Monsieur Duchesne has devoted in all 65 experiments to this research of the total heat of superheated steam. The figures of Regnault relative to saturated steam at seven mean pressures furnished seven points, making 72 in all. Monsieur Duchesne has discarded eight of these, the results of which fell off of the line more than 0.8 per cent. There remained, therefore, 64 points.

In this way has been formed the group

of curves, Fig. 3, the tangent of which makes with the axis of temperatures an angle of which the tangent is equal to the specific heat at constant pressure C_p. It will be seen that as the pressure increases the curves of λ' approach each other more and more, so that for a sufficient pressure they will become identical; that is to say, the specific heat at constant pressure would become independent of the pressure whatever was the temperature.

This fact is perfectly explained by the diagram, Fig. 4, obtained in part by extrapolation, the ordinates of which are the values of C_p at different pressure for the temperatures indicated by the abscissas between 10,000 and 200,000 kg. per square meter of pressure. It will be seen that at about 360 deg. the specific heat C_p becomes nearly independent of the pressure and depends only upon the temperature with which it augments regularly.

The last chapter of this remarkable study of Monsieur Duchesne, in which he resumes his conclusions, calls attention to the divergencies which exist between his results and those relative to the values of C_p found experimentally by Messrs. Knoblauch and Jakob. These divergencies are explained by the fact that these gentlemen used in their experiments for the measurement of the temperature of superheated steam the mercury thermometer in the usual cup of oil, which is massive and slow in responding to changes of temperature. It follows that there is a considerable error in their estimation of the difference of the temperatures of superheat between which they have evaluated the mean specific heat at constant pressure C_p. This difference increases as the steam approaches the point of saturation. It tends to disappear toward 360 deg. C.

Two difficulties are to be found in the measurement of the temperature of superheated steam passing even slowly through a space, as of a pipe. First, the temperature is not uniform throughout the mass as it would be with saturated steam, on account of the slow conductibility of the superheated fluid. An instrument is necessary which gives the average temperature, as the piston of an indicator gives the average pressure, and instantaneously as does the indicator piston. In this connection it is safe to say that the mercury thermometer can give only the local temperature, and that tardily, especially if it is in a cup of oil. The hyperthermometer of Duchesne, with its numerous junctions disseminated through a large space and its excessively small mass sensible to heat, meets these conditions as far as is practically possible.

The importance of the effect of these two factors will be appreciated when it is remembered that in some of the experiments the difference between the mercury thermometer in its oil bath and those of the more responsive and sensitive thermocouple amounted to more than 80 deg. C.

Immense Power Plant Project

Significant of the progress of constructive operations in the inauguration of the largest electrification project since the Niagara power plants, is the recent execution of a contract by the Lehigh Navigation Electric Co., Philadelphia, Penn., with the General Electric Co. The contract calls for three 10,000-kv.-a., 11,000-volt, three-phase, 25-cycle, horizontal turbo-generating units with two 300-kw. turbo-exciters and one 300-kw. motor-generator exciter to be installed in the new main generating station at Hauto, Penn.

An initial expenditure of $3,000,000 is contemplated and an eventual investment of $10,000,000, with an assured supply capacity of 20,000 kw. from the first installation of the three 10,000-kw. turbo-generators and a gradual enlargement to 100,000 kw. capacity. This is said to be the beginning of the most important step ever taken in the United States toward the economic use of coal and generation of electrical power on a large scale using fuel.

Hauto is about 10 miles west of Mauch Chunk, at the mouth of a railroad tunnel leading from the Lehigh Coal & Navigation Co.'s main body of anthracite coal at Lansford. From here the distribution of current for heat, light and power by high-voltage transmission lines reaching out through all eastern Pennsylvania and New Jersey, will reach 2,500,000 people.

By transforming the coal into electric energy at the mines and transmitting it direct to consumers, handling, loading, hauling and dumping will be eliminated. At present the company has an annual waste of about 500,000 tons of coal particles and dust. As this culm is unmarketable, it will be burned in the company's boilers, thereby effecting another large saving.

The main generating plant was located at Hauto because of an abundant water supply, the reservoir being increased to 1,000,000,000 gal. capacity, with an area of about 400 acres by the construction of a higher dam. The water-supply needs of the ultimate plant will be some 300,000,000 gal. daily, most of which will be carried back into the reservoirs after condensation. To run the plant at the contemplated maximum capacity of 100,000 kw., approximately 1,000,000 tons of coal will be needed yearly, which would require 20,000 cars to carry it to market.

Among the first power consumers will be the slate and cement industries in Lehigh and Northampton Counties, all from 20 to 40 miles from the Hauto plant. Substations will probably be established at once at Coplay and Pen Argyl for stepping down and distributing the current to users. Within a radius of 50 to 75 miles from the main station are some 20 large cement mills. The transmission lines will be strung largely along the Lehigh Navigation Canal and on the right-of-way of the Lehigh & New England R.R. which the company controls.

A complete power census of the district reached by the distribution system of the first section of the Hauto plant shows that over 100,000 hp. is now used. The Lehigh Navigation Electric Co. has made up a schedule of prices, running from 8 mills to about 2½c. per kilowatt-hour, according to the power contracted for and the steady continuance of consumption. Current can be delivered cheaper than the industries can produce it by steam power.

As under Pennsylvania laws the right of an electric-power company operating with steam to distribute energy is confined to a single township, charters have been obtained in Lehigh, Schuylkill, Carbon, Northampton, Bucks and Montgomery Counties for about 60 electric companies. The first companies organized are the Hazard Co., Hanover; Oakdale Co., Salisbury; South Whitehall Co., South Whitehall; Equitable Co., Whitehall; North Whitehall Co., North Whitehall; Diamond Co., Washington; Upper Macungie Co., Upper Macungie, and Upper Milford Co., Upper Milford. All subsidiary companies will be consolidated under the management of the Lehigh Navigation Electric Co.

It is expected that the first installation at the Hauto plant will be in operation before the end of the year. As the plant capacity is gradually increased, energy will be carried to Allentown and Easton, Penn., Trenton, N. J., and finally into Philadelphia.

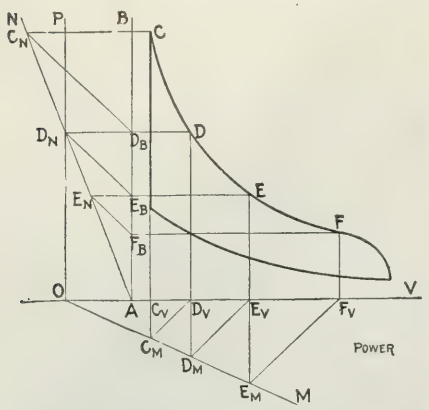
A New Analysis of Indicator Cards

In the article under the above title in the issue of June 18, page 859, an error was made in the drawing for Fig. 3 by not copying correctly the figures on the horizontal scale of volumes. On the right side of the diagram the figure 2 should be where 3 is, 3 where 4 is, 4 where 5 is and 5 should be at the end of the horizontal line. The curve is plotted to the correct scale.

The Modern Engineer is a little monthly magazine published by the Modern Science Club, of Brooklyn, and will supersede its year book. Editor Johnson says in its first issue that "the *Monthly* will be purely practical and written for engineers by engineers." POWER wishes it a long and useful life.

Determining Clearance from the Indicator Diagram

In his paper describing a new method of analysis for indicator diagrams, presented to the American Society of Mechanical Engineers at its recent spring meeting, J. Paul Clayton described the application of this method to the determination of the clearance. For the normal case the expansion and compression lines of the indicator diagram will become straight lines when plotted upon logarithmically ruled paper, but, in order that this may follow, the absolute zeros of pressures and of volume, that is, the axes of the curves, must be accurately determined. If the percentage of clearance and thus the position of the clearance line upon the indicator diagram are not known, different values for the clearance may be assumed and the diagram plotted upon the logarithmic paper with the assumed values. If the value is assumed too low the plotted curve will not become a straight line but will lean toward the axis of zero volume, while if too large a clearance is assumed the line will lean in the other direction. By trial with different assumptions for the



clearance, until a straight line is obtained, the actual clearance may be found when the absolute zero of pressure is known, or if the clearance is known accurately the method may be worked in the other direction to definitely determine the zero line of pressure or the barometric reading at the time that the indicator diagram was taken.

This method was illustrated and described in POWER, June 18, page 862.

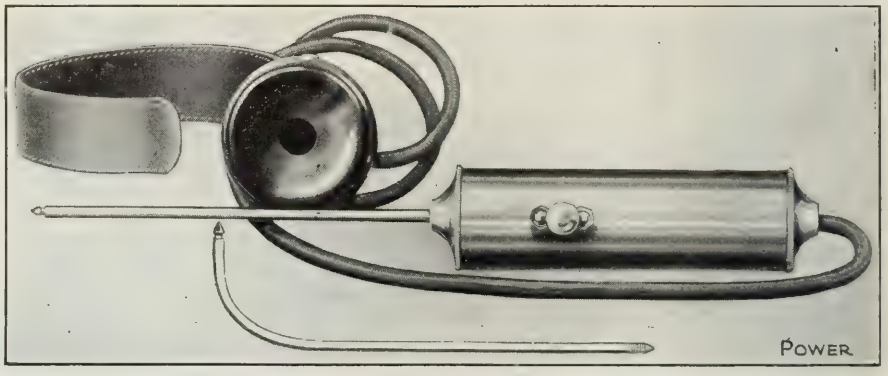
In discussing this paper, Prof. F. E. Cardullo, of the New Hampshire College, presented the method described in the accompanying diagram as being more elegant, more accurate and more simple than the method proposed in the paper. The proof of this method will be found on pages 33 to 35 of Prof. Cardullo's "Practical Thermo-dynamics."

In the figure, *CF* is the expansion line of a gas-engine diagram and *OV* the axis of zero absolute pressure. At any point, as at *A*, on this axis erect the perpendicular *AB* and draw *AN*, making any convenient angle to *AB*. Draw CC_n and draw C_nD_b making an angle of 45 deg. with *AB*. Through D_b draw D_nD and draw D_nE_b at 45 deg. Similarly, draw E_nE, E_nF_b and F_bF. From *F* drop a perpendicular to *OV* and draw F_vE_m, making an angle of 45 deg. with *OV*. From *E* drop a perpendicular which will intersect F_vE_m at E_m. Draw E_vD_m, dropping a perpendicular from *D* to determine D_m. Draw D_vC_m and drop a perpendicular from *C* to determine C_m. If the expansion line *CF* fulfills the equation $PV^n = C$, C_m, D_m and E_m will lie on the same straight line and that line will intersect *OV* at the point of zero volume, or the location of the line bounding the clearance.

Detectorphone

This is an instrument for detecting sound, whether water in pipes or noises of any kind in running machinery, which cannot be heard otherwise. It is a "mechanical stethoscope" and has been designed to locate trouble by applying it to a machine.

The device consists of two parts, a detector and a receiver. The pin of the detector is placed on any part of the machine suspected of derangement and any unnatural sound is transmitted to the receiver, which is held to the ear. This instrument is made by the Boston Talking Machine Co., 41 West St., Boston, Mass.



THE DETECTORPHONE, A MECHANICAL STETHOSCOPE

Circulator for Scotch Marine Boilers

A fault of the Scotch marine boiler is the lack of free circulation of the water, to assist which various circulating devices have been invented.

A device brought out in 1907 under foreign patents is being manufactured in this country by the Ross Schofield Co., 39 Cortlandt St., New York City.

Fig. 1 is a working model of a Scotch marine boiler used for demonstrating at the company's office. The barrel of the boiler is glass through which the circulation of the water is observed. The water currents indicate the path of circulation.

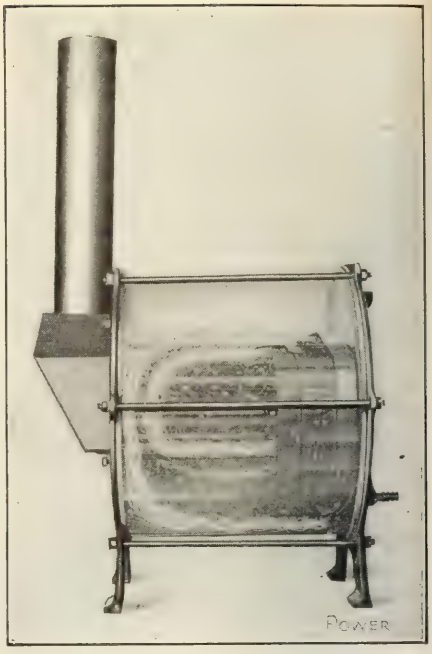


FIG. 1. DEMONSTRATING BOILER

The circulating system consists of a steel plate, supported on the short stays, completely inclosing the water space back of the combustion chamber except at the top and between the furnaces at the bottom; steel plates, supported by the short stays, not from the tubes as shown, placed in front of the combustion chambers and each side of each bank of tubes and back and front hoods. A side view of the combustion chamber and tubes, showing the application of the circulator, is shown in Fig. 2, and a front view of the device in Fig. 3. The circulator is installed without interfering with the boiler and produces a rapid circulation over all parts, increases the capacity of the boiler, lengthens its life, and improves the working conditions in every respect.

The arrangement of the plates provides for two spaces which operate in parallel, their superimposed curved hoods promoting longitudinal and elliptical circulation of the water by utilizing the natural force of the steam bubbles. These

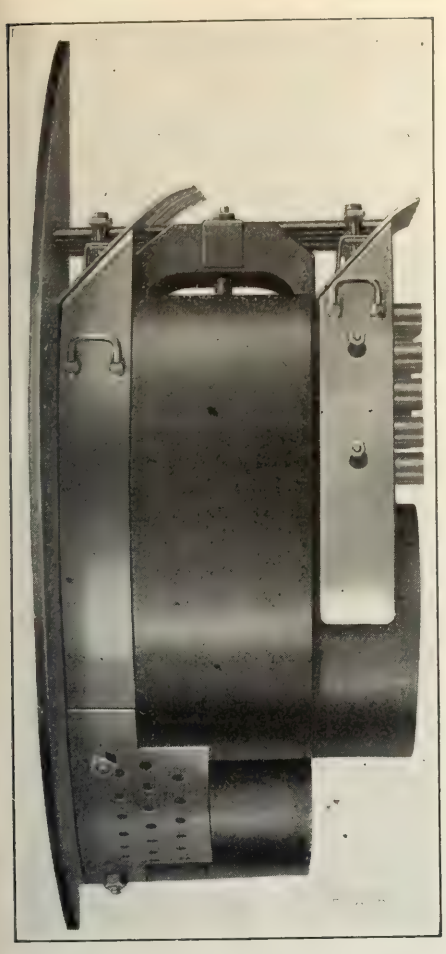


FIG. 2. SIDE VIEW OF THE CIRCULATOR

bubbles, on escaping from the centers of violent ebullition, pass up through the guiding compartments and issue under the curved hoods which deflect the ascending streams of water horizontally and prevent priming. The effect is the promotion of a flow that will continuously draw the water from the lower and colder part

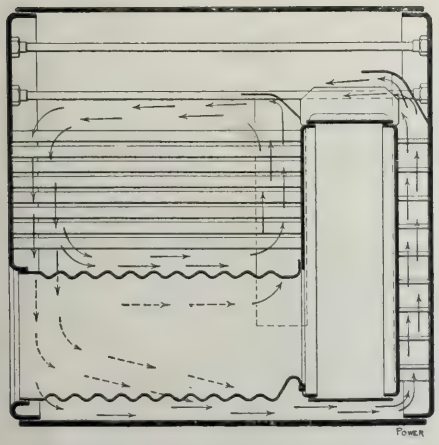


FIG. 4. SECTIONAL VIEW OF SCOTCH BOILER

of the boiler, up between the guiding plates and propel it longitudinally to the front. These conditions are quickly established, are constant, effective circulation is secured and regular and rapid steaming maintained. The sectional view, Fig. 4, indicates the position of the hoods and the path of circulation.

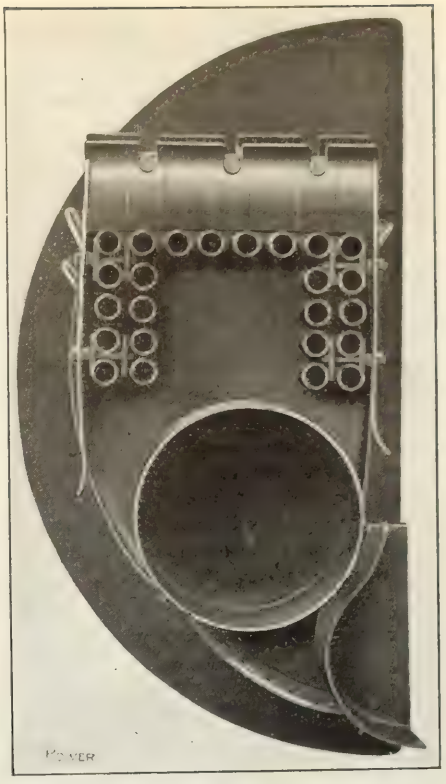


FIG. 3. FRONT VIEW OF CIRCULATOR FURNACE AND TUBES

Electric Heater Prevents Freezing

Electric heaters are used to prevent the water freezing in a large roof tank which supplies the fire-sprinkler system of a New York City loft building. When the system was put in experiments were made with electric heating, and it was found that heaters set in the 18-in. air chamber in the top of the tank, attached to the under side of the air-tight cover, gave the best results. Accordingly, four 1200-watt heaters were installed, with thermostatic control to keep the temperature in the tank from falling below 34 deg. F. Last winter the installation worked with perfect satisfaction, switching the current on and off automatically, and the tank was kept entirely free from ice.

Purchase of Coal on an Ash Basis

The principal impurities in coal are nitrogen, water and ash, all of which may be present in independently varying quantities. From this it would appear that any system of evaluating coal which considers the variation of only one of its three impurities must lead to erroneous conclusions, and while such assumption is correct in general, it is nevertheless true that the method of estimating coal values by ash content alone has, within certain limitations, decided advantages over the calorimeter method.

This is because coal from any particular vein is reasonably constant in its nitrogen and moisture content, however widely its ash may vary; and, moreover, different samples of coal from the same vein vary in commercial value in an inverse but much greater ratio than their variation in ash. It has been found in one series of tests reported in the *Journal of the Western Society of Engineers*, September, 1906, that coal containing 40 per cent. of ash will not generate high-pressure steam when burned in a boiler furnace. Its value is, therefore, nil, yet a calorimeter test would indicate that such a sample of coal has a value equal to 60 per cent. of that which a sample entirely free from ash would have. From this it is apparent that a formula which evaluates coal in inverse proportion to its ash content will give erroneous results.

Assuming that coal containing 40 per cent. of ash has no value as fuel, and assuming the rate of decrease of value is constant as the amount of ash increases, the value of a coal for varying amounts of ash would be represented by the formula:

$$Value = 100 \text{ per cent.} - (2\tfrac{1}{2} \times \text{per cent. ash})$$

But, as the ratio of fuel values to ash content is not a straight line but a curve, decreasing slowly at first and falling off rapidly for ash quantities greater than 30 per cent., a formula to represent the entire range would be quite an involved expression; but as coal containing more than 30 per cent. is rarely used, and as the curve for 5 to 30 per cent. ash is nearly a straight line, it can with a fair degree of accuracy be represented within the range mentioned by the formula:

$$Value = 100 \text{ per cent.} - (1\tfrac{1}{2} \times \text{per cent. ash})$$

which is a formula easy to remember and easy to apply and is much more accurate than the formula:

$$Value = 100 \text{ per cent.} - \text{per cent. ash}$$

In the excellent paper by L. P. Cecelius, entitled "The Purchase of Bituminous Coal under B.t.u. Specifications," which appeared in the report of the Committee on Power Generation, which was presented at the 1911 convention of the American Electric Railway Engineering Association, the author says:

> The presence of an excessive amount of ash in any coal manifests itself in the nature of a reduction of capacity, because of its occupying a relatively large amount of effective grate area with inert matter. It is also a source of additional expense in the matter of cost of removal, and extra wear and tear on ash-handling apparatus. It should therefore, be restricted to a certain limited amount, no premium being allowed for a minimum. Penalties for amounts in excess of the allowable limit should be provided, and these preferably should be-

come excessive after a certain value. The reason for this is principally that the amount of ash is, in a measure, the key to the nature of the coal, and the principal purpose of any coal specification should be to provide a steady and uniform supply of fuel, previously determined upon as best suited to existing plant conditions and to limit all other grades of coal.

This clearly indicates that the author realizes that penalties in direct proportion to the amount of ash would be inadequate to protect the consumer, and that deductions from a price fixed for a standard quality of coal should be in a greater ratio than any increase in amount of ash.—From report of Prime Movers Committee, N. E. L. A.

Chart for Calculating Boiler Tests

By F. O. Pahmeyer*

By use of the chart, the horsepower, actual evaporation per pound of fuel, equivalent evaporation and efficiency may be determined direct from the data of a boiler test, without going into laborious calculations. Examples are given illustrating the use of the chart.

*Engineer with Heine Safety Boiler Co.

The accompanying chart has been constructed to facilitate working up results from a boiler test, and the closeness with which results can be determined will be within the limits of the error of the observations making up the test records. Examples of the several processes of which determinations are made are as follows:

Capacity

Knowing the average steam pressure and the average feed-water temperature, the factor of evaporation can be ascertained from a table or by calculation. The only other item that must be known is the amount of water evaporated per hour.

From the point on the lower scale, representing the amount of water, follow upward the slightly inclined line to its intersection with the horizontal line representing the factor of evaporation found on the left-hand scale. Follow vertically through this point of intersection to the top scale, the reading of which will be the required capacity in boiler horsepower. One horsepower is defined as the evaporation of 34½ lb. of water from and at 212 deg. F, per hour.

By following down instead of up from the point of intersection, the pounds of water from and at 212 deg. per hour may be read off on the bottom line.

Example—Assume that 10,250 lb. of water per hour have actually been evaporated, that the average temperature of the feed water is 110 deg., and that the average steam pressure by gage is 140 lb. per square inch. The factor of evaporation for this temperature and pressure is 1.15. The point of intersection, as just described, carried to the top scale, reads practically 341 hp. This point of intersection, carried down to the bottom scale, reads practically 11,800 lb., which is the equivalent evaporation of water per hour from and at 212 deg. F.

Actual Evaporation per Pound of Fuel

From the point on the lower scale repsenting the amount of water, follow vertically to the intersection with the horizontal line representing the amount of coal. This intersection will fall on one of, or more likely between two of, the inclined lines representing the evaporation per pound of fuel. If between the lines, the reading may be interpolated.

Example—Assume that 10 250 lb. of water have been actually evaporated and 1580 lb. of coal have been fired; it is immaterial whether this is for one or more hours. The point of intersection, as previously found, falls practically on the line representing 6.5, which is the actual evaporation of water per pound of coal as fired.

Equivalent Evaporation from and at 212 Deg. per Pound of Fuel

The actual amount of water evaporated, the total amount of fuel burned, the average feed-water temperature, and the average gage pressure of the steam for any period must be known. Determine the factor of evaporation and find the equivalent evaporation from and at 212 deg. as described under "capacity." Follow vertically or down from this point of intersection to the horizontal line through the amount of fuel consumed, which point of intersection will fall on one of, or between two of, the much inclined lines, which represent the water evaporated per pound of fuel.

Example—Assume that 10,250 lb. of water have actually been evaporated and that 1580 lb. of coal have been burned in the same period of time. Let the temperature of the feed water be 110 deg., and the steam pressure 140 lb. per square inch; these conditions corresponding to a factor of evaporation of 1.15. The equivalent evaporation from and at 212 deg., as found under "capacity," gives 11,800 lb., although it is not necessary to follow down to the bottom line. The intersection, with the amount of coal burned, falls between the lines representing 7.25 and 7.5, which would be estimated as 7.45, the amount of water evaporated from and at 212 deg. per pound of coal as fired. This same method may be followed to determine the evaporation per pound of dry coal if the moisture is determined and deducted from the total weight of coal, and as well for the evaporation per pound of combustible if the moisture and ash are deducted from the weight of coal as fired, and the remainder used as the amount of fuel burned.

Efficiency

From the point on the bottom scale representing B.t.u. per pound of fuel, follow vertically to the much inclined line representing the equivalent evaporation from and at 212 deg. per pound of fuel. From this point of intersection follow horizontally to the efficiency scale at the right-hand side.

Example—Assume that the coal burned has a heat value of 12,100 B.t.u. per pound and that the equivalent evaporation from and at 212 deg. is 7.45 lb. of water, in accordance with the preceding examples. The point of intersection will be on a line, which will follow horizontally so as to give a reading of 60 % on the efficiency scale.

Remarks

In the foregoing examples it is assumed that there is no moisture in the steam. If it is desired to take any possible entrainment into consideration that quantity must be determined and the total amount of water evaporated decreased by the amount of moisture so found, or correction can be made by reducing the factor of evaporation by the same percentage. For example, if the total amount of water evaporated is 10,250 lb. and the moisture in the steam is 2 per cent., the true amount of water evaporated would be

$$10{,}250 \times 0.98 = 10{,}045 \ lb.$$

which quantity should be used in place of 10,250 in the determinations. Or the factor of evaporation could be reduced by 2 per cent., making it 1.127, which should be used instead of 1.15. The final result would be the same in either case.

Where the quantities of water and fuel determined are beyond the limits of the diagram, results may be determined by dividing these amounts by two, three or any other factor that may be necessary, then proceed as before. To determine the capacity the result found must be multiplied by the same factor that was used individing. The results for evap-

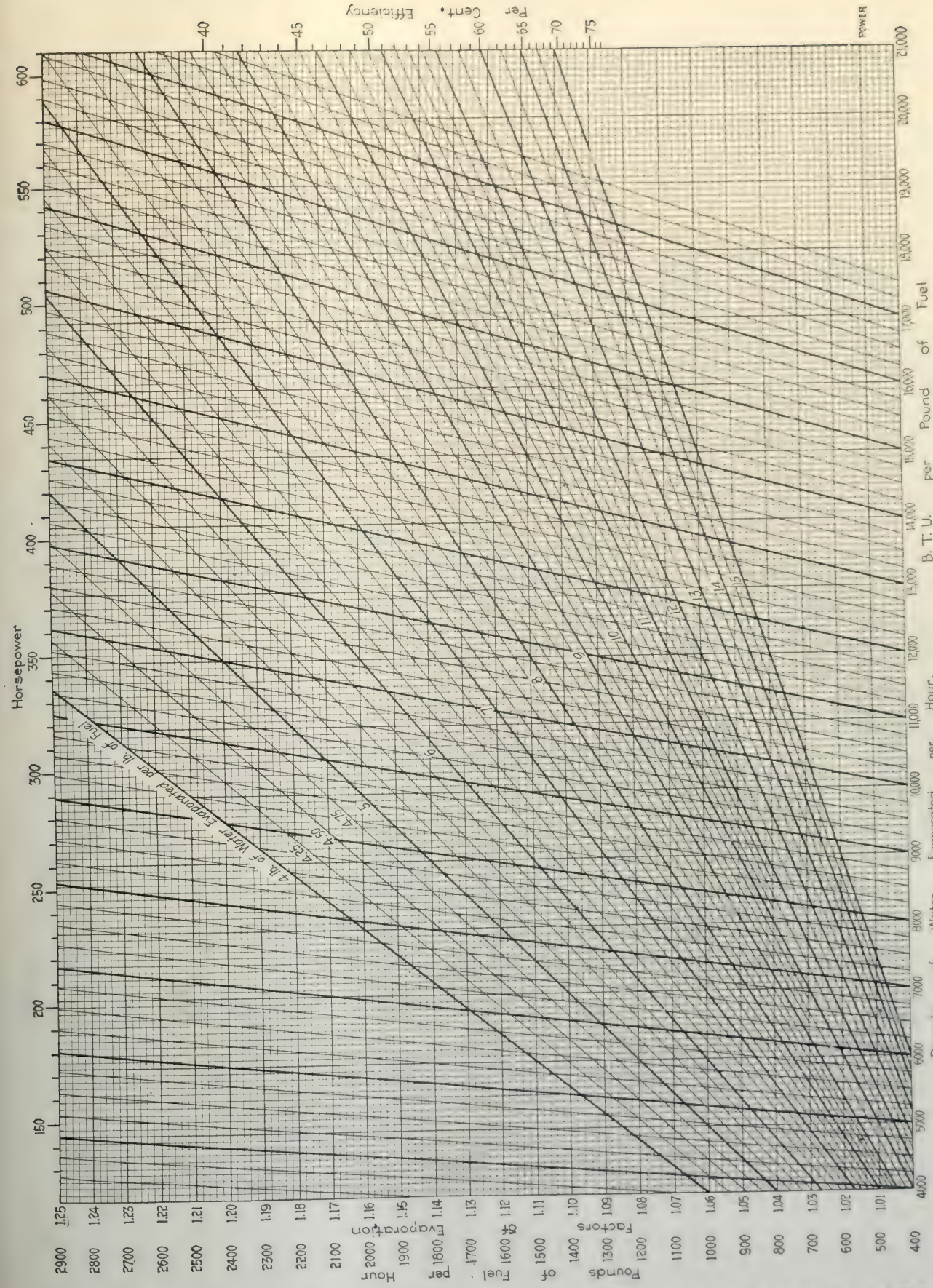


CHART FOR CALCULATING RESULTS FROM A BOILER TEST

oration per pound of fuel must not be again multiplied, presuming, of course, that the fuel is reduced in the same proportion as the water.

The chart will give results sufficiently close for all but very refined determinations, and will be found of great assistance in quickly arriving at results during a test, and for the use of those who have not the time or detailed knowledge to perform the various calculations involved.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Setting Motor Brushes

By J. S. Kinney

In reassembling direct-current motors after repairs have been made, the brush-holder arm is often moved, accidently or otherwise, so that the correct position has to be found before the motor will give good commutation. The correct position of the rocker-arm is marked on some but not on all motors. There are several ways to set motor brushes, but generally they are set, either mechanically by the eye, or electrically by the "kick method."

Interpole motor brushes must be very accurately set to obtain satisfactory operation. If the armature is parallel wound

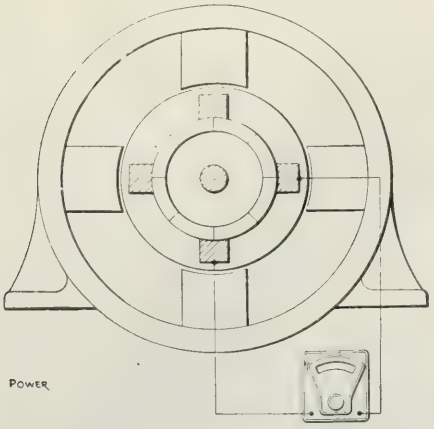


Fig. 1. Voltmeter Connection for Determinating Neutral Position by "Kick Method"

and the winding is sufficiently open to enable the armature conductors to be readily traced, the brushes should be set so that a coil, the two halves of which are directly under the centers of two consecutive interpoles, will have its ends connected to two adjacent commutator bars which are centrally located under the brush that is being adjusted. This armature coil, together with its commutator bars, can be marked and the remaining brushes set by rotating the armature until the marked coil comes under the next interpole and the last one of the first pair, and so on until all the brushes have been set. If there are several turns per coil, the middle turn should be used.

With wave-wound machines, the spread of the coils, or the winding pitch, as it is called, is not the same as the angular pitch of the poles. With this style of winding, which is quite common on small machines and necessary on multipolar machines having but two sets of brushes, it is best to have one side of the coil slightly back of the middle of its interpole, and the other side of the coil a little ahead of its interpole. The commutator bars to which this coil is connected should be under adjacent brushes of the same polarity, with one brush of the opposite polarity between. Usually the number of commutator bars will not be an exact multiple of the number of

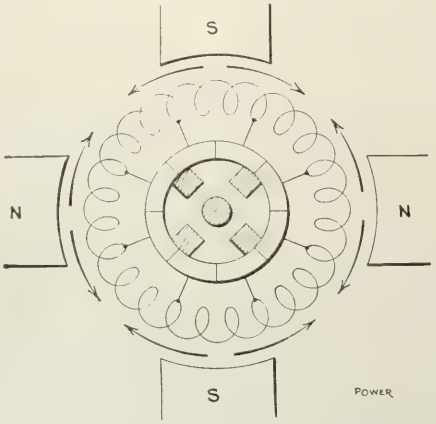


Fig. 2. Ring Wound Armature

poles, and the final position of the rocker-arm will have to be an average of neutrals found for each set of brushes.

The "kick method" is more often used to determine the no-load neutral position of the brushes. A low-reading voltmeter, of about 5 volts full-scale deflection, is connected to two adjacent brushes, as in Fig. 1. The field is excited with normal field current on small machines, and about 25 per cent. of normal field on large machines, and is "flashed" by alternately making and breaking the field circuit. If the brushes are not on neutral, the voltmeter needle will be deflected in one direction when the circuit is closed, and in the opposite direction when it is opened. The kick is greater when the field switch is opened, as the time element is very much less than when the switch is closed. The deflection obtained on opening the circuit should be used in setting the brushes. The rocker-arm is shifted while the field is being flashed, until a position is found in which no deflection is obtained. This should be tried with the meter leads connected to each pair of brushes, and the position of the rocker-arm which gives the best average is the correct setting. With a two-circuit wave-wound armature, a position of no deflection cannot be found because the number of commutator bars is not an exact multiple of the number of poles. In this case, the proper position is that which will give the minimum sum of readings.

Closing and opening the field circuit, with the armature stationary, causes

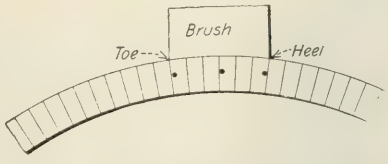


Fig. 3. Showing Where to Hold Voltmeter Leads

equal electromotive forces to be induced in the armature coils under adjacent poles. These forces oppose each other and neutralize at the correct brush position. A ring-wound armature, for example, is shown in Fig. 2. The neutral for this style of winding is half-way between the main poles. The arrows indicate the manner in which the electromotive forces oppose each other, and neutralize at the neutral position of the brushes. In the actual machine of today, the neutral is on the center line of the main poles.

An easy and convenient method of checking the neutral is to run the motor under full load, at its highest rated speed, in both directions. These speeds should agree within 2 per cent. Any difference in speeds may be decreased by shifting the brushes slightly in the direction in which the motor runs the faster. In an interpole machine, an armature conductor comes under a main pole and then an interpole of the same polarity, so that in reversing the direction of rotation, the armature and the interpole should be considered as a unit, and the current reversed in both, to maintain the proper relation between the main poles and interpoles.

It is well to see that the interpole is of the proper strength, if it has in any way been changed during the overhauling of the motor. One lead of the 5-volt voltmeter is attached to the brush-holder arm, and the other end is held against the end of the commutator under the brush, at each edge of the brush and in the middle. The dots in Fig. 3 indicate the proper positions for holding the lead. This should be done at each of the rated speeds of the motor, and at several loads. The interpole is of proper strength if

the three voltmeter readings are approximately the same; but if the voltmeter readings increase from the heel to the toe, the interpole is too strong. Where the motor has to carry heavy overloads, a slightly overcompensated field is desirable to take care of the saturation of the interpoles at the overloads. Removing one or two of the sheet-iron shims under the interpole, thereby increasing the bore, will weaken the interpole, or it may be weakened by removing some of the turns. Still another way is to shunt the interpole winding with a strip of german silver. Increasing the number of turns or decreasing the bore by the addition of shims will strengthen the interpole.

Exactly the same procedure is carried out in finding the flash neutral on non-interpole machines. Motors which are to be operated in one direction only are usually given a backward lead of from 1½ to 2 per cent. of the total number of commutator bars. A backward lead of from ¾ to 1 per cent. is given to variable-speed machines. The brushes are left on the flash neutral position on machines designed to operate in both directions, such as elevator and crane motors. Increasing the backward lead increases the speed of the motor, while decreasing the amount of lag slows it down. If a motor fails to come up to its rated speed

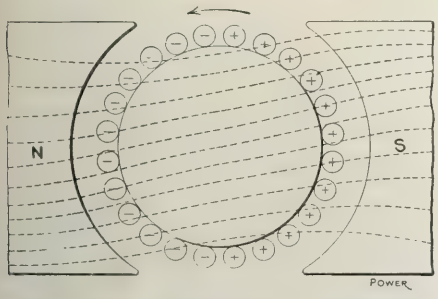


FIG. 4. FIELD DISTORTION BY ARMATURE REACTION

at full load and increasing the backward lead occasions excessive sparking, either the bore of the main poles must be increased by removing shims or the shunt field weakened by introducing external resistance in the field circuit. Often serious sparking is caused by the brushes not fitting, or becoming wedged in the brush-holders.

The distortion of the main field by armature reaction is illustrated in Fig. 4, and is due to the magnetic field produced by the armature acting at right angles to the main field. As the load on a motor increases, the armature current becomes greater, and consequently the armature flux is strengthened since it depends on the ampere turns. This changes the neutral and the brushes must be given more lag. In nearly all motors, the magnetizing force of the field is great as compared with that of the armature, which reduces to a minimum the amount that the neutral shifts.

A slightly greater amount of lag than that required by armature reaction is always given to the brushes to take care of the self-induction of the armature coils. This additional lag brings the coil in which current is to be commutated into a field of opposite polarity just before it comes under the brush, and the self-induction is taken care of and the coil commutated with minimum sparking.

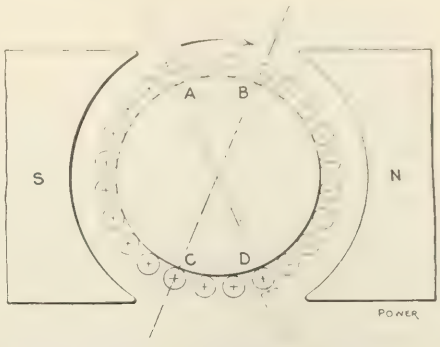


FIG. 5. SHOWING WHY MOTOR SPEEDS UP WITH INCREASE OF BRUSH LAG

The reason a motor speeds up with an increase of brush lag can be seen from Fig. 5. The armature coils included in twice the angle of lag (between *A* and *B*, and *C* and *D*) produce a flux which opposes that of the main field. These turns are called the back-ampere turns of the armature and are partially compensated for when the motor is designed, by increasing the number of ampere turns on the field. As the angle of lag increases, the number of back-ampere turns increases, and the main-field strength is thereby decreased, with a resultant increase in the speed of the motor. Since the speed of the motor is proportional to the counter-electromotive force in the armature (the field flux remaining constant) the effect of the increase of back-ampere turns is augmented by a decrease in the number of armature conductors available to produce the counter-electromotive force. These conductors, which are represented by those between *A* and *D* and *B* and *C*, are decreased as the brush lag is made greater.

Starter for Direct-Current Motors

A direct-current motor-starter, designated as "type S," has recently been placed on the market as by the Westinghouse Electric & Manufacturing Co.

The magnet switch, shown in the front view, Fig. 2, opens and closes the circuit, and its blowout coils prevent injurious arcing. With the interlocking devices and the overload relay, this switch affords low-voltage and overload protection. Opening the double-line switch (in the middle of the panel) disconnects the motor and starter from the line. The multipoint starting switch (at the left) cuts out by steps the starting resistance shown in the rear view, Fig. 1. This switch has an interlocking contact at each end, either one of which must be closed before the magnet switch can operate.

With the starting switch wide open, the operator closes the line switch and then the starting switch, step by step, while he holds its upper interlocking device closed. Owing to the action of the two interlocking devices, he cannot start until all the resistance is in circuit, nor can he leave the motor operating with part of the resistance in circuit. When closed, the starting switch holds the upper interlocking device closed.

An overload, either while starting, or while running, operates the relay, causing the magnet switch to open the circuit. Failure of voltage also opens the magnet switch. After either, the starting switch must be completely opened, placing all the starting resistance in circuit,

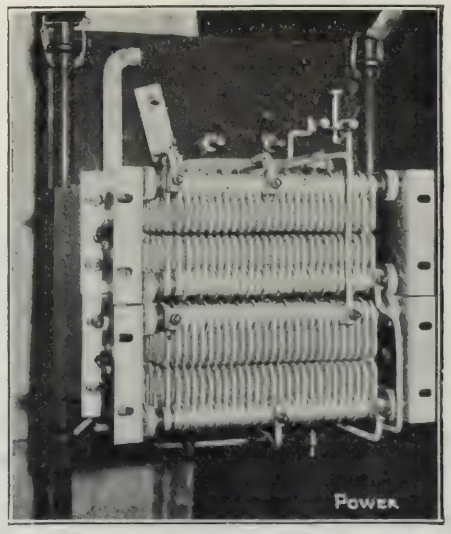


FIG. 1. BACK VIEW OF STARTER

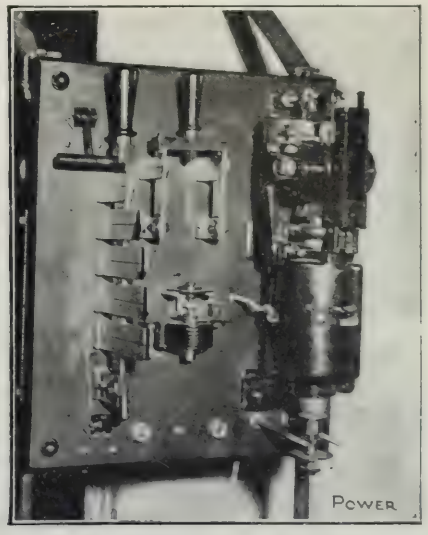


FIG. 2. FRONT VIEW OF STARTER

before the motor can again be started. Merely pulling out the starting switch far enough to release the upper interlocking device stops the motor, the circuit then being opened by the magnet switch dropping.

The starter is built in capacities suitable for 115-volt motors of from 10 to 60 hp., 230-volt motors of from 12½ to 125 hp. and 550-volt motors of from 30 to 250 hp.

CORRESPONDENCE

Motors Will Not Carry the Load

In reply to J. L. Shrode's inquiry, in the June 18 issue, my experience teaches that an unusual drop in speed in an induction motor, if it is not overloaded, is generally a sign of high resistance in the rotor. Thorough soldering at the ends of the bars should remedy the trouble, even when screws are used. From outside appearances, the soldering may seem sufficient, but upon thoroughly sweating in the solder a great difference will be found. This does not mean simply sticking on the solder, but heating the rotor with an iron and torch so that the solder will run freely; before cooling, the joint should be wiped smooth and clean with a cloth or waste.

I have had motors, normally running at 1800 r.p.m., drop to 1200 r.p.m. and be brought back to their full load speed by applying solder as directed. Contrary to Mr. Shrode's experience, I generally find them running very warm when the trouble is due to this cause.

Another source of trouble may be found in the running connections. In starting, the motor is thrown on the line and, after reaching the proper speed, is thrown on the fuses or running side. If a fuse is blown, a wire broken, or there is a poor contact in the starter, this would cause the motor to run single phase and it would not carry much load. This condition may be detected by testing with a lamp, if the voltage is not prohibitive, or a voltmeter and ammeter connected beyond the starter.

An uneven air gap may cause the motor to slow down, and if a heavy load is thrown on, the rotor will strike the laminations and stop; this will blow the fuses. If the motor is in a good location, this may be observed by taking off the belt and looking through the air gap, placing a light on the opposite side, if natural light is not sufficient. Have an assistant start the motor and, when up to speed, shut off the current. If the bearings are very much worn, the rotor will be seen to rise and fall. When having bushings or bearings babbitted, I have them bored out to within 0.005 in. and have never had one fail.

R. LENON.

Keene, N. H.

Regarding Mr. Shrode's trouble with motors not carrying loads, I submit the following suggestions:

It would seem that there is too much resistance at the connection of the rotor bars to the short-circuiting rings, which condition would increase the slip, thereby accounting for the drop in speed at full load. A very little resistance at these points would tend to affect the speed materially. In a 7½-hp. machine, possibly the resistance is introduced by shellac or some kindred material used for insulating the rotor bars, or dirt, the rings not being drawn to a firm, clean electrical connection with the rotor bars.

Regarding the second motor: this is a fairly common occurrence with soldered squirrel-cage windings. The local heating on one end most likely is due to imperfect or loosened soldered connections. The usual remedy is to resolder the bars, clinching them to the rings with a set and hammer, if possible.

C. E. MEAD.

New York City.

Commutator Lubrication

In a recent issue I noticed an inquiry as to the lubrication of generator commutators. We have a 10,000-kw. plant and once had considerable trouble due to "shorts" in the commutators caused by the charring of the mica between the commutator segments.

Then we were using clean engine oil on the commutators. I tried paraffin, but that was not satisfactory. I then prepared a compound of two parts by weight of paraffin, two parts of petroleum and one part of kerosene, mixed together and melted. When cold it forms a thick paste which can be applied to the commutator in a small quantity with a rag. The commutator should then be rubbed off good with a dry cloth to remove any surplus grease. This compound proved highly satisfactory and our commutators are now in excellent condition.

JOHN M. GLENN.

Chicago, Ill.

I have found that the best way to lubricate commutators, keep the brushes from chattering and avoid sparking, is to boil the brushes in a good grade of cylinder oil for about 24 hours once a month.

One of your correspondents recommends rod packing, but this did not work well in my case, on account of the commutator cutting the packing. Our commutator is slotted. Packing might be used on slow-speed machines having the mica cut down.

J. B. LINKEN.

Charlotte, N. C.

The dynamo Mr. Drewry mentions, in the June 4 issue, is certainly exceptional, and must be working under a steady load to give such results. He does not say what it is used for, but I do not think it can be an exciter. We have a very unsteady load, the exciter current varying from 60 to 95 amp. every few minutes. The exciters are controlled by a regulator, and, with the best of care, the commutators have to be turned down at least once a year and sometimes more often.

Mr. Hansa's rod packing is probably all right. If it is lubricated hemp it will pick up the dirt and lubricate a little at the same time, and if a rubber packing, the graphite will smooth up the brushes, thus stopping the cutting and sparking.

We have one exciter that causes considerable trouble. It has been turned down several times and sandpaper has been used to scour it, oiling slightly after scouring. Gasoline has also been tried after sandpapering and has given good results.

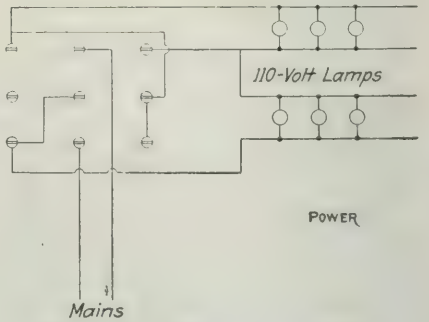
When the brown color of the commutator begins to turn black, it is a sign that it is getting dirty. We then moisten a piece of waste with gasoline and hold it on the commutator for a few seconds; this cleans off all the dirt, and the gasoline that gets on the brushes quickly evaporates before any dirt has a chance to collect. We use a part-graphite brush, as it has been our experience that a carbon brush will harden next to the commutator in a short time.

V. C. WOOD.

Copenhagen, N. Y.

Lighting Circuit for 220 or 110 Volts

The connections shown in POWER of May 21 for operating a lighting circuit on either a 220-volt or a 110-volt supply circuit with 110-volt lamps could be made with any standard triple-pole double-throw knife switch (leaving the middle clip on the left-hand end dead) without



WIRING FOR 220 OR 110 VOLTS

any special attachments, as shown in the accompanying diagram.

Closing the switch to the left connects the two groups in series for operation on 220 volts and closing it to the right puts the two groups in parallel for a 110-volt circuit.

P. JUSTUS.

Cleveland, Ohio.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

American Practice in Rating Internal Combustion Engines* †

By T. C. Ulbricht and C. E. Torrance, Jr.

During the past two years, the authors have been investigating this subject, in order to determine empirical formulas which would represent the average practice of American manufacturers, in the rating of gas engines of the stationary type. Data sheets were sent to all the prominent manufacturers and to the majority of chief engineers operating gas-power installations, with the request that these sheets be filled in. The information called for was the size, indicated horsepower, brake horsepower, revolutions per minute, kind of gas used, various efficiencies, etc., for all the standard makes of internal-combustion engines of the four-stroke-cycle, stationary type only, that are at present manufactured in the United States.

As a result, valuable data on some six hundred different sizes and types of machines were obtained, and from this information, in addition to as many other data as could be found in various engineering publications, textbooks, etc., curves and equations were worked out, showing the average practice current in America at the present time.

In addition to these curves and formulas, relating strictly to the average rating of stationary engines, the authors have been able to derive an efficiency table for American practice similar to the one derived some years ago by Güldner for German practice, which was the first object of this investigation. This table will appear in a later article.

The types of stationary engines to which this work is entirely confined, are:

Single-acting horizontal	Single-cylinder Twin-cylinder
Single-acting vertical	Single-cylinder Two-cylinder Three-cylinder Four-cylinder
Single-acting tandem	Single Twin
Double-acting horizontal	Single-cylinder Twin-cylinder Tandem Twin-tandem

The principal gases used are: Producer gas, from anthracite and bituminous coals, coke, lignite, oil, peat and wood; illuminating gas, from uncarbureted water gas, carbureted water gas and coal or bench gas; natural gas, including that high in hydrogen, that low in hydrogen and average composition; blast-furnace gas, coke-oven gas and oil gas.

The fuel oils used are: Gasoline, kerosene, crude oil and fuel oil.

Rating for Horizontal and Vertical Engines

Due to the large number of variable factors entering into the design and rating of stationary gas engines, it seemed at first impossible to derive any formulas which would take into account all of these variables with any degree of accuracy. However, it was found that empirical formulas could be obtained, representing an average of American practice and, although not applying in every detail to any one set of engines, yet they show to a nicety wherein lies the safe and sane design.

In order to determine average curves for gas-engine rating it is necessary to consider several fundamental principles, such as the type of engine, horizontal or vertical; arrangement and number of cylinders; kind of fuel used; volumetric, thermal and mechanical efficiencies, etc. These facts are all accounted for in the curves, which are derived in logical order.

Determination of Rated Brake Horsepower

The rated brake horsepower (b.hp.) may be considered as a function of the product d^2ln, where d is the diameter of the cylinder in inches, l the length of the piston stroke in inches, and n the revolutions per minute.

Figs. 1 to 7 show the average, maximum, and minimum curves obtained for the various gases and fuel oils, by plotting the values of d^2ln against the normal brake horsepower per cylinder per end (manufacturer's rating). The equation of each average curve is as follows, in each case representing engines employing the four-stroke-cycle and of the stationary type only.

For engines using producer gas,

$$b.hp. = \frac{d^2ln}{18{,}500} - 2$$

(Average for all values, Curve 2, Fig. 1.)

$$b.hp. = \frac{d^2ln}{17{,}900} - 1$$

(Average for single-acting horizontal and vertical engines, Curve 4, Fig. 1.)

$$b.hp. = \frac{d^2ln}{20{,}600} - 4$$

(Average for double-acting horizontal gas engines, Curve 5, Fig. 1.)

For engines using natural gas,

$$b.hp. = \frac{d^2ln}{15{,}200} - 5 \text{ (Curve 2, Fig. 2)}$$

For engines using illuminating gas,

$$b.hp. = \frac{d^2ln}{15{,}700} - 2 \text{ (Curve 2, Fig. 3)}$$

For engines using blast-furnace gas,

$$b.hp. = \frac{d^2ln}{21{,}000} - 5 \text{ (Fig. 4)}$$

For engines using gasoline,

$$b.hp. = \frac{d^2ln}{16{,}400} - 0.5 \text{ (Curve 2, Fig. 5)}$$

For engines using oils and distillates,

$$b.hp. = \frac{d^2ln}{21{,}875} - 0.75 \text{ (Curve 2, Fig. 6)}$$

In Fig. 7 are plotted the average curves for all the fuels for the purpose of comparison.

In this way the normal rated brake horsepower of an engine can be determined when the cylinder diameter, stroke, revolutions per minute and kind of fuel are known.

In order to determine the relation existing between the revolutions per minute and brake horsepower, curves (Figs. 8 to 11) were plotted with revolutions per minute as abscissae, and brake horsepower per cylinder per end as ordinates. An average curve passed through the points for each type of engine takes the form of a rectangular hyperbola, the equation of which can be readily determined.

The results are as follows: Single-acting vertical engines, single and multi-cylinder,

$$r.p.m. = \frac{5368}{b.hp. + 14} + 176 \text{ (Fig. 8)}$$

Single-cylinder, single-acting horizontal engines (Fig. 9),

$$r.p.m. = \frac{3020}{b.hp. + 9} + 128 \text{ (for gasoline)}$$

$$r.p.m. = \frac{6580}{b.hp. + 21} + 131 \text{ (for gases)}$$

Single-acting tandem engines,

$$r.p.m. = \frac{3650}{b.hp. + 4.5} + 156 \text{ (Fig. 10)}$$

Double-acting horizontal engines,

$$r.p.m. = \frac{9500}{b.hp. + 29} + 72 \text{ (Fig. 11)}$$

It will be noted that there are two distinct curves on Fig. 9, one for gasoline and one for gases, showing that for a given brake horsepower per cylinder per end the gasoline engines have a smaller number of revolutions per minute than those operating on gases. This fact is

*This series of articles is the result of a thesis investigation made under the direction of Professors C. F. Hirshfeld and H. Diederichs, of Cornell University.

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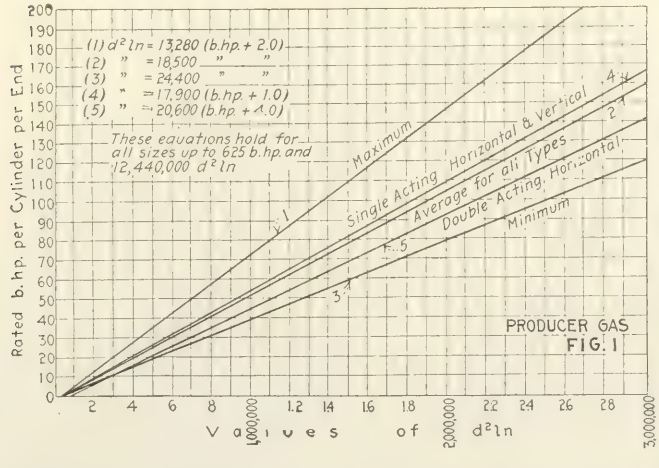
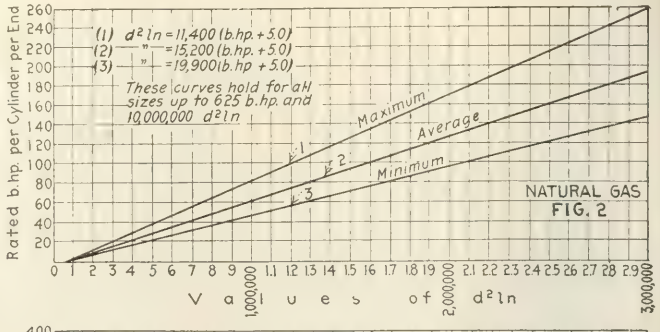
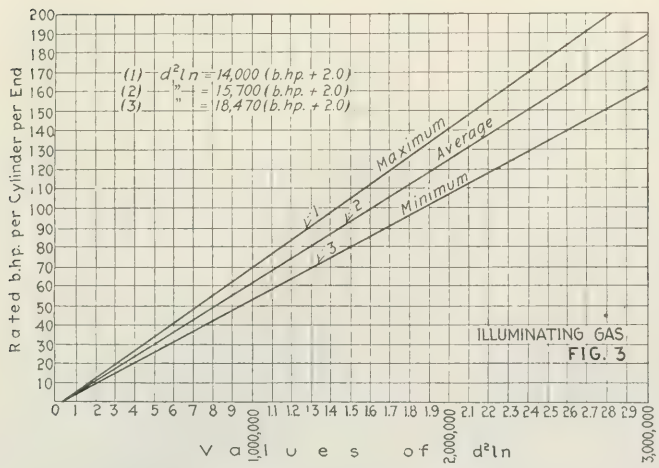
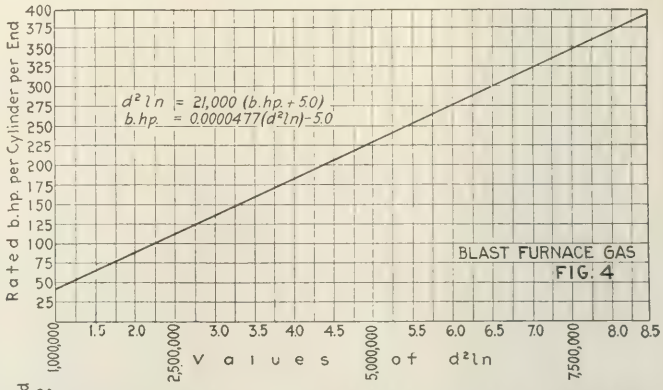
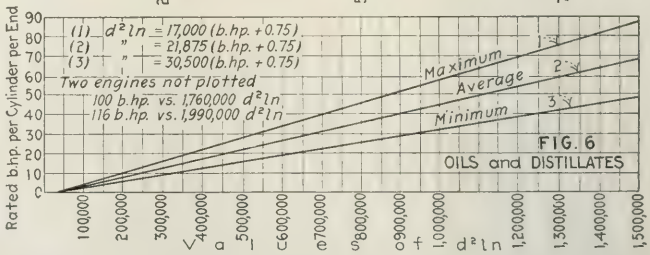
accounted for in two ways: first, that in small gas engines, the inlet valve is usually automatic, thereby reducing the charge volume and hence the speed, and, secondly, that some makers use the same engine for both gasoline and gases, by simply changing the valves. The gasoline engine will have a lower number of revolutions per minute, a larger d^2l, and a lower volumetric efficiency, the latter due to automatic inlet valves and to throttling losses on account of friction in the carburetor and pipes. The gasoline engine will also have a lower keep the d^2ln values constant for a given horsepower output. From a study of the preceding curves it will be noted that the d^2ln values for gasoline and illuminating gas are almost the same, showing that the d^2l values for gasoline must be larger than for other gases.

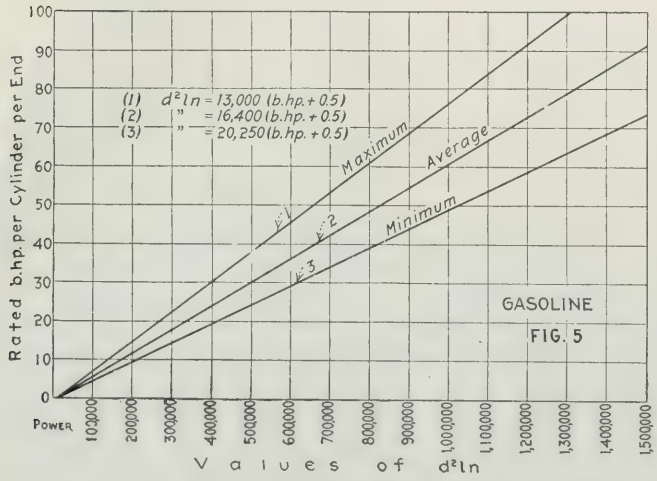

(1) d²ln = 13,280 (b.hp. + 2.0)
(2) " = 18,500 " "
(3) " = 24,400 " "
(4) " = 17,900 (b.hp. + 1.0)
(5) " = 20,600 (b.hp. + 1.0)
These equations hold for all sizes up to 625 b.hp. and 12,440,000 d²ln
Maximum
Single Acting, Horizontal & Vertical
Average for all Types
Double Acting, Horizontal
Minimum
PRODUCER GAS
FIG. 1
Rated b.hp. per Cylinder per End
Values of d²ln

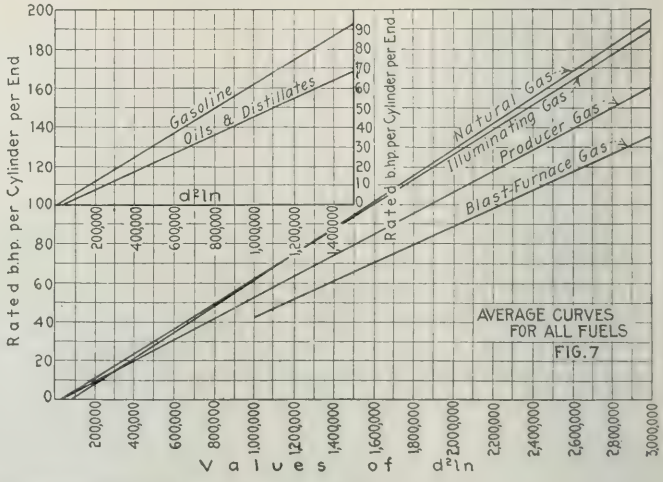

(1) d²ln = 11,400 (b.hp. + 5.0)
(2) " = 15,200 (b.hp. + 5.0)
(3) " = 19,900 (b.hp. + 5.0)
These curves hold for all sizes up to 625 b.hp. and 10,000,000 d²ln
Maximum
Average
Minimum
NATURAL GAS
FIG. 2
Rated b.hp. per Cylinder per End
Values of d²ln






(1) d²ln = 17,000 (b.hp. + 0.75)
(2) " = 21,875 (b.hp. + 0.75)
(3) " = 30,500 (b.hp. + 0.75)
Two engines not plotted
100 b.hp. vs. 1,760,000 d²ln
116 b.hp. vs. 1,990,000 d²ln
Maximum
Average
Minimum
FIG. 6
OILS and DISTILLATES
Rated b.hp. per Cylinder per End
Values of d²ln





VARIATION OF d^2ln WITH BRAKE HORSEPOWER FOR DIFFERENT FUELS

line has a high heat value and therefore requires a lower speed than the fuel gases, which have much lower heat values.

Assuming two engines developing the same brake horsepower, one operating on gas and the other on gasoline, the following facts, which a study of the curves will show, at once become apparent. The thermal efficiency, due to heat lost to the jackets and to a lower compression pressure.

In order to increase the volumetric and the thermal efficiencies, the manufacturers have attempted to correct the foregoing losses by increasing the cylinder sizes in gasoline engines and decreasing the revolutions per minute in order to

Another important fact is that in a gasoline-vapor mixture, there are apt to be small drops of gasoline, surrounded by vapor films, the latter only being explosive. Due to this imperfect mixture, the cylinder volume has been increased in order to admit more volume charge, thereby increasing the thermal efficiency.

Having determined the brake or de-

veloped horsepower, the revolutions per minute and the kind of fuel used, the cylinder diameters and length of stroke must next be found.

Determination of Stroke and Diameter

The definite relation between the cylinder diameter (d) in inches and the length of stroke (l) in inches, is shown in Fig. 12, from which equations were derived.

$$d = 0.91\ l - 0.45 \qquad \text{(Curve 1)}$$

For single-acting horizontal engines, single cylinder,

$$d = 0.667\ l + 0.4 \qquad \text{(Curve 4)}$$

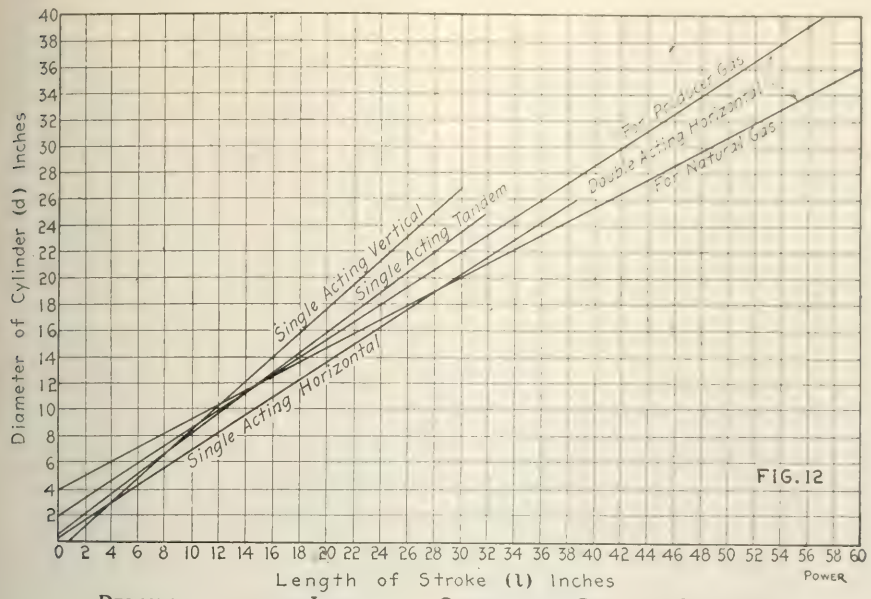
For single-acting tandem engines,

$$d = 0.772\ l + 0.55 \qquad \text{(Curve 2)}$$

For double-acting horizontal engines,

$$d = 0.533\ l + 4 \qquad \text{(Curve 5)}$$

$$d = 0.667\ l + 2 \qquad \text{(Curve 3)}$$



Relation between Length of Stroke and Cylinder Diameter

These equations follow the straight-line law, $y = ax + b$, where b is the y intercept, a the tangent of the angle made with the horizontal and $y = d$ and $x = l$.

For single-acting vertical engines, single and multi-cylinder,

From the preceding equations, the length of stroke (l) can be determined in terms of the cylinder diameter (d) and therefore gives a method for determining the final unknown in the various equations, relating to engine rating.

Problem

Assume a 100-b.hp., single-acting, single-cylinder, horizontal engine, operating on producer gas, and running at its normal speed for rated load. To determine the dimensions of this engine by means of the preceding equations:

From equation 2, Fig. 1,

$$b.hp. = \frac{d^2 ln}{18{,}500} - 2$$

or

$$100 = \frac{d^2 ln}{18{,}500} - 2$$

Hence,

$$d^2 ln = 1{,}887{,}000$$

From equation 1, Fig. 9,

$$r.p.m. = \frac{6580}{b.hp. + 21} + 131$$

or,

$$r.p.m. = \frac{6580}{100 + 21} + 131 = 186 \text{ approx.}$$

Therefore, $d^2 ln \div n = 1{,}887{,}000 \div 186 = 10{,}145 = d^2 l$ and $l = 10{,}145 \div d^2$.

But from the equation of curve 4, Fig. 12,

$$d = 0.667\ l + 0.4 \text{ or}$$

$$l = 1.5\ d - 0.6$$

Substituting this value of l in the equation $l = 10{,}145 \div d^2$

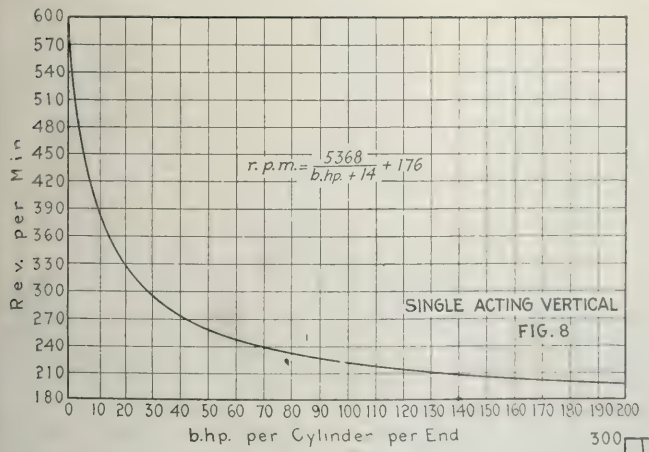
$$d = \frac{0.667\ (10{,}145)}{d^2} + 0.4$$

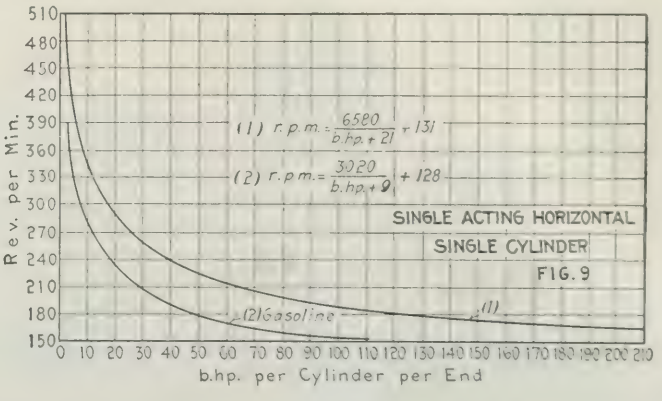
or, $d^3 - 0.4\ d^2 = 6766.715$

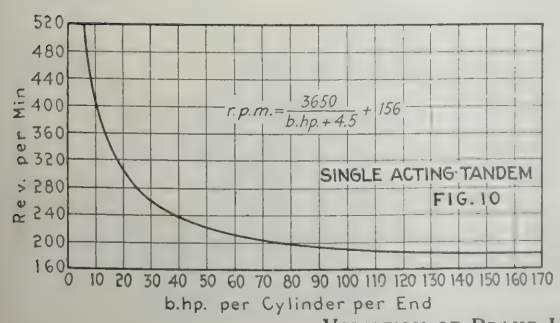
whence, solving by inspection, d is approximately 19 in. Therefore,

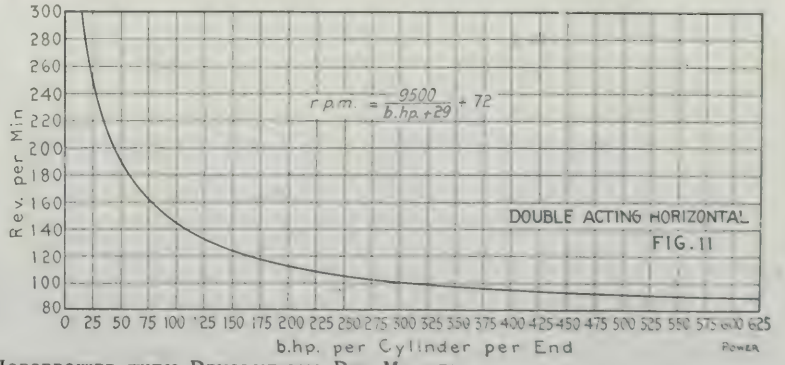
$$l = 1.5\ d - 0.6 = 28\ in.$$

The required engine dimensions then are diameter of cylinder = 16 in.; stroke = 28 in.; revolutions per minute = 186, and brake horsepower (total) = 100.




Rev. per Min.
(1) r.p.m. = 6580 / (b.hp. + 21) + 131
(2) r.p.m. = 3020 / (b.hp. + 9) + 128
SINGLE ACTING HORIZONTAL
SINGLE CYLINDER
FIG. 9
(2) Gasoline
(1)
b.hp. per Cylinder per End





Variation of Brake Horsepower with Revolutions Per Minute

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilating Park School, Rutherford, N. J.

By Charles A. Fuller

The use of hot-air furnaces for heating and ventilating is gradually being abandoned except in small residences. Here they are probably better adapted for the work they have to perform than the more recent steam, vapor or hot-water heating plants. In the larger residences and schools, particularly where a definite amount of ventilation is necessary, the hot-air furnace is of little value.

Often a fan has been used to force the proper amount of air through the furnace and into the rooms with bypass dampers arranged to control the temperature. Sometimes this has been fairly satisfactory, but generally is a failure. It is difficult to obtain the proper temperature regulation and nearly always there is an odor of coal gas in the rooms at times. The air also lacks the proper humidity and contains considerable dust which is unavoidable with furnaces.

The heating equipment in the Park School at Rutherford, N. J., is of the type described and steps are now being taken to replace it by an indirect steam plant. The accompanying drawings show the general arrangement of the new layout as changed from the old system.

Originally five separate hot-air furnaces were in the five chambers where the banks of reheaters are now shown. These were all fired from a central corridor, the fire doors opening through the brickwork. Considerably more coal was necessary to keep the five separate furnaces in operation than would be burned if the firing were done in one central boiler.

The hot-air system is also supplemented by a low-pressure direct-steam system. Direct radiation is installed in each room and supplied from the old boiler shown in Fig. 1. The amount of radiation in each room is sufficient to take care of the heat loss from the walls and windows with no allowance for air change.

In each of the division walls shown in the plan between the fan room and the plenum chamber was installed originally a large propeller fan belted to an alternating-current motor. These fans served to force air through the furnaces and up into the rooms, but were seldom used. Each hot-air flue over the top of the furnaces had a bypass and a mixing damper to control the temperature.

The five furnaces are to be replaced by a 70-hp. boiler set beside the present boiler, which is to supply the tempering coils and reheater stacks.

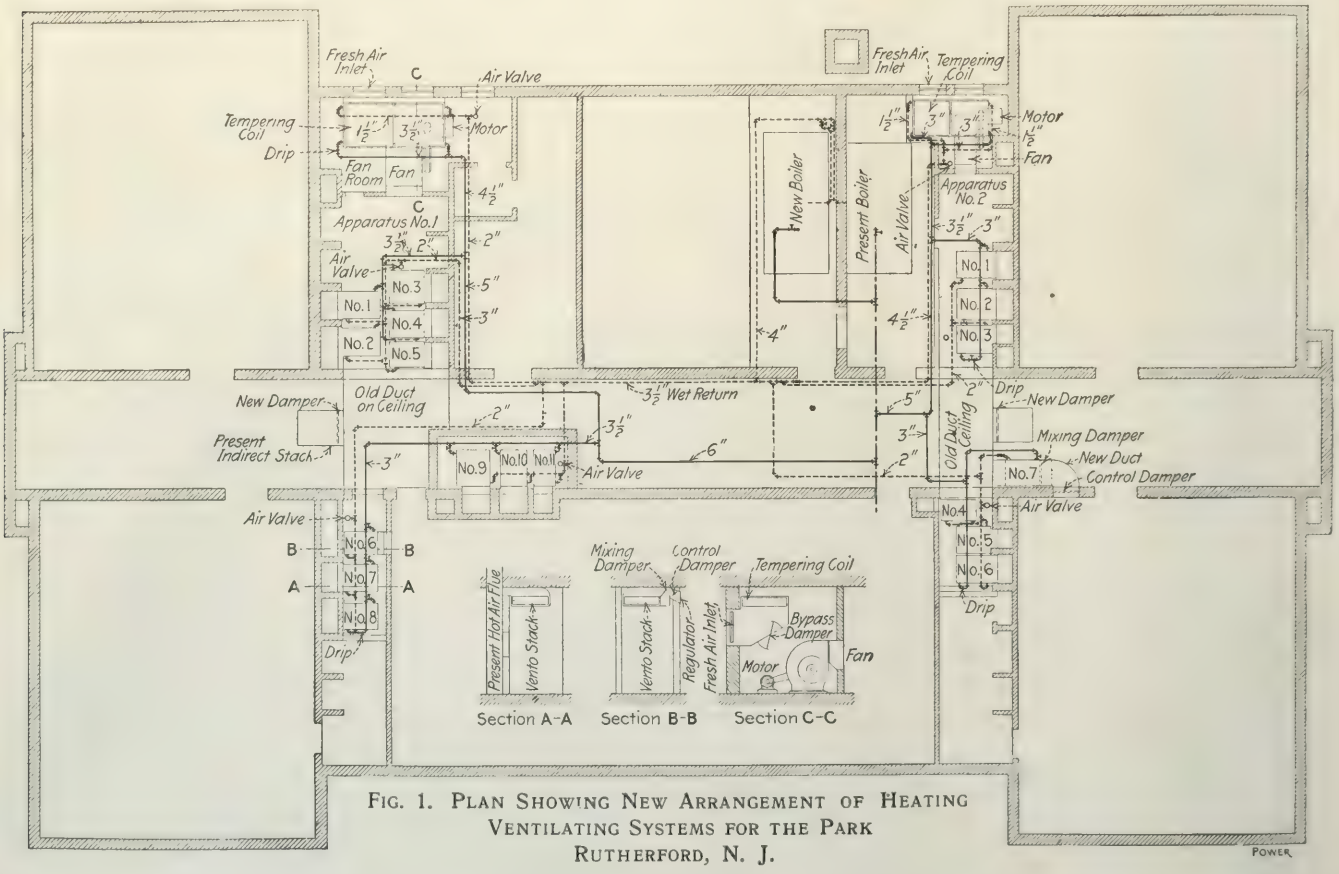


Fig. 1. Plan Showing New Arrangement of Heating Ventilating Systems for the Park Rutherford, N. J.

The two propeller fans are to be replaced by two multivane fans, chain-driven by alternating-current motors. The air is to enter through a tempering coil composed of one bank of Vento radiation hung horizontally about 12 in. from the ceiling. It will be inclosed on the four sides by a galvanized-iron casing with its top left open and the bottom connecting with the fresh-air inlet, as shown in the section CC. The air must then pass in through the fresh-air inlet and up through the tempering coil into the fan room from which the fan takes its supply. Thus arranged the tempering

coils will be 24 in. above the water line of the boiler, which permits the condensation to be returned by gravity.

The fans will discharge through the partition walls into the plenum chambers formed in the space formerly occupied by the hot-air furnaces. These chambers are to be as nearly air-tight as possible, the openings left by removing the furnaces, fire doors, smoke flues, etc., to be bricked up and sealed. Inspection doors will be left in each chamber.

At the base of each hot-air flue in the plenum chambers will be a reheater stack composed of one bank of Vento radiation. The casings around these stacks will be the same as those for the tempering coils, only reversed. The bottoms will be left open and the upper openings connected through short ducts to the present hot-air flues. The old mixing dampers are to be used, no change in these being necessary.

The air forced into the plenum chambers from the fans will have to pass either up through the reheater stack and into the hot-air flue, or through the opening below the stack into the flue, depending on the position of the mixing damper. If the damper is set midway, part of the air will pass through each opening.

The tempering coil will be large enough to raise the temperature of the air from zero to 50 deg. F when the required quantity of air is passing. A bypass opening with a hand-controlled damper in this casing will allow air to be drawn into the fan room without passing through the tempering coil if desired.

The reheater stacks will be large enough to raise the temperature of the air from 50 to 75 deg. The tempering coils will have thermostatic control, consisting of two Regitherms in the plenum chambers directly beyond the discharge of the fans. These will operate directly upon balanced valves in the steam-supply mains to the tempering coils and the temperature of the air in the plenum chambers will be held at approximately 50 deg. if the outside temperature is lower.

This arrangement will give ample control of the room temperature by means of the mixing dampers which are operated by chains from the rooms. It will also obviate any chance of cold drafts, which can hardly be avoided if the air is heated entirely by the stacks at the base of the flues without using tempering coils.

When the mixing damper is set to pass part of the air through the reheater and part through the bypass below, the two currents will rise through the flue without being thoroughly mixed. If the register happens to discharge the colder stratum of air below the other it will immediately descend and cause a noticeable cold draft.

If the cold air is partially heated by a tempering coil before reaching the reheater, even with the damper set in a mid-position, the draft will be less severe and not objectionable. Two rooms on the basement floor will receive their air supply from this system, one through reheater No. 6 of apparatus No. 1, and the other through reheater No. 7 of apparatus No. 2. The arrangement for introducing the air to these rooms is shown in section BB, Fig. 1. The heater will have a casing open at the bottom, like the heaters for the rooms above, and the top connected to a duct leading directly to the supply register. At the side of the casing a bypass will open directly into the duct leading to the register. At the top of this opening a mixing damper will shut off either the hot air, or the tempered air, unless set in an intermediate position similar to the others.

Besides the mixing damper a control damper in the duct will be arranged to set permanently in a fixed position when the proper amount of air is passed. The supply registers will be as usual, about 8 ft. from the floor, and the vent registers at the floor line.

The vent flues will all combine into two main vent flues in the attic space, passing through the roof and discharging through ventilators. Besides the ventilators, the main vent flues will have aspirating coils, as may be seen in the sectional elevation, Fig. 2. The system was designed by Clark, MacMillan & Riley, consulting engineers, New York City.

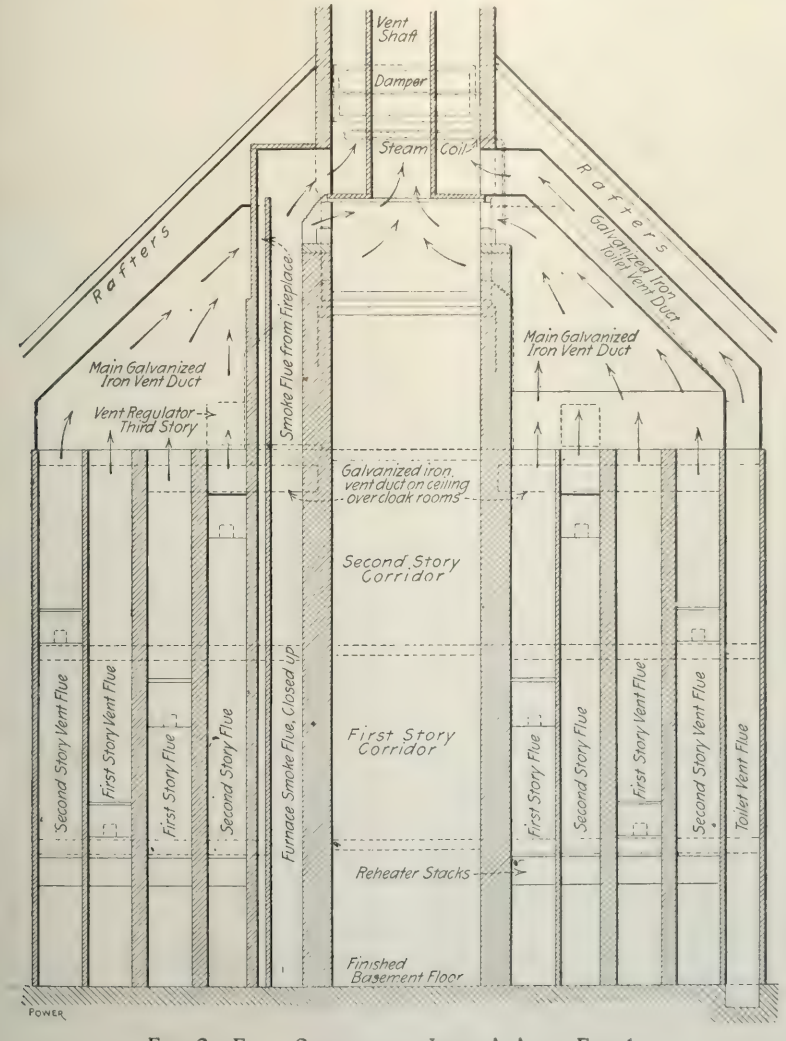


FIG. 2. FLUE SECTION ON LINE A-A OF FIG. 1

Return Trap Is Unsatisfactory

BY W. H. WAKEMAN

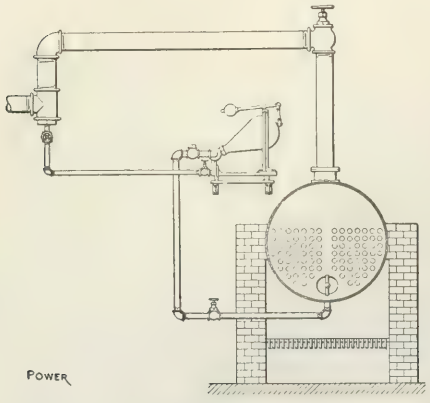
In a certain shop the heating system was unsatisfactory, and a return trap was installed as part of the improvements made. It was located above the water line, as shown in the illustration, and the feed pipe was connected to feed hot water into the boiler. Every return trap must have a live-steam connection to equalize the pressure above and below the water, and it is always a good plan to take steam for this purpose direct from the boiler.

In this case it was not convenient to do this, so a drip from a tee in the main steam pipe was used to supply the necessary pressure. When put into service the device would work well as long as the engine was shut down, but when

steam was admitted to the cylinder, the trap failed to operate, and it became necessary to blow hot water out of the heating system into the sewer.

It was claimed that the failure was due to wet steam, as it was taken out of the drip pocket formed by the lower part of the tee, but this is not correct, as a return trap does not require dry steam. If hot water is mingled with steam coming to a return trap it simply causes more water to go back into the boiler, but it does not interfere with the operation of the trap. It is necessary, however, for full pressure to be obtained in the trap as otherwise water cannot go into the boiler.

Consider the principle which causes a return trap to operate. Pressure in the return pipes of a heating system sends hot water into the trap, which has been relieved of all pressure for this purpose. The weight of water in the trap causes the steam valve to open and admit full



CONNECTIONS WHICH WOULD NOT RETURN CONDENSATION

pressure to the body of it, and the weight causes the water to seek a lower level; hence it runs into the boiler.

Under conditions commonly found in practice, a column of hot water 2 ft. 6 in. high gives a head of approximately 1 lb.; hence if a return trap is set this distance above the water level, there is 1 lb. pressure available to force water into the boiler. If live steam is taken from a drip pipe, as shown in the illustration, and pressure in the main pipe is reduced 1 lb. or more by the engine taking steam, there is no pressure available to send water into the boiler; hence the trap fails to work. As soon as the engine is shut down the reduction ceases and the trap does the work for which it was designed.

In 1895 the maximum distance over which power was transmitted was eight miles and the voltage 5500. Now, says *Cassier's Magazine*, the Southern Power Co., operating six hydro-electric plants and two auxiliary steam plants, sends 121,000 hp. over 1380 miles of transmission line at 50,000 and 100,000 volts.

District Heating Business*

The district heating business is nearly as old as the electric lighting business. As early as 1890 a number of electric lighting companies were selling exhaust steam for heating purposes. At the present time there are over 300 heating plants in the United States. The National Electric Light Association committee gives answers from 100 companies, one-half considering the business directly profitable, while practically all agree that it is of great advantage in connection with electric supply.

While the value of a combination electric and heating system is being recognized more fully as the companies extend their field to include the large commercial establishments formerly considered out of the reach of the central station, it is impossible to find another business so well established and in which so many companies are engaged where one-half believes that the business is yielding no profit.

It is doubtful if a majority of heating companies have the necessary data to make up a reliable operating balance. Heating plants are outgrowths of the electric business and generally are operated in connection with it; therefore the expenses frequently are not separated. The heating revenue is credited to the fuel account and often assumed to be practically all profit; a cost analysis is considered unnecessary. Where an analysis is made, it is many times inadequate, the general expenses, fixed charges and depreciation being omitted.

Failure to show profit may be due to inadequate rates. Many managers consider heating revenue practically all net gain and they establish rate schedules without considering the value of the service to the consumer or of its cost to the company. The popularity of the service shows conclusively its value.

The value to the consumer is easily from 10 to 25 per cent. more than the cost of maintaining his own plant, due to the absence of smoke and dust, to lower insurance rates, to the release of valuable space otherwise occupied by the plant, and to the greater convenience and reliability of the district heating service. The value of the service will generally permit rates based on the cost of supplying live steam, and rates should always be so based, rather than on the not always correct assumption that the steam is a byproduct.

Faulty construction or improper operation may cause a financial loss. The same care should be exercised in designing and constructing a heating system as in the electric-supply system. In general, the distribution system should be designed to secure the maximum density of business with the minimum length of mains. The load on existing mains should be increased and extensions made only where it is certain profitable business can be obtained.

Customers' installations should be subject to supervision. Where flat rates are in use, rigid regulations should be enforced, underground construction should be of best quality to reduce maintenance costs and minimize heat losses. But it must be kept in mind that investment charges on expensive underground construction form a serious item in the cost of district heat supply, and construction costs must not become so heavy that the charges more than offset the resulting operating economies. Where meters are used, a thorough system of inspections and tests is necessary to prevent serious losses of revenue. Above all, adequate records are essential for intelligent and economical operation.

Local conditions have much to do with the success or failure of heating systems; and plants are operated in many places where local conditions do not warrant the investment. This is partly due to the idea that any heating revenue is net profit, and to considerations of policy, such as necessity for occupying the territory to prevent undesirable competition and the ability to offer heating service in connection with electric service. Among the points to be considered in local conditions are location of steam plant, density of available business, probable growth of district, obstructions to underground construction, probable loss of electric-plant capacity and efficiency due to back pressure, length and character of heating season, probable coördination of heating load and electric loads and relative costs of steam coal and domestic coal. These points apply to exhaust heating. Where live-steam heating is contemplated, other considerations enter and the chances of loss become much greater.

Plants depending principally on electrical business may be operating at a loss because they carry undesirable business, such as restaurants, etc., which require summer service. Low rates may be intentionally established to facilitate taking over private plants, with combined heating and electrical load, but this does not seem warranted except where the value of the service to the consumer will not permit higher rates and where the net result is an advantage to the entire electric business.

*Address by President A. D. Spencer, before the National District Heating Association, at Detroit, Mich., July 25.

Power at ½c. per kilowatt-hour is furnished to the Baltimore Copper Smelting & Rolling Co., according to recent testimony given before the Maryland Public Service Commission. The revenue derived from the copper works by the Consolidated Gas, Electric Light & Power Co. was stated to be about $125,000 per year. Power is furnished by the Falls of the Susquehanna.

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POWER

A Short Cut for Boiler Test Calculations

There is always a satisfaction in being able to bring to our readers' attention, means for saving time or effort in connection with their work. For this reason we are particularly gratified at being privileged to present the chart for calculating boiler tests as evolved by F. O. Pahmeyer, of the Heine Safety Boiler Co. (page 116). The ingenuity of the scheme speaks for itself, and did it give only approximate results, it would be worth employing in the majority of cases, but still more fortunately its accuracy is well within the limits of error, attending the taking of observations for the test data. It therefore may be confidently used for calculating any test, however important, and if the engineer still feels that he would prefer to carry out the calculations in the ordinary tedious manner, he will find the chart a useful check on his mathematical work.

Here is a single chart easily contained on one of POWER's pages, yet with it, from any set of boiler-test data, including the average steam pressure, the average feed-water temperature, the factor of evaporation (ascertained from the two foregoing from a table or by calculation) the amount of water evaporated per hour and the amount of coal fired, can be determined the capacity in boiler horsepower, the total equivalent evaporation, the actual and equivalent evaporation per pound of fuel, and the efficiency. By determining the percentages of moisture and combustible in the coal, the evaporation per pound of dry coal and per pound of combustible can be determined, and by correcting for the moisture in the steam, the true amount of water evaporated and the true factor of evaporation can be found.

Special advantages in the use of the chart will be recognized in the means for quickly arriving at results during the test, without waiting until its conclusion to get the refined figures, even supposing the latter to be considered at all necessary. Further, it should encourage more frequent tests of the performance of a boiler plant than would be likely to be made were it necessary to take the time for the various calculations.

It is hardly needful to remind those who have had to do with boiler testing, that this chart, with the explanation of its use, is worth preserving, but this advice may well be heeded by those who, at some future time, are likely to be concerned with it.

Evaluating Coals

It is quite generally appreciated that an indiscriminate selection of fuel for an existing plant is incompatible with the securing of the highest economy. Boiler furnaces are directly designed with consideration for the kind of fuel to be burned and size alone is not the determining factor simply because a certain form of grate bar is used; quite as important is the percentage of ash, for upon this depends the proportion of grate area.

The purchase of coal is not a matter of getting the cheapest on the market regardless of its grade, or heat value. As has just been shown, due allowance must be made for the nature of the furnace, to select the coal for which it is best adapted. Presupposing that the furnace is one originally chosen as suitable for the fuel most readily obtained in the region where it is installed, it is unwise to make any radical departures such as an attempt to burn an entirely different grade of coal or one from a different locality.

Such considerations as the above are fundamental to the trained purchaser of coal, but what is not so generally known, or if known is usually unheeded, is that coal from the same region and even the same vein, may vary very materially in its per cent. of ash content and thereby very decidedly in value. Some very pertinent hints to purchasers of coal are given on page 115, from a report of the Prime Movers Committee of the National

Electric Light Association, a valuable part in which is the extract from the paper by L. P. Crescelius on "The Purchase of Bituminous Coal on B.t.u. Specifications," presented before the American Electric Railway Association. This shows the detrimental effect of high ash content and suggests a reasonable protection of the consumer by imposing penalties that increase rapidly the more the ash content exceeds the prescribed amount.

The body of the report is particularly interesting for the empirical formula it presents whereby a coal may be evaluated on a basis of the percentage of ash it contains. Many have held that such a procedure is unwarranted inasmuch as variation in ash alone is not a safe measure of the value of a coal. Under all conditions it is not, but under the limited conditions that obtain in any individual case it is quite fair to apply, providing the proper relative weight is ascribed to the ash percentage. Within the usual limits, for example, it is shown that 1½ times the percentage of ash, and not the percentage itself alone, should be deducted from the unity, or 100 per cent., value of the fuel.

This simple and easily remembered rule should prove helpful to those having to do with the purchase of coal, and particularly those for whom a calorimetric determination of the value is either out of the question, or at least inconvenient. Moreover, as the report shows, the ash basis is quite as apt to be reliable, for at times the calorimeter may induce very misleading conclusions.

Flange Standards

To have two standards is better than to have no standard, but to have one standard and that the best possible is the consummation devoutly to be wished.

Until recently we had no standard for pipe flanges and flanged fittings. The American Society of Mechanical Engineers and the National Association of Steam and Hot Water Fitters, through joint committees, evolved the U. S. 1912 Standard. The manufacturers claim that they have been ignored in its making and a "Committee of Manufacturers on Standardization of Fittings and Valves" at a meeting held on July 10 adopted the "Manufacturers 1912 Schedule" of flanged fittings and flanges to take effect the first of next October.

We have already published the "U. S. 1912 Standard" (see POWER, Mar. 19, 1912). We shall try to present the "Manufacturers 1912 Schedule" in an early issue. The discussion will then be on the comparative merits of the two. Since the manufacturers have decided to have a uniform schedule the main objection to changing present conditions, the cost of new patterns and catalogs, has apparently been met, and now that this expense, which must in the last analysis be borne by the user, is to be incurred, it is very much to be desired that it should be used to bring about a single standard which would meet all requirements and be acceptable to manufacturers and users alike.

An Ancient Steam Plant

Those who think all the historic old engines are now in museums, if in existence at all, will be surprised to read John S. Leese's description of the Madeley Wood Co.'s plant at Iron Bridge, Shropshire, England, on page 108, and learn that some veritable patriarchs in the power-plant world are still in operation. Quite as interesting as the old beam engines with open-top cylinders are the ancient boilers supplying them with steam at eight to ten pounds pressure. Truly it may be said of the creators of these old plants, "They builded better than they knew."

To us who are quite accustomed to see a plant replaced before it is two decades old, and sometimes not even one, it is scarcely believable that engines which have passed the century mark have not yet outlived their usefulness in the estimation of their present owners. Even as measured by modern standards the fuel consumption relative to the work done is not very high, so the fact that this equipment is still in operation cannot be attributed entirely to sentiment.

Massachusetts Inspection Law Threatened

Not seriously nor permanently; it has done too well and resulted in too general satisfaction for that. Even the manufacturers who opposed it are satisfied of its beneficence and it is serving as a model for other states and cities. But by one of those vagaries of that intangible and incomprehensible bulwark, the law, it is threatened with a period of suspended animation.

The law as passed authorized the governor to appoint one of the members of the Boiler Inspection Department as chief inspector. The governor appointed Joseph McNeill and all was well for several years. Joseph resigned to take a better job and the law does not tell the governor to appoint any successor to him. Hence the perplexity.

One cannot run a boiler in Massachusetts until it has been inspected, vicariously at least, by the chief boiler inspector. But there is and can be no chief boiler inspector until the Legislature repairs its oversight to the satisfaction of the attorney general.

There are several inspectors, insurance and otherwise, on the job, however, and it is to be inferred that the wheels of industry will continue to revolve in the old Bay State until the General Court can get into action, even if they do put some kinks into the legal crowbar that the attorney general has poked between their spokes.

A delectable example of inconsistency is furnished by the adoption by the Interstate Commerce Commission of New Jersey of a rule providing that, "If boilers are equipped with fusible plugs they shall be removed and cleaned of scale at least once every month."

One need not use fusible plugs if he does not want to, but if he does he must clean them once a month. How much safer is a boiler without a fusible plug than one with a scaled plug?

No longer will the vocations of the plumber, bricklayer and carpenter be held by many operating engineers as shining examples of what engineering is not. A school teaching women these trades will be started in New York in October. This ought to convince the disgruntled that even operating engineering has its compensations.

A news item reports a shortage of men in the railroad shops of a large city due to the employees becoming berry pickers at $3 per day. In the same edition a benevolent employer advertises for an engineer at $2 per day. Let's all pick berries.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Water Level Indicator

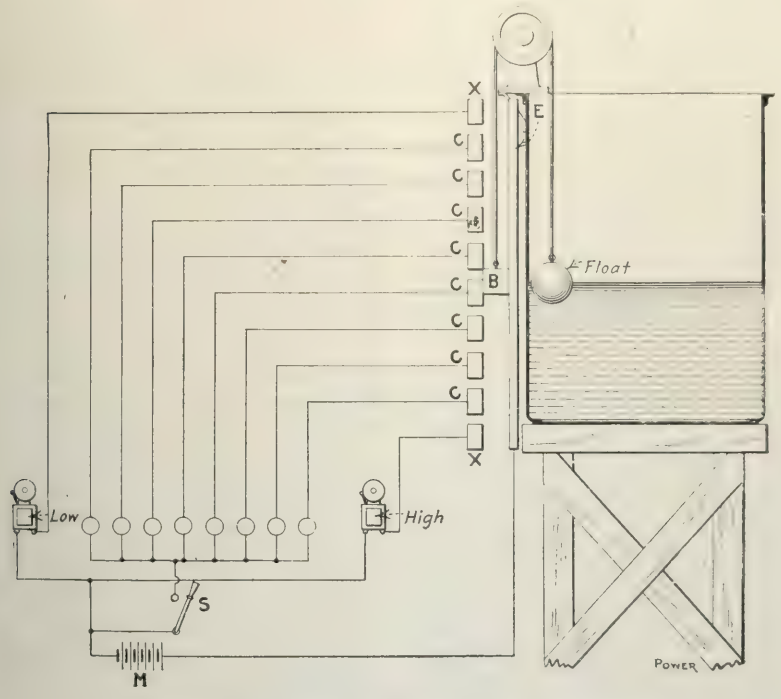
Edward L. Johnson's water-tank gage, shown in the Apr. 23 issue, reminds me of one I recently erected. The one illustrated was used to indicate the water level in a tank about 200 ft. from where the water was used and where it was necessary to know its level in the tank.

The somewhat elaborate arrangement was made possible by using a discarded telephone cable; otherwise its cost might prove prohibitive. In a box are eight low-voltage lamps with a glass front painted black, except for a round disk opposite each lamp, on which the depth of water indicated by that lamp is marked. The bells at each end are connected to the contacts *X* and are in circuit at all times to indicate high or low water. The lamps are controlled by the switch *S* which is only used to ascertain the level of water at any intermediate point.

The contact block *B* is attached to the float by a flexible wire rope passing over a sheave at the top of the tank. This block slides up and down between the two surfaces *E* and *C*. *E* is one continuous strip of iron while *C* is in sections, each connected to a lamp except the top and bottom ones. The contact block *B* is so long that it always touches one contact before leaving another, as otherwise at times the water level would not be indicated. The battery of dry cells shown at *M* operates the system.

P. JUSTUS.

Cleveland, Ohio.



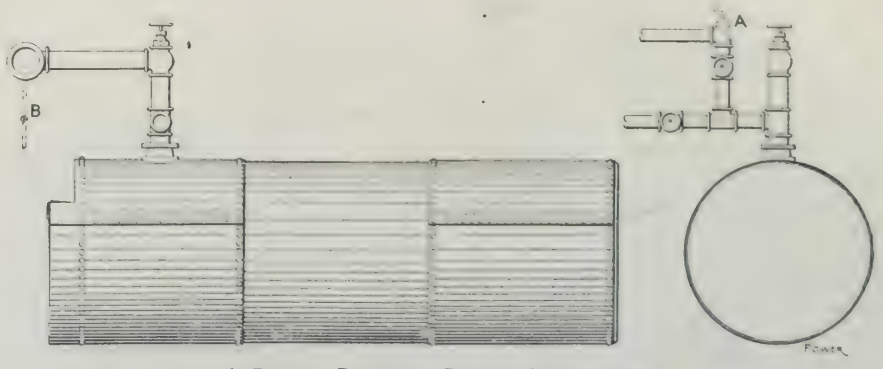
ELECTRICALLY OPERATED WATER LEVEL INDICATOR

Poorly Designed Boiler Connections

The illustration shows the shell of a tubular boiler fitted with an ordinary steam nozzle. Above this is a riser surmounted by an angle valve, but instead of using one piece of pipe for this, the steam fitter (evidently following the designing engineer's ideas) used a nipple, a reducing tee, then another nipple. By means of suitable pipe, this angle valve is connected to a header, out of which steam is taken to operate an automatic engine. The end elevation shows the piping as it appears when viewed from the rear of the boiler. The back of this angle valve is seen, and piping for taking steam from the reducing tee. If an angle valve had been used as indicated by dotted lines *A*, a more direct passage for steam, with two less screwed points, would have resulted.

Steam rises through the nozzle, a portion going to the engine while the remainder passes to a heating system through the smaller pipes, and on a level higher than the boiler. This system has suitable drip pipes, which dispose of the water resulting from the condensation of steam. No danger arises from this arrangement as long as steam is passing rapidly to the heating system, but, should there be only a moderate demand for steam, it would result in slow circulation through these two supply pipes, when there would be nothing to prevent them from acting as connections to two parts of a single pipe system. In this case water would flow back toward the boiler, and entering the riser would be caught up by steam going to the engine and discharged into the header.

If this header had a 1¼-in. drip pipe at each end, as shown in dotted lines at *B*, discharging into the boiler below the water line, the water might thus be disposed of, but, as this header has no drip pipe, the water must go to the engine resulting in a wreck unless large relief valves are provided.



A POORLY DESIGNED BOILER CONNECTION

In all such cases only one pipe should take steam from the boiler, and discharge it into a header large enough to act as a reservoir, and prevent rapid reduction of pressure. Steam for all purposes should

be taken from this header, and the latter should have enough drip pipes to keep it free from collected water.

W. H. WAKEMAN.

New Haven, Conn.

Boiler Explodes

A 22-hp. boiler exploded near here recently at a country sawmill, injuring three men. One of them will die; another will be crippled for life; the other man was blown some distance, but received only slight injuries. It is said that the connections to the water column were clogged.

This and other boiler explosions should be a lesson to all who have boilers intrusted to their care. Had the water column and gage glass been tested the explosion would probably have been prevented. An article dealing with the proper care of water columns and gage glasses might be beneficial.

P. F. ROBNETT.

Kinmundy, Ill.

Worn Piston Rod Allows Oil in Boilers

On taking charge of a plant recently, I learned that considerable trouble had been experienced with oil getting into the boilers. With a view to offsetting this trouble, quantities of sal-soda had been put into the boilers. This in turn had brought about other undesirable conditions, as leaky pipe joints and badly corroded pipes. An emulsion was also formed, which settled to the bottom of the boilers, and was nearly as obnoxious as the oil itself.

An oil separator formed part of the installation, but as the drainage was piped to the sewer, it was impossible to ascertain whether it did its work properly or not. This piping was first disconnected and it was found that little or no oil was discharged. This indicated that all the cylinder oil used in the engines and pumps was being conveyed with the feed water into the boilers.

The separator was opened and a gallon measure full of fine fragments of packing was found lodged in the lower funnel-shaped section leading to the oil outlet.

The mystery was then solved as to what had become of the packing put into the stuffing box of one of the engines. The piston rod of this engine was badly worn and could not be kept packed for even a short time. As it was not possible to shut this engine down long enough for proper repairs, a brass-fiber woven ring was placed in the bottom of the stuffing box next the cylinder. The rod was then packed with good packing and another brass ring inserted next to the follower.

No more packing got through the exhaust, and, after a thorough cleaning, the oil separator did its work. With the judicious use of a specially prepared boiler compound, no more trouble was experienced with oil in the boilers.

EDWARD T. BINNS.

Philadelphia, Penn.

Making Logarithmic Cross Section Paper

When engineers realize the advantages gained by J. Paul Clayton's new method of analyzing indicator diagrams, every progressive one will want to apply the analysis to his engine.

This method is very simple, in that it shows, with little labor and no calculation, such an important factor as quality of steam at cutoff. It allows leaks to be detected with certainty and precision and also enables one to determine the clearance of an engine from a diagram.

A sheet of logarithmic cross-section paper (necessary for the analysis) may not be at hand, and, anyhow, that which can be purchased is not quite satisfactory, as many of the spaces are too large for close estimating. That made as illustrated gives best results. Tack the paper and lay a slide rule on it. The rule acts as a straight-edge and indicates where the triangle is to be placed to obtain straight and logarithmically spaced lines.

Most engineers have, or should have, a slide rule, the scales on which are very accurate logarithmic scales. If the double scale on the upper edge, reading from 1 to 100, could be projected vertically across a sheet of paper and then horizontally across, keeping the 1 at the left and bottom. The result would be a very accurate and complete sheet of logarithmic paper of four squares. Equally good results may be obtained by drawing in the principal lines and using the slide to estimate intermediate distances.

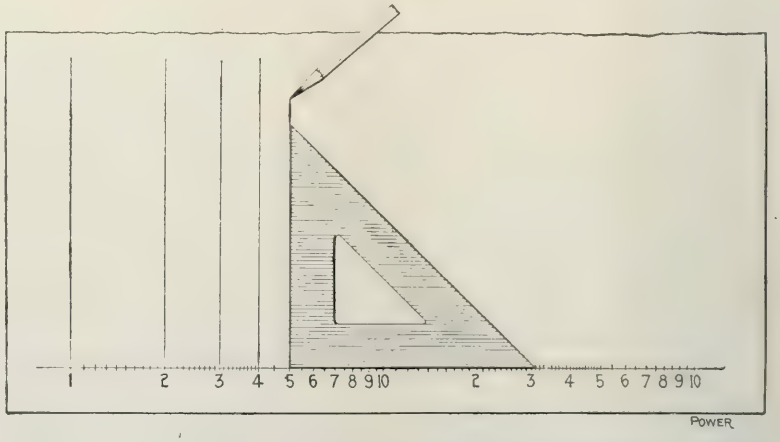
By making this diagram on tracing cloth, light prints may be taken on which to make the actual analysis, but the shrinkage of the blueprint when drying makes it too small for the scale on the slide. This may be allowed for by removing one of the full-size scales from the instruction book, which comes with a slide rule, wetting it thoroughly, and, while still damp, using it as a guide in drawing in the principal lines on the tracing cloth. The tracing-cloth scale will then be too large, but blueprints from it shrink enough in drying so the slide scale exactly fits, and plotting can be done on it as accurately as the slide rule can be read on the corresponding scale.

It will take about one hour to do this work.

G. W. MUNRO.

La Fayette, Ind.

[It now remains for slide-rule or instrument makers to bring out a logarithmically divided triangle for laying off such paper, so that the more expensive and less convenient slide rule will not have to be used for such work.—EDITOR.]



SHOWING MANNER OF LAYING OFF LOGARITHMIC CROSS-SECTION PAPER WITH SLIDE RULE

Why Did the Engine Stop?

On May 9, our high-speed slide-valve engine, directly connected to a 125-kw. direct-current generator, stopped while under full boiler pressure, with the throttle valve wide open. The valve was perfectly set and the cylinder and valve were well lubricated. The exhaust pipe and passages were also free of obstructions.

An effort to turn the engine over with starting bars was unsuccessful. Someone suggested that the flange couplings on the crankshaft be removed, and also the top half of the field and armature, to determine if the cause was magnetic or electric. We could not move the engine, even after the generator and shaft were dismantled. The parts were reassembled and four hours later the throttle was opened and the engine started. Full load was put on and the engine has run well ever since.

What caused our trouble?

GEORGE W. BULLIS.

Des Moines, Iowa.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Clayton Analysis of Indicator Diagrams

J. Paul Clayton has presented to the American Society of Mechanical Engineers a paper of much practical importance. Generally a steam engine is subjected to a test but once in its lifetime, to determine its compliance with the guaranteed performance. Mr. Clayton has made it possible to determine the steam rate with a fair degree of approximation with no other expense or trouble than that of taking and analyzing an indicator diagram. This, at least, if the foundation of his method is sound. It is the invention of the logarithmic diagram which has led to these consequences. The idea of plotting the logarithmic diagram is a real stroke of genius.

Mr. Clayton finds that the expansion and compression lines of the diagram are, for the normal case, true exponential curves and that there is a more or less definite relation between the exponent of the curve, which is simply determined from the logarithmic diagram, and the quality of the steam at the commencement of expansion. The values of the exponents and of the quality are the average of those taken on the two sides of the piston.

The fact that up to the present we have not been able to evaluate even approximately the initial condensation is explained by the number and the variety of factors which influence that quantity during the admission: The size, valve design, relative roughness of the interior surface, the temperature range, length of cutoff, speed, location of ports and port passages, quality of the steam supplied, and the jacketing and lagging. Mr. Clayton has overcome these difficulties by an entirely new departure—calculating the initial quality from its effect after all of these factors have operated; that is to say, after the admission has ceased and, according to the value of the exponent n. This is an admirable achievement if it can be demonstrated experimentally that, under the given circumstances, the exponent depends upon nothing but initial quality.

Unfortunately, the relations given between the exponent and the initial quality of the steam do not apply to cylinders provided with jackets, because the jacket influences the phenomena which follow the closing of the admission valve, and it is necessary, further, that the counter-pressure should be very nearly equal to that of the atmosphere. When the back pressure is raised to 30 lb. absolute, for instance, there is a new series of relations existing for the same initial pressure, due to a different temperature range in the cylinder and the consequent alteration of the phenomena occurring after cutoff. On account of these two influences during the expansion, it is necessary to examine each case in particular to determine the relation of the exponent and the quality of the steam at cutoff.

As this method is based entirely upon the information given by the indicator diagram, it is necessary that this should be taken with the greatest precaution to guarantee its exactness. In this respect, Mr. Clayton does not insist enough upon the nature of these precautions. To my mind they are of two sorts:

1. In that which concerns the volume, represented by the abscissas of the diagram. In this connection reducing motions with long cords are to be severely condemned on account of their whipping. The essential condition is that when the piston is exactly at 0.0, 0.1, 0.2, 0.3, etc., of its stroke, the pencil should be also exactly at the same relative point in its stroke. This condition cannot be realized if the reducing motion contains long cords which stretch and contract, or if rigid connections are allowed to have play. The reducing motion should be positive and should be inspected often for lost motion in all its joints, which should take account of the obliquity of the connecting-rods. It is necessary also that the clearance space should be exactly calculated or measured, or, better still, both calculated and measured, that the methods may check each other.

2. In that which concerns the ordinates, or pressures, one should not trust the scales furnished with the indicator, but calibrate the springs for himself with the greatest care, and frequently.

As the two strokes of the piston constituting one revolution are not exactly equal (for the clearance spaces, the degrees of expansion and of compression and the leads are not the same on the two ends) a diagram which has the average values of the two ends is ordinarily taken; and this average diagram, necessarily incorrect, one counts as though it were experimental. But the experimental consumption per revolution is divided into two equal parts with the result that the value of the quality at cutoff resulting from the average is not exact, which can be asserted in advance. What is the degree of error? What consequences will follow this error? It is difficult to answer these questions.

Only one method presents itself for obtaining the greatest exactness possible; that is, to employ two surface condensers, one for each end of the cylinder. On the experimental engine at the University of Liége, to avoid the expense of this complication, dealing, as I was with an unsympathetic administration, I had the valve-gear of the two ends made entirely independent so that they could be adjusted for exactly the same degrees of expansion and of compression, of lead and admission, and the piston was so adjusted as to give exactly the same clearance on both ends. In the course of my experiments, I found, however, differences between the diagrams from the head and crank ends, for which the only probable cause seemed to be the difference in velocity of the piston at the opposite ends of the cylinder, due to the obliquity of the connecting-rod. This convinced me of the necessity of a separate condenser for each end.

I cannot repose absolute confidence in Mr. Clayton's methods. They are only approximations, more or less exact, which, it is true, throw a great deal of light upon the practical operations of an engine and present perhaps more actual usefulness than theories that are more exact, but more complicated. In the issue of July, 1901, of the *Revue de Mécanique*, my assistant, Mr. Duchesne, and I published an article, the object of which was to demonstrate experimentally that the laws of expansion and compression differ from one another. It is evident that they should, since, during the expansion, ordinarily the metallic walls are furnishing heat to the working fluid, while during the compression, or, at least a great part of it, the superheated fluid is furnishing heat to the inclosing metal. All the operations and measurements were made with exceptional care

The ordinates and abscissas of the indicator diagram are indicated in figures in the article; also the logarithms of the same are given. It is easy therefore, to plot therefrom the logarithmic diagram. For the expansion, this logarithmic diagram has the appearance of

a straight line without exactly being one. For the compression the curve is so pronounced, especially near the end, that no confusion is possible. The commencement of a hook manifests itself at the end of the compression on the pressure-volume diagram. The novice would attribute this to leakage, but we know that it is condensation. The late George Duchesne demonstrated positively that we were right, by compressing in the same cylinder and under the same condition, noncondensible air with which there was no hook. The only reasonable objection that has been raised is that the slow speed of the engine, less than 60 r.p.m., for a 60-cm. stroke, magnifies the effect of the action of the cylinder wall. This objection is far from invalidating our claim that the law of compression in the steam engine is other than that of the expansion.

Even if it leads only to rough approximation, the work of Mr. Clayton is a step in advance that is certainly novel as well as important.

V. Dwelshauvers-Dery.

Liége, Belgium.

A Boiler Inspector's Experience

Your account of the boiler explosion at San Antonio, Tex., attributes it to the combination of a defective steam gage and a locked safety valve. Does it not seem as though anyone who would let such conditions exist is on a par with the fool who "did not know it was loaded"? This is only one of several similar accidents from the same cause in the past few years, notably at Pittsfield, Mass., and on the M. K. & T. R.R. The latter explosion was identical with the San Antonio disaster, although the loss of life was not so great, but the destruction of property was, and all because some shortsighted official thought he could save a few dollars by placing a worn-out and defective gage on the boiler.

While I was a boiler inspector some years ago, I seldom had less than five or six gages and as many defective or corroded safety valves to report weekly. Two cases I recall particularly, the first of which was in North Dakota. I inspected two of the three boilers and on trying one under steam pressure to see if the safety valve was free I found I could not lift it. This led me to take the valves off the idle boilers and test them with the feed pump. One valve opened at 205 and the other at 230 lb., enough pressure to disrupt either boiler. After testing and correcting the gages and slacking off three full turns on the adjusting bolt, the valves were set to operate at a safe pressure. I found on a return visit that the valves would not work at the pressure for which they were set, and no one at the plant knew who readjusted them, as frequent changes were made in the operating crew.

The second case was on a large locomotive boiler in a forge shop in western New York. This boiler was over a furnace used for heating billets and supplied steam at 120 lb. to a steam hammer and other units. At the time of my first visit, Sunday morning, I could not get into the firebox on account of its dirty condition, but I examined the boiler as completely as possible. I reported that the risk should not be carried. The insured would not accept this but arranged to clean out the firebox and take the dome head off if I would examine it again in two weeks; he felt sure I had made a mistake, as it was a first-class boiler, etc. The following is from the report of the defects found on a reëxamination, the external condition having been reported on the first inspection:

"The right side of the furnace has four patches of the following dimensions located at various points on the furnace sheet and all fastened with patch bolts: 15x12 in., 12x9 in., 11x8 in. and 12x6 in.; also three cracks with a total of eighteen ⅝-in. plugs in them. The left furnace sheet has two patches: 23½x11 in. and 16x14 in., also one crack 6 in. long, plugged.

"The back furnace sheet has four patches: 10x6½ in., 10x4 in., 11x5 in. and 16x6 in. This sheet is also badly corroded from the furnace door down to the mud ring and contains pits from $\frac{1}{16}$ to $\frac{3}{32}$ in. deep. In the door flange are three patches: 8x8 in., 6x4 in. and 8x7 in., also six fire cracks, plugged. The crown-sheet has one patch 13½ in. in diameter with two stay-bolts passing through it, and the additional disadvantage that the patch is on the fire side of the sheet. The flue sheet is badly corroded from the lower row of flues to the mud ring. On the right side of the water leg outside is one patch 6x6 in., and the whole is badly corroded. The joint between this sheet and the front of the water leg leaks, and the left side has one patch 8x7½ in. The back-door sheet shows rivets passing through the mud ring with the heads nearly all eaten away. Internally the boiler is heavily scaled on the tubes and crown-sheet.

"The boiler was given a hydrostatic test and at 140 lb. pressure the side sheets bulged considerably; at 155 lb. the crown-sheet let go in the bracing so that it was impossible to supply the leakage. The steam gage operated in jumps and showed a variation of from 5 to 30 lb. by the test gage. As this examination bears out the opinion formed of this boiler at the first visit, I recommend that the boiler be replaced at once or the insured dropped."

As the engineer told me the safety valve operated at 120 lb., I desired to see it do so, so I put no "gag" on the valve. The test gage showed 155 lb., and when the pump would supply no more water due to excessive leakage, the valve had shown no signs of blowing. I have found safety valves to stick in the best regulated plants, but as testing them is invariably hot and disagreeable, some engineers take a chance, trusting that when the steam gets to the blowoff point, they can check the draft or put the feed on to hold the pressure down.

F. J. Nayler.

Cos Cob, Conn.

The V-Thread on the Bolt

L. F. Brown's letter, in the May 14 issue, goes straight to the point in regard to the passing of the V-thread and the final standardization of nuts. I agree with him that this question has long caused much worry and confusion among those associated with machinery.

Simplicity and accessibility in the construction of a machine are desired by the purchaser, or engineer who is to operate it. Consistency demands that this question be given immediate attention and adjustment, since bolts and nuts are common to all machinery and ought to be interchangeable, size for size for all purposes. I cannot see why a square nut is made at all, since it is just as easy to stamp a hexagon nut as a square one. Of course, bolts and nuts of this style can be made in as many different grades of finish and material as they are now, so there is no real ground for objection on this point. It would be a pleasure to everyone to be able to renew a nut on the first trial even with any stray nut picked up in the plant, and also to remove all nuts of a certain size without changing wrenches, as Mr. Brown says.

It is becoming more and more rare for the average manufacturer to make the bolts, nuts and similar parts that enter into his finished product, and more and more are such parts being purchased from those making a specialty of producing them in large quantities. Consequently a sane standard should be established; constant protests will soon bring this about.

Lloyd V. Beets.

Nashville, Tenn.

Negative Lap on Double Eccentric Corliss Engines

In the May 21 issue under "Inquiries of General Interest," your reply to inquiries of "J. K." and "A. D. K.," regarding negative lap of steam valves on double-eccentric Corliss engines, is not in accord with modern practice nor with my experience with this type of engine.

There may be a few instances, especially on double-eccentric engines in the early days of their manufacture, when the

steam valves on Corliss engines were not given more than 3/32 in. to 1/16 in. port opening, or negative lap when the wristplate was central. With the amount of port opening in the central position which you suggest in reply to these inquiries, the latest possible cutoff would be about 55 per cent. of the stroke, which is not a "long range cutoff" from a present-day viewpoint. Corliss engines which have been kept abreast of the times by their manufacturers, and especially within the last 10 or 12 years, have a much larger port opening in the central position than you mention, which permits setting the steam eccentric at so small an angle ahead of the crankpin with a direct motion, that cutoff may take place quite late in the stroke, and a cutoff of 82 per cent. of the stroke or even a little more is easily obtained.

With care, the steam valves and steam eccentric can be so adjusted that the steam valve closes just before the exhaust valve at the same end opens. It is not contended that this is economical by any means, as terminal pressure may be as high as 85 or 90 per cent. of the initial pressure, but in modern practice conditions are such that at times engines are compelled to take care of peak loads that even 10 years ago would have been considered dangerous. But the condition must be met and it is generally considered more economical, all things considered, to waste steam on an overload for an hour or two during the day, than to provide the necessary equipment to take care of this excess load under economical conditions, and have the units running under a wasteful fluctuating underload for a greater period of time, or else stand idle for the greater part of the 24 hours. The great range of cutoff now used on engines in power houses, rolling mills and other severe work, is the result of a demand on the manufacturer caused by the conditions to be met.

With the ordinary single-eccentric engine, as soon as full load is exceeded by a small margin, the engine becomes a fixed cutoff engine, operating like a plain slide-valve engine having its speed regulated by a throttle valve (much the same as a locomotive), but which is always open to the limit; therefore when the load becomes so great that it over-balances the steam pressure, the speed reduces with serious results.

Thus was the double-eccentric developed on Corliss engines, which was improved so that engine speed is practically constant at all points of cutoff from no load to 60 per cent. or more overload. But to get this late cutoff to take care of peak loads, the angle of the steam eccentric ahead of the crankpin was gradually reduced to a very small angle, and as the angle of the eccentric was retarded, it was necessary to increase the port opening on the steam valves to a considerable extent, and also lessen the lap, thus ignoring the old injunction that unless a valve had considerable lap it would not be steam-tight.

I have set the valves on Corliss engines equipped with double eccentrics throughout a wide range of sizes and in no instance has the port opening with the wristplate central, been less than 1/32 in. which was on an 11x30-in. direct-connected engine running 165 r.p.m. The steam valves had 1/8-in. lap in extreme position.

On an engine with an 84-in. cylinder the valves were given no less than 1¾ in. port opening with a correspondingly small lap. It must be understood that with this late cutoff care must be taken that the valve-gear unhooks at all points of cutoff and that the dashpots seat properly; otherwise, the steam valve will remain open at any given end until the exhaust valve at that end opens, thus allowing steam to pass through the cylinder directly to the exhaust.

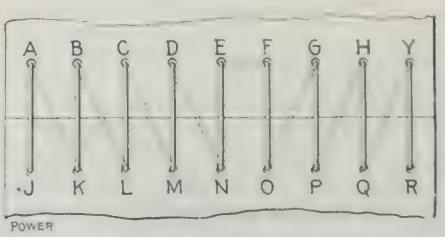
When I refer to the wristplate being "central" I refer to the position of the valve-gear when the valve is at the middle of its travel, with both steam valves hooked on. However, Corliss engines are becoming common now which do not have a wristplate at all, especially to drive the steam valves, these being connected together through their steam levers by a steam rod, and the hook rod with its releasing mechanism attached to the forward steam lever.

GEORGE H. WALLACE.

Racine, Wis.

Wire Lacing for Belts

Mr. Fries, in the issue of Apr. 16, speaks of the tendency of wire lacing to pull the ends of the belt out of line. W. G. Hawley's letter in the issue of May 28 shows that he noticed the same thing, but I think Mr. Hawley's lace will be more likely to bend, which in time will cause failure, than the one herewith illustrated. I have used this lacing with much success. The bending in the joint is very deleterious and the ends of the



STYLE OF WIRE LACING FOR BELTS

lacing should be very tight together, and the holes should not be any farther from the end of the belt than necessary. With Mr. Hawley's lace the wire is longer on the back from hole to hole than if it were straight with the belt. This tends to make the joint bend more easily.

I make all the holes first, using a punch about 1/8 in. in diameter and punching the holes ½ in. apart. In drawing the lacing tight with pliers, I always use them on the end of the lace only, as they would otherwise nick the wire and cause it to break. The end chewed by the pliers will be cut off, so there is no harm done in this way.

To lace this joint, double the wire in the middle and put the ends through adjoining holes, as *E* and *F*. From the back side pull both ends through till the loop is up tight on the belt, then take one end across on the face to the opposite hole as from *F* to *O*, then back to *F* and across to *O* again and then from *O* to *G* across the back, and so on to *R*. From *R* run to *Y* across the back, then through *Y*, and pull the wire tight in the direction of *R*. Finally cut off the ends fairly close and hammer them down with the rest of the lacing. Treat the other end the same way.

This style of lacing equalizes the lateral strain and prevents disalignment of the belt. I have also had success with annealed-iron wire when the regular wire lacing could not be obtained.

CLIFFORD SMITH.

Midland, Calif.

Standard Indicator Cocks

The editorial on standardizing indicator cocks is timely. I realize this very much as I have made a business of indicating engines as a side issue to my regular work, and in many engines I found that the indicator could not be attached without repiping, as sometimes there is no valve except the 3-way cock. I believe three standard sizes of connections are in use in the various makes of indicators, probably because each designer has his own idea as to what the size should be, and as a cock is furnished with each instrument, it is presumed that different sizes make no trouble.

Nearly all engine builders fit their new engines with indicator piping, and I would suggest that they get together and decide on one standard size of connection, and to accommodate indicators that are not made to that standard, furnish a nest of two or more bushings so that the same 3-way cock may be used with three indicators of different make. But if the indicator manufacturers get together with the engine builders, no extra bushings would be needed.

Under present conditions I find these bushings necessary, as often a job is lost through having to wait till some convenient time to repipe the engine. Some standard for indicator connections should be established.

J. CASE.

Hyattsville, Md.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Flame in Boiler Tubes

Do the gases of a boiler furnace continue to burn with flame in passing through the tubes of a boiler?

J. D. M.

Little, if any, when the tubes are clean, as combustion ceases as soon as the temperature of the gases is reduced to a certain point by transfer of their heat to the boiler. Incombustible matter, carried along like incandescent dust, may remain luminous after being swept along with the heated gases and be mistaken for flame of combustion after the combinations of combustion have ceased.

Boiler Joints

How many kinds of riveted joints are used in boiler work and what is the effective strength of each compared with the solid part of the boiler plate?

P. C.

The different kinds of boiler joints and their approximate efficiencies, as usually designed in good practice, are as follows:

	Per Cent.
Lap-joint, single-riveted, longitudinal or girth	57
Lap-joint, double-riveted, longitudinal or girth	74
Butt and double-strap joint, double-riveted	82
Butt and double-strap joint, triple-riveted	87
Butt and double-strap joint, quadruple-riveted	93

Approximate Rule for the Transmitting Capacity of Leather Belts

What rule can be given for estimating the number of horsepower which leather belts are capable of transmitting, under average power-plant conditions, where the belts run over cast-iron pulleys 24 to 60 in. in diameter?

G. B. D.

The horsepower which a belt is capable of transmitting depends on its tension, strength, speed and frictional adhesion to the pulleys. No exact rule can be quoted for use in all cases, because the conditions are variable, but a good rule for estimating the transmitting capacity of leather belts as used in average practice is to allow that one horsepower is transmitted per inch width of single-thick belt for travel of 785 ft. per minute, which corresponds to the peripheral speed of a 30-in. pulley making 100 r.p.m., from which the proportionate power due to different sizes and speeds of pulleys and widths of belt are readily calculated by the formula:

$$N = \frac{D}{30} \times \frac{S}{100}$$

where

N = Number of horsepower transmissible per inch width of single belt;

D = Diameter of pulley in inches;

S = Speed in revolutions per minute.

For double belts allow 90 per cent. more.

Smoke Stack Joints

For greatest durability of a sheet-iron or steel smoke stack which is exposed to the weather, should the girth joints be made with the lower end of each course set inside or lapping outside of the course below?

A. H.

Each girth joint should have the upper course set inside of the course below it, for when a course laps over the course below it, fine soot will sift through the joint from the inside of the stack, and becoming moist will cause discolored streaks of soot on the outside, forming a deposit which will eat away the stack paint and cause the material of the stack to corrode very rapidly.

Piston Rings and Leakage

Before it is cut, how much larger should a piston packing ring be turned, than the diameter of the cylinder in which it is to be used?

How can it be determined when the valve or piston rings of a simple slide-valve engine leak?

G. W.

Piston rings for cylinders up to 20 in. in diameter are generally turned solid about ¼ in. larger in diameter than the bore of the cylinder, and for larger cylinders the rings are turned about 0.1 per cent. larger than the bore of the cylinder, before being cut to spring them in place.

The best way to determine whether a slide valve leaks is to throw the crank on the dead center toward the cylinder, remove the cylinder head, disconnect the valve and set and block the valve so as to take steam only on the crank end of the cylinder and with the exhaust open to the head end. If the valve leaks, steam will escape from the port on the head end. If the piston leaks, steam will be seen to blow past the piston rings. For testing leakage of the piston at an intermediate point of the stroke, hold the piston at the desired point by blocking the crosshead.

Force Exerted by Expansion Contraction of Iron

How can the force exerted by the pansion or contraction of iron wl heated or cooled be estimated? If bolts which hold together the halves a flywheel hub are each 12 in. long 1½ in. in diameter, and are set up ti by a wrench, when the temperature of bolt is 212 deg. F., how much tighter each bolt hold after cooling down 60 deg. F.?

D. W

The force exerted by expansion wo be equal to that which would be quired as a compressive force to bri the expanded body back to its origi volume; and conversely, the force erted in contracting would be the sa as the force required to expand t contracted body to its original dime sions. The coefficient of longitudinal e pansion of iron being 0.0000067302 its length for one degree Fahrenhe then, as expanding from 60 deg. to 2 deg. would be through 152 deg., the line expansion of 12 in. of iron would be

$$152\ deg. \times 0.0000067302 \times 12 = 0.01228 + in.$$

From the formula

$$E = \frac{Pl}{K\lambda}$$

where

E = Modulus of elasticity (22,000 000 for wrought iron);

P = Tensile stress applied;

l = Length under consideration (in.);

K = Sectional area (for a 1½-i bolt = 1.767 sq.in.);

λ = Elongation in that leng (0.01228 in.).

Then

$$22{,}000{,}000 = \frac{P \times 12}{1.767 \times 0.01228}$$

$$P = \frac{22{,}000{,}000 \times 1.767 \times 0.01228}{12}$$

$$= 39{,}781\ lb.$$

This, then, will be the strain impos on each bolt in cooling from 212 de to 60 deg. F., to which must be added t strain produced in tightening the b with a wrench. Some compression a ways will take place reducing the di tance between the bearings of the n and bolt head, and the strain from co traction is thereby reduced by the pe centage which the compression is of t contraction in length that the bolt wou undergo with the reduction in temper ture if unrestrained.

Study Questions

This Week's Questions
Last Week's Answers

(51) While a hoisting engine is raising a load of 15,000 lb. at 200 ft. per minute, what is the mean effective pressure in the cylinder if its diameter is 20 in., the diameter of the piston rod 3½ in, the stroke 3 ft., the speed 60 r.p.m. and the mechanical efficiency, including the drum and sheaves is 70 per cent.?

(52) If the fuel consumption per indicated horsepower per hour is 1¾ lb., how many tons will be required in 24 hr. for the running of a 6000-hp. engine?

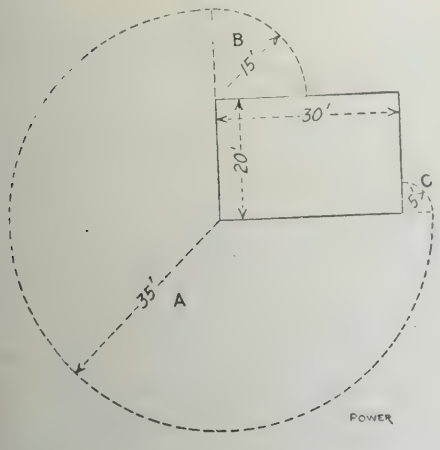
(53) Between the terminals of an unknown resistance x in an electrical circuit the drop in potential is 35 volts and between the terminals of a resistance of 300 ohms, the drop is 50 volts. What is the resistance x?

(54) To what uniform thickness can a 2-ft. sphere be covered with 1 cu.ft. of material?

(55) A steel standpipe for water is 185 ft. high and 20 ft. in diameter. How thick should be its lower plates if the joints have an efficiency of 75 per cent. and the allowed stress is to be 16,000 lb. per square inch?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(46) As shown by the diagram the extreme swing of the cable is over an area comprising ¾ of a circle of 35 ft.



radius, ¼ of a circle of 15 ft. radius and ¼ of a circle of 5 ft. radius. The area of a circle being πr^2

(47) Let

x = Age of steam engine;
y = Age of steam turbine;
z = Age of gas engine.

From the conditions of the problem

$$x = y + z \qquad (1)$$

$$z + 5 = \frac{y + 5}{2} \qquad 2z + 10 = y + 5 \qquad (2)$$

$$z + 10 = \frac{x + 10}{2} \qquad 2z + 20 = x + 10 \qquad (3)$$

Substituting value of x from (1) in (3)

$$2z + 20 = y + z + 10 \qquad (4)$$

Subtracting (2) from (4)

$$(2z + 20) - (2z + 10) = (y + z + 10) - (y + 5)$$
$$10 = z + 5 \qquad z = 5$$

Substituting value of z in (2)

$$10 + 10 = y + 5 \qquad y = 15$$

Substituting values of y and z in (1)

$$x = 15 + 5 = 20$$

Therefore the steam engine has been in service 20 years, the turbine 15 years and the gas engine 5 years.

(48) The boiling point of water is 212 deg. F. Then 212 — 62 = 150 deg. is the rise in temperature. At 62 deg. F., 1 gal. of water weighs 8.3356 lb. Since 1 B.t.u. = amount of heat required to raise 1 lb. of water 1 deg. F.

$$8.3356 \times 150 = 1250.34 \text{ B.t.u.}$$

is the heat energy required and 1 B.t.u. being equivalent to 778 ft.-lb.

$$1250.34 \times 778 = 972{,}764.52 \text{ ft.-lb.}$$

is its equivalent in units of work.

(49) The volume of metal will equal the area of the cross-section multiplied by the length. The area of a circle being ¼ πD^2 or 0.7854 D^2, the cross-sectional area is

$$(0.7854 \times 6.625^2) - (0.7854 \times 5.75^2)$$
$$= 0.7854\,(6.625^2 - 5.75^2) = 8.5044 \text{ sq.in.}$$

Then

$$\frac{8.5044}{144} \times 12 = 0.7087 \text{ cu.ft.}$$

is the volume of metal and

$$0.7087 \times 480 = 340.176 \text{ lb.}$$

is the weight.

(50) The formula for the space covered by a body falling freely from rest is

$$s = \tfrac{1}{2} gt^2$$

where

s = Space in feet;
g = Acceleration of gravity = 32.2 ft. per second;
t = Time in seconds.

Substituting

$$s = \tfrac{1}{2} \times 32.2 \times 100 = 1610 \text{ ft.}$$

The velocity formula under the same conditions is

$$v = gt$$

where

v = Velocity.

Substituting

$$v = 32.2 \times 10 = 322 \text{ ft. per second}$$

Heating and Ventilating Engineers Meet in Detroit

Hotel Tuller was headquaraters for the American Society of Heating and Ventilating Engineers during the fifteenth semiannual convention held in Detroit, July 11, 12 and 13. In the absence of President John R. Allen, First Vice-President John F. Hale presided and delivered the address of welcome, concluding with some mention of the object and aims of the society and the advance made during the last six months. The membership now totals 444.

Mr. Hale continued by reading a paper sent by the president from Constantinople on "Heating in Turkey," in which Prof. Allen showed the progress now being made in Turkey as evidenced by the erection of modern office buildings and improved methods of heating. Most Turkish houses are heated by open charcoal fires, or perhaps by a mongol, but in the business section of Constantinople several new buildings have been equipped with modern heating arrangements, and almost all are hot-water systems. As a rule the pipes are small and their arrangement crude, but fortunately not much heat is required in Turkey. Heating a number of buildings from a central plant is new to this country but two installations of this kind are now in progress, one at Armeautkeny in the American school for girls, and the other the writer is installing at Robert College.

Following some routine business, communications were read from the Illinois, New York and Massachusetts chapters, each reporting what had been done since the last annual meeting in January and the topics that had been discussed at the various meetings. Committees on Testing of House-Heating Boilers, Standard Outside Weather Temperature, Ventilating Standards for Motion Picture Theaters, Tests, Compulsory Ventilation and Heating Guarantees reported progress and promised complete reports at the next annual meeting.

In response to an invitation from the National District Heating Association, the convention instructed the board of governors to appoint a committee of five to work with a similar committee from the other society along educational lines, choosing their own subjects and taking up the most urgent current topics.

The committee on the 1912 standard for pipe flanges and flanged fittings spoke of the attitude of some of the manufacturers and their hesitancy about adopting the new standard. A resolution was passed recommending that members of the society and other engineers use and specify the 1912 standard. The election of a nominating committee for officers for 1913 completed the business session.

The tellers reported the following names: George Mehring, Chicago; Joseph Graham, New York; H. W. Whitten, Boston; C. F. Newport, Chicago, and F. R. Still, Detroit.

Papers

In his paper on "Temperature Equivalents of Wind Velocities," H. W. Whitten showed, from a comparison of records that one mile of wind movement per hour, between 40 and 15 deg. plus, requires about the same amount of heat supply as 1 deg. drop in temperature. From 15 to 20 deg. minus, the equivalent of one mile wind velocity is 1.15 deg. This is for ordinary buildings constructed without protected windows. The impact effect of wind of uniform velocity, with the barometer constant, was reported to increase 0.4 per cent. per degree drop in temperature.

"Open Windows with Mechanical Ventilation" was the subject of a paper by Ralph C. Taggart. He sides with neither the advocates of open-window ventilation alone, nor those favoring mechanical ventilation without the use of windows, for the author believes that no ventilating system should be installed, the balance of which is appreciably affected by opening windows. The system should give as good, or better results, when used in conjunction with open windows than in any other way, and the engineer who insists on closed windows is making almost certain the unsatisfactory operation of his apparatus. The author discussed to some length both sides of the question and ended with the above conclusion.

In his paper on "The Time Element in Heating Apparatus," James A. Donnelly presented several charts, showing the time required to heat a cold radiator, the variations in temperature in a concrete factory building and temperature changes in a fan-operated drying room.

John D. Small presented a paper in question-and-answer form on "Office Practice in Estimating Heating and Ventilation," including the subjects of estimating radiation, air changes per hour, hot-air heating and ventilation vs. tempered air and direct radiation, mains as heating surface or covered throughout, vacuum devices in heating systems, standard methods of estimating heating and ventilation, the placing of radiation, and window construction.

Samuel R. Lewis described the heating and ventilation arrangements of the factory of the Defiance Tick Mitten Co., in Toledo. The blast system is used. Fresh air is supplied through special steel casement sash and is drawn through tempering coils and steam-jet humidifiers to the supply fan. The steam supply to the tempering coils is controlled by a thermostat and an air temperature of 60 deg. is maintained. The steam jets are controlled by a sensitive element operating within a range of 5 per cent. from the point at which it is set (about 40 per cent. relative humidity). A minimum of 30 cu.ft. of fresh air per occupant per minute is provided.

A paper on "Humidity in Relation to Heating and Ventilation," by L. C. Soule, dealt briefly with humidifiers and air washers and the power required to provide proper humidity. A paper along the same lines was presented by J. I. Lyle, his topic being "Methods of Automatic Humidity Control for Air Washers." The author describes a system of automatically regulating the humidity by controlling the dew point, or saturation temperature of the air leaving the washer. Saturation is produced by heating the spray water and varying its temperature as much as necessary to maintain a constant dew point under variable conditions of the entering air. Analyses were made with the assistance of the Carrier psychometric chart of the various methods which have been proposed, showing their fallacies, either in being wrong in theory, or impossible practically. Attention was also given to the practical difficulties encountered with bypass dampers and the regulation of the steam supply to the tempering coil.

In a paper on "Brick Drying," H. C. Russell discussed dryers of the so-called progressive type, considering capacity of the machine, time and temperature for drying, construction of kilns and dryer, the heat losses including the British thermal units required to evaporate the moisture, the heat absorbed by the cars and clay, and that lost by radiation; also the amount of waste heat available. To dry each brick requires about 4200 B.t.u. and during the process of waste heat drawing, about 3200 B.t.u. may be recovered, so that only 30 per cent. above the amount obtained from waste heat is required. These figures are based upon the most extreme conditions with an outside temperature of 35 deg.

Following the above paper an animated discussion was started by D. M. Quay on "Upward vs. Downward Ventilation in Theaters." Mr. Quay condemned the mushroom system of introducing air under the seats and drawing it out at the ceiling on account of the contamination resulting from contact with the clothing and perspiration on the occupants. The system should be reversed: fresh air introduced above the breathing line and drawn out at the floor.

S. R. Lewis objected to the downward system on account of opposing currents set up by heating or glass surfaces or perhaps the occupants themselves interfering with the circulation. He favored upward movement as did the majority of those discussing the question.

A. M. Feldman described the ventilating system of a new dispensary building attached to the Lebanon Hospital, New York City. Radiators in the clinical rooms are arranged, direct-indirect, that is, part of the sections of each radiator are inclosed. Fresh air is drawn through screened openings under the windows, passed over the inclosed part of the radiator and out into the corridor through louvers in the bottom panel of the doors, air flow in this direction being induced by exhausting the air from the corridor. The corridor serves as a waiting room for patients, and on account of the numbers usually present requires thorough ventilation.

"Removal of Refuse and Waste by Fans and Blowers," was the subject of an interesting paper by F. R. Still. For economical operation it is very important to know the minimum velocity or volume of air required to convey substances of varying specific volumes and densities as the frictional loss multiplies directly as the square, and the power to drive the fan directly as the cube of the velocity. Mr. Still gives a general insight into the methods which have been employed for years and points out opportunities for improvement. He takes up the general scheme and sizes of piping, the size of fan, proportion of dust separators and the proper form and construction of hoods.

The final paper, by J. M. Stannard, dealt with the heating and ventilating of the Northwestern University buildings at Evanston, Ill., which is accomplished by hot-water forced circulation and direct and indirect steam plants under both vacuum and gravity methods. The paper describes the equipment and arrangement of the plants and includes a table giving operating costs, favoring the steam system.

Entertainment

Detroit's reputation for entertainment was upheld at the present convention. There was plenty to see and do and the committee provided an excellent program. On Thursday afternoon the ladies were taken to Bois Blanc and in the evening all went to Belle Isle. Friday's entertainment consisted of automobile sight-seeing trips for the ladies and in the afternoon the men inspected the plants of the American Blower Co. and the Ford Motor Co. The entire delegation of 110 met for dinner at the Detroit Golf Club where special entertainment, followed by dancing, was provided. Those remaining until Saturday morning rode out to the Delray station of the Detroit Edison Co. and were served luncheon at the plant.

In England it is proposed to erect a memorial at Liverpool to form a permanent record of the heroism of the engine-room force of the "Titanic." The money will be raised by popular subscription. It is fittingly suggested that the memorial be placed somewhere on the bank of the Mersey River.

Massachusetts State N. A. S. E. Convention

The seventeenth annual convention of the Massachusetts State Association of the National Association of Stationary Engineers was held in Northampton, July 11, 12 and 13, with headquarters at the Draper House.

There were about fifty delegates in attendance and many visiting engineers. The several business sessions of the convention were held in Carnegie Hall, located close to the hotel. The main floor of the City Hall was tastefully arranged for the display of power-plant machinery and supplies under the direction of the New England Association of Commercial Engineers. The exhibit opened at noon on Thursday.

At 10 o'clock on Thursday evening the convention was entertained with a smoker given by the supplymen at Rahr's Inn. Interesting talks and vocal and instrumental music combined to make an interesting program.

At 11 o'clock Friday morning at Carnegie Hall the formal opening of the convention took place, Charles H. Hill, chairman of the local committee presiding.

William H. Feiker, Mayor of Northampton, gave the company a warm welcome to the city, and Past National President Edward H. Kearney responded on behalf of the association.

Frank E. Davis, president of the Board of Trade of Northampton, spoke upon the industries and beauties of the city. He was responded to by Claude D. Allen, president of the New England Association of Commercial Engineers.

Past National Presidents P. H. Hogan and T. N. Kelsey also made addresses. After the chairman made announcements and appointed committees, the meeting adjourned until 2:30 p.m. On Friday afternoon an automobile trip was taken through the city under the auspices of the Board of Trade, and in the evening an entertainment was given at Cook's Theater.

Saturday afternoon, headed by Charles Van de Wall and the Northampton Brass Band, the convention marched to the Driving Park to enjoy a clam bake. The baseball game between the engineers and supplymen was won by the former with a score of 10 to 5.

During the convention, James Sumner, retiring state president, was presented a handsome silver service. Frank E. Davis was given a tourist's hand bag and outfit, and Jack Armour, of POWER, received a gold signet ring.

The newly-elected state officers are J. T. Maloney, president; A. M. Day, vice-president; Ole B. Petersen, secretary; Walter H. Damon, treasurer; George W. Furbush, conductor; Arthur A. Austin, doorkeeper; Thomas H. Ray, state deputy.

The New England Association of Commercial Engineers elected officers as follows: Albert C. Ashton, president; Frank A. Morrison, vice-president; Herbert E. Stone, treasurer; Claude D. Allen, W. K. Campbell, John B. Harrington, W. C. Barnes, John D. Stiles, William Coppins, W. D. Hasley, B. L. Ames and W. H. Walker, directors.

EXHIBITORS

Autogenous Welding Co., the Roto Co., *Southern Engineer*, Patterson Lubricating Co., Lagonda Manufacturing Co., Universal Supply Co., Gardner Grate Co., Curry Brothers Oil Co., Quaker City Rubber Co., Johns-Manville Co., Burke Engineering Co., American Steam Gauge & Valve Co., Hart Packing Co., Jenkins Brothers, *National Engineer*, V. D. Anderson Co., McLeod & Henry Co., Hyland Manufacturing Co., Mason Regulator Co. Philadelphia Grease Co., the Texas Co., A. W. Chesterton Co., Durabla Manufacturing Co., Dearborn Drug & Chemical Works, Perfection Grate Co., Walter G. Ruggles Co., Monarch Valve Co., Evans Mill & Supply Co., Manning, Maxwell & Moore, Robb Engineering Co., Lunkenheimer Co., American Steam Packing Co., Ashton Valve Co., W. R. Winn, Greene, Tweed Co., Schaffer & Budenberg Manufacturing Co., Enterprise Rubber Co., Peerless Rubber Co., Holyoke Valve & Hydrant Co., George W. Knowlton Rubber Co., Keystone Lubricating Co., Adam Cooks' Sons, Economy Lubricating Co., Chicago Pneumatic Tool Co., POWER, Charles A. Claflin & Co., Strong Machine & Supply Co., Garlock Packing Co., Eagle Oil & Supply Co., Home Rubber Co., Nonotuck Silk Co., Northampton Silk Co., *New England Engineer*, Chapman Valve Co.

The city of Lawrence was chosen for the 1913 convention.

Manufacturrs Adopt Flanged Fitting Schedule

At a meeting of the Committee of Manufacturers on Standardization of Fittings and Valves, held at 30 Church St., New York City, July 10, 1912, the "Manufacturers" 1912 schedule of flanged fittings and flanges was adopted to take effect Oct. 1, 1912. There was practically no opposition among the manufacturers present to the adoption of this schedule, but one vote being recorded against it. Copies of this schedule will be printed and distributed to the manufacturers and the trade generally as soon as possible.

New list prices for brass and iron-body swing-check valves, standard and extra heavy, were adopted at this meeting, to take effect Oct. 1, 1912, copies of which will be printed and distributed to the trade as promptly as possible.

Flywheel Explosion in Brooklyn, N. Y.

At 9 a.m. Saturday, July 6, the 16-ft. by 25-in. flywheel of a 20x38-in. Watts-Campbell Corliss engine exploded in the National Sulphur Works plant, at 618 Kent Ave., Williamsburg, Brooklyn, N. Y. Contrary to daily-press reports, the boilers did not explode as the result of being hit with a piece of the wheel.

The engine is located one floor below the street and directly beneath a room in which the employees handle the sulphur. On the level with the engine room and separated from it by a 12-in. brick wall, is another work room. Fortunately, no workmen were in this room at the time of the accident, as most of the wheel went through the wall into it, but three men on the floor above, as a large piece of the wheel crashed through it, were thrown 20 ft. through a doorway into the street. They were slightly injured.

The cause of the explosion is not known, even by the engineer, but it is assumed that the governor belt-lacing broke and that the pin was left in the governor, not allowing it to close the admission valves, as the engine ran away.

The wheel weighed about 17,000 lb. and every arm was broken close to the hub. The large driven pulley back of the head end of the cylinder was likewise broken and the shaft moved back 4 in. The eccentric strap and wrist plate were also broken, the crank-end dashpot plunger pulled from its cylinder, and the entire engine moved about 1 in. in a line parallel with itself.

Photographs of the wreck are unavailable, as the company desired that none be taken. Four years ago a similar accident happened, and it is understood that still another occurred seven years ago.

Court Sustains Long Acre Co.

On July 11, by a vote of three to two, the Appellate Division of the Supreme Court sustained the Public Service Commission ruling giving the Long Acre Electric Light & Power Co. authority to issue $2,000,000 of bonds after $1,000,000 of new stock has been subscribed and paid for, and an additional issue of $2,000,000 of bonds after another $1,000,000 stock issue has been subscribed and paid for.

The order also provided that the bonds were to be disposed of only at public sale, and at not less than 90 per cent. of their real value, and also that the proceeds of the stock and bonds were to be expended only after an itemized bill for expenditure had been submitted.

The case was taken before the court by the New York Edison Co. on an order to review the commission's ruling on the ground that the commission had no power to authorize the stock and bond issue, and for the further reason that since the Edison company is furnishing electricity at reasonable rates there is no occasion for a competitor to enter the field.

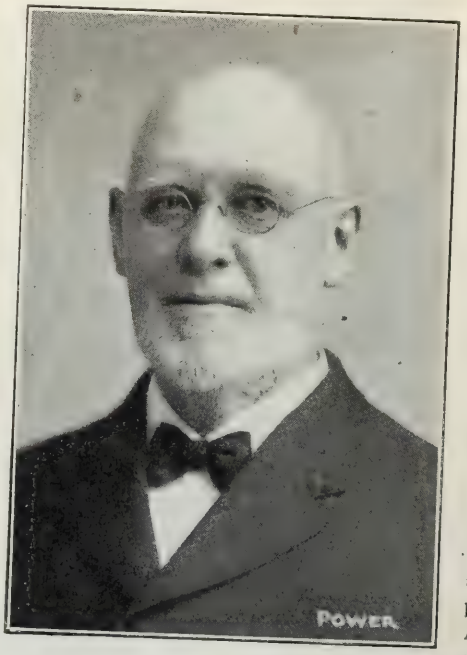
The case will now go to the Court of Appeals and if the decision of the Appellate Division is sustained, the Long Acre company will begin active business as soon as it is financed.

OBITUARY

George R. Babbitt

George Rodney Babbitt, the well known consulting mechanical engineer, died at his home, 130 Shaw Ave., Edgewood, R. I., on June 29, in his seventy-first year.

Mr. Babbitt was born in Berkley, Mass., and was educated in the local public schools and at the Hermitage Home



George R. Babbitt

School, Fall River. In 1860, he sailed around the Horn to San Francisco in the "Sierra Nevada" and later was second mate in the "Maria Theresa," one of the fleet of New Bedford whalers sunk off the southern ports in the early Civil War.

In April, 1862, Mr. Babbitt entered the employ of the Burnside Rifle Works, Providence, R. I., and became a subcontractor. Early in 1866, he was employed by William A. Harris, the Harris-Corliss steam-engine builder, filling the positions of machinist, foreman and superintendent. Mr. Babbitt was a natural mechanic, never having served an apprenticeship. While in this engine shop he invented and patented several improvements, one of which was a piston packi for which he received a medal at t Cincinnati Industrial Exposition, in 187

In 1888, he declined the superinte dency of the heavy-ordnance departme of the Government, a Washington, D. He retired from business on Mar. 3 1891, having completed a quarter of century in building and designing e gines. Mr. Babbitt invented a releasi gear for the Corliss engine, also an e tended piston. In April, 1892, he boug an interest in the American Oil C Providence, R. I., which connection he r tained until his death.

Mr. Babbitt was a member of the Ame ican Society of Mechanical Engineers an the National Association of Stationar Engineers, and was commodore of th Edgewood Yacht Club.

A. W. Harris

On June 30, A. W. Harris, presider and general manager of the A. W. Harri Oil Co., Providence, R. I., died at hi home in that city after a brief illness His widow and a daughter survive him.

In the early eighties Mr. Harris bega the manufacture of high-grade steam en gine and cylinder oil, and was amon the leaders in that line. When the auto mobile industry first attained prominence more than a decade ago, he began t supply high-grade lubricants for gasolin motors.

Mr. Harris had a wide reputation i business for integrity, and a large socia acquaintance. He was descended from a long line of Americans who have done things in the mechanical world, among them his uncle, William A. Harris, who built the Harris Corliss engine.

PERSONAL

Albert C. Ashton, treasurer of the Ashton Valve Co., Cambridge, Mass., has been nominated by the New England Association of Commercial Engineers for president. Mr. Ashton is a member of the American Society of Mechanical Engineers, Engineers Blue Room Club, Railway and Supply Manufacturers' Association, and several boards of trade.

Byron T. Gifford, formerly with the Central Station Engineering Co., Chicago, Ill., is manager of the recently inaugurated general engineering department of the American District Steam Co., North Tonawanda, N. Y. James A. Bendure, of the Economy Light, Fuel & Power Co., Lockport, N. Y.; A. E. Duram and C. A. Gillham, of the Central Station Engineering Co., Chicago, have resigned and become associated with the American District Steam Co. in its contracting department. The company's Chicago offices have been moved from the Monadnock Block to the ninth floor of the First National Bank Building.

POWER

Vol. 36 · NEW YORK, JULY 30, 1912 · No. 5

The Drink Question

HOWEVER much an engineer may strive to advance by study, or work, he is severely handicapped, if he drinks. Entirely aside from moral considerations there are powerful arguments against the immoderate use of liquor, and by this is not necessarily meant getting intoxicated. At best drinking is expensive, not only in money, but in time that might be better employed. The most that has ever been said in its favor is that it promotes good fellowship, but unfortunately that is very apt to degenerate into the acquisition of undesirable friends.

The necessity, not only of sobriety, but of total abstinence on the part of those whose duties are so important to the lives of others as those of the locomotive engineer, is universally appreciated. Only four weeks ago 46 lives were lost in a train wreck for which an intoxicated engineer is believed to have been responsible.

The most striking testimony against drink that we have read in a long time is that of John L. Sullivan, the ex-champion pugilist, in an interview in a recent issue of the *New York Sunday World*. Coming as it does from a man who has no prejudices in the matter, it is especially worth attention. Asked what was the greatest battle he ever fought he replied, "My battle with the Black Bottle." Then followed an intensely interesting story of his struggle when he quit in 1905 after 30 years of hard drinking. He acknowledged spending nearly half of all he ever earned for drink, that it prejudiced thousands of people against him, lost him hundreds of thousands of dollars in contracts, and would have been a great injury to his health but for his constant training. With it all he claims he never had the appetite, but that he drank "like most people do—for convivial ity's sake, to be a good fellow."

After he quit he says: "I soon found I was feeling better. My head was clearer; my stomach better. After I had cut out the booze, I was free from ever the slightest ailments." He disclaims being a temperance crank, but, as he puts it, "The man who drinks is a ——— fool." His concluding advice was: "The man, and especially the young man, who has any ambition, and hopes to succeed in life, has got to knock out the Black Bottle. The sooner he does the sooner he will succeed."

The effect of booze on the power plant of man is a decreased efficiency. Men, if the Black Bottle blocks your way, give it a knockout wallop.

Compression in Engine Cylinders

By W. J. Attick*

Under certain conditions, compression is valuable as a means of regulating the ratio of expansion and inducing quiet running. The mathematical analysis is included to trace the action of the steam within the cylinder. Its comprehension, however, is not essential to an understanding of the conclusions.

When a perfect gas expands at constant temperature its pressure is represented by a curve whose equation is $PV = C$. In the ideal indicator diagram, Fig. 1, in which the expansion curve V_1V_2 is of the form PV = Constant, if any point D in this curve is taken, the product of the pressure and the volume is the same as the product of the pressure and the volume at any other point, as

$$PV = P_1V_1 = P_2V_2 = Constant$$

This constant varies for different temperatures and different gases.

Let the lower line, Fig. 1, represent the line of zero pressure or P_0, and the next one above, the line of atmospheric pressure or P_a. The total work done during one stroke is represented by the total area included between the admission and the expansion lines and the line of zero pressure; that is, the area $AV_1V_2P_0ECA$. That part of the diagram $ACEV_1$ under the admission line represents the work done during admission and its area is $P_1 \times V_1$. The area under the curve V_1DV_2 represents the work done during expansion. This can be found by any one of three ways: by using a planimeter; dividing the area into a number of equal parts by vertical lines, finding their mean height and multiplying this by the length of the area measured; or by means of a formula derived by calculus which states that the area under the curve is equal to $P_1V_1\ log.e\ r$, the values P_1 and V_1 being respectively the pressure and the volume at cutoff and $log.e\ r$ the hyperbolic logarithm of the ratio of expansion. The ratio of expansion r is equal to

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

This formula applies only to a curve of the form PV = Constant. However, in a steam engine with well fitting valves and piston, the steam in expanding follows this law so closely, due to an interchange of heat between the steam and the metal, that no great error is made in applying it. The compression curve of an indicator diagram also closely follows the same law.

In an ideal diagram for an engine without clearance or compression, the total work done during one stroke of the engine is the sum of the areas under the admission line AV_1 and the expansion curve V_1V_2, or

$$P_1V_1 + P_1V_1\ log.e\ \frac{P_1}{P_2}$$

The useful work is the total work minus that done on the steam during the return stroke. The latter is represented by the area under the back-pressure line P_2B and is equal to $P_2 \times V_2$. But

$$P_2V_2 = P_1V_1$$

hence the useful work is

$$P_1V_1 + P_1V_1\ log.e\ \frac{P_1}{P_2} - P_1V_1 =$$

$$P_1V_1\ log.e\ \frac{P_1}{P_2}$$

This is the maximum useful work obtainable from a perfect steam engine, using steam between the pressures P_1 and P_2. But steam engines are not perfect, and one of the greatest causes of loss is due to clearance. If an engine has no clearance, then for every inch of the cylinder that is filled with steam during admission the piston moves 1 in. and a proportional amount of work is done, but if there is clearance it must be filled with steam at admission pressure before any work can be done, and this steam performs work only during expansion; therefore the work it is

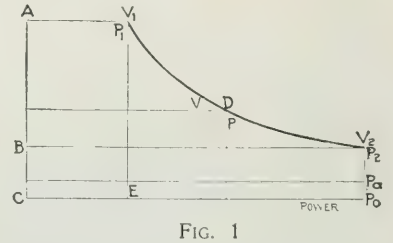


FIG. 1

capable of doing during its admission to the cylinder is lost.

It is common practice to fill the clearance with exhaust through compression and thus save whatever live steam would be necessary for this purpose. Just how much compression compensates for the loss due to clearance has not yet been decided definitely. To show the loss of useful work due to clearance, consider Fig. 2. This diagram represents the work done if there is clearance but no compression. The exhaust steam which fills the clearance space at the end of the stroke is compressed by the entering steam so that for the most part the clearance space is filled with live steam. To understand this action, suppose that when the piston is at the end of the stroke as in position D, Fig. 2, there is a thin diaphragm close against the piston and that the live steam is admitted between the piston and this diaphragm. The space Vc representing clearance, is filled with exhaust steam and as the live steam enters between the diaphragm and the piston, the former is forced back into some such position as is represented by the dotted line. The space C_1 may then be considered as filled with exhaust steam compressed to initial pressure, and the space C as filled with live steam.

To find the portion of the clearance space filled by live steam, consider the exhaust compressed along the line AB, which would be a curve following the same law as the expansion curve or PV = Constant. In Fig. 2, V_1 is the volume of steam admitted up to the point of cutoff and is the same as V_1 in Fig. 1; Vc is the clearance space; V_4 the part of the clearance space filled by live steam; V_3 that filled with compressed exhaust; V_5 is the total volume of steam in the cylinder at cutoff, and V_2 the total volume after expansion. It is assumed that expansion is carried to back pressure.

The work done during the admission period would be

$$P_1\left(V_1 - Vc + Vc\frac{P_2}{P_1}\right) =$$

$$P_1V_1 - P_1Vc + P_2Vc$$

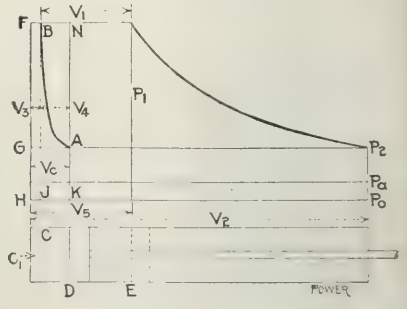


FIG. 2

The total volume of steam at cutoff would be

$$V_5 = V_1 + Vc\frac{P_2}{P_1}$$

and the work done during expansion would be from the general formula, $PV\ log.e\ r$,

$$P_1\left(V_1 + Vc\frac{P_2}{P_1}\right) log.e\ \frac{P_1}{P_2} =$$

$$P_1V_1\ log.e\ \frac{P_1}{P_2} + P_2Vc\ log.e\ \frac{P_1}{P_2}$$

The total work during one stroke is represented by

$$P_1V_1 - P_1Vc + P_2Vc + P_1V_1\ log.e\ \frac{P_1}{P_2} +$$

$$P_2Vc\ log.e\ \frac{P_1}{P_2}$$

The negative work done during exhaust would be

$$P_2(V_2 - Vc) = P_2V_2 - P_2Vc = P_1V_1$$

The useful work done during one stroke would be

$$P_1V_1 - P_1V_c + P_2V_c + P_1V_1 \log.e \frac{P_1}{P_2} +$$

$$P_2V_c \log.e \frac{P_1}{P_2} \qquad P_1V_1 = -P_1V_c +$$

$$P_2V_c + P_1V_1 \log.e \frac{P_1}{P_2} + P_2V_c \log.e \frac{P_1}{P_2}$$

This expression may be considered as made up of two parts

$$\left(P_1V_1 \log.e \frac{P_1}{P_2}\right)$$

and

$$\left(P_2V_c + P_2V_c \log.e \frac{P_1}{P_2} - P_1V_c\right)$$

The first part is the expression for the useful work done with the volume V_1 of live steam when there is no clearance; the latter part may be negative or positive so far as the expression is concerned, depending on whether

$$P_1V_c \text{ or } \left(P_2V_c + P_2V_c \log.e \frac{P_1}{P_2}\right)$$

is the greater. If the value of this expression is negative, then the useful work obtained from a given volume V_1 of

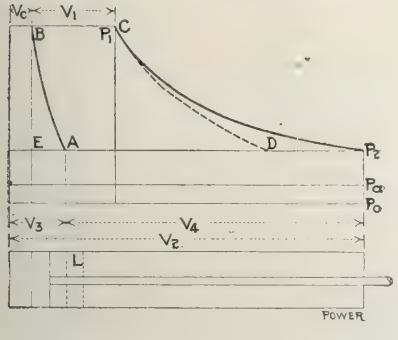


FIG. 3

steam in an engine with clearance is less than that obtained from the same amount of steam under the same conditions except with no clearance.

By referring to Fig. 2, it may be seen that

$$P_2V_c \log.e \frac{P_1}{P_2} = \text{area } ABJKA$$

so that

$$P_2V_c + P_2V_c \log.e \frac{P_1}{P_2}$$

represents the area *FHKABF* and is always less than P_1V_c by the area *ABNA*. This area represents the useful work obtained from a volume V_1 of live steam in an engine with clearance, less than that obtained under the same conditions but without clearance.

That compression under certain conditions compensates for the loss due to clearance is shown by Fig. 3, which represents the work done during one stroke with a volume V_1 of live steam but with compression to initial pressure. When the piston on the return stroke arrives at the position *L*, shown in dotted lines, the exhaust valve closes and the steam in the cylinder is compressed along the line *AB*, so that when the piston is at the end of the stroke the pressure is equal to the initial pressure. Then consider, as before, a diaphragm placed against the piston and that the live steam is admitted between the diaphragm and the piston. As the pressure in the clearance volume is equal to the initial pressure, the diaphragm will not move during admission, and the only live steam required is that necessary to fill the space back of the piston from the beginning of the stroke to the point of cutoff. The work done during admission would be P_1V_1. The total volume of steam in the cylinder at cutoff would be $V_1 + V_c$ and the work done during expansion would be

$$P_1(V_1 + V_c) \log.e \frac{P_1}{P_2} = P_1V_1 \log.e \frac{P_1}{P_2} +$$

$$P_1V_c \log.e \frac{P_1}{P_2}$$

The total work then is

$$P_1V_1 + P_1V_1 \log.e \frac{P_1}{P_2} + P_1V_c \log.e \frac{P_1}{P_2}$$

The negative work during the return stroke would be that during the exhaust period plus the work done in compressing the steam. The work during the exhaust period would be

$$P_2V_1 = P_1V_1$$

The work of compression is

$$P_2V_c \log.e \frac{P_1}{P_2} = P_1V_c \log.e \frac{P_1}{P_2}$$

The total negative work then is

$$P_1V_1 + P_1V_c \log.e \frac{P_1}{P_2}$$

and the useful work would be

$$P_1V_1 + P_1V_1 \log.e \frac{P_1}{P_2} + P_1V_c \log.e \frac{P_1}{P_2} -$$

$$P_1V_1 - P_1V_c \log.e \frac{P_1}{P_2} = P_1V_1 \log.e \frac{P_1}{P_2}$$

This is the same expression that was derived for the useful work under the same conditions and the same volume V_1 of live steam except that in the first case there was neither clearance nor compression. It is seen that, working under theoretical conditions, expanding to back pressure and compressing to initial pressure will neutralize any loss due to clearance.

This is true, however, only if the mechanical efficiency of the engine is perfect. The steam in the clearance space in expanding furnishes the power necessary to compress it. If the expansion is not complete it will not do this.

It will be noticed that in each of the three cases discussed the negative work done during the exhaust period is the same, $P_2V_2 = P_1V_1$, and this is also true only when the expansion is complete. As the useful work represented in Figs. 3 and 1 is equal, the areas AV_1V_2B (Fig. 1) and $ABCP_2$ (Fig. 3) are equal, and if the expansion curve of Fig. 1 is drawn on Fig. 2, as shown by the dotted line *CD*, then length P_2D will equal *AE*, and area *ABEA* will equal CDP_2C.

In an engine taking steam full stroke, $V_1 = V_2$ and the useful work done would be

$$P_1V_1 = P_1V_1 = cP_1 = P_1V_1$$

with neither clearance nor compression. If in this case there were clearance and compression, the clearance space would be filled with compressed exhaust steam at the expense of energy taken from the flywheel, and as there is no expansion this compressed steam could do no work.

In ordinary cases, steam is not carried for full stroke nor expansion to the back pressure, although the direct-acting pump is an instance of the former, and an automatic cutoff engine running light, of the latter. As with complete expansion, compression to initial pressure completely neutralizes the loss due to clearance, and with no expansion, compression increases the loss; then for each different ratio of expansion between these extremes there must be some degree of compression which is the most economical.

A general expression for the useful work under any condition could be derived, but it would be complicated, and

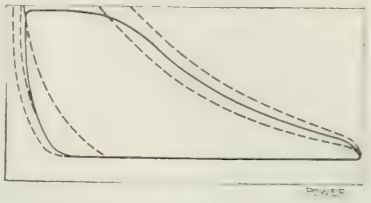


FIG. 4

it is much easier to find the proper amount of compression by drawing a theoretical diagram with various compression curves, and the corresponding expansion curves, keeping the volume V_1 up to the point of cutoff the same. Then with a planimeter measure the areas representing the useful work for the different degrees of compression, and that for which this area is the greatest will be the most economical compression to use.

If on an actual indicator diagram, Fig. 4, a series of compression curves and corresponding expansion lines are drawn, keeping the area constant, it will be found that the admission line V_1 will be shorter for one pair of curves than for any other pair. This then will be the proper compression for this particular load.

If an engine is overloaded and cutting off late, then low compression is to be desired. Cutting down the compression and leaving the ratio of expansion the same would increase the efficiency, but in the actual engine it would also give a better ratio of expansion. On the other hand, an engine may be underloaded, and with low compression will be exhausting below the back pressure, and there will

be a negative loop in the diagram which perhaps can be removed by increasing the compression. These effects of compression are entirely separate from its effect in filling the clearance volume.

Also, with the single-valve automatic engine, in which the compression changes with the cutoff, this change of compression aids the regulation as the governor does not have to move so far to change the area of the diagram a given amount as when the compression is constant. This is one reason why the single-valve engines usually regulate better than four-valve engines.

Commercial Efficiency

In what has been said so far, the thermodynamic efficiency of the engine has been considered and the results were derived by assuming theoretically perfect conditions. In actual practice it is commercial efficiency which is desired, and the conditions are not ideal. Due to heat absorption by the metal of the cylinder and piston and to mechanical losses,

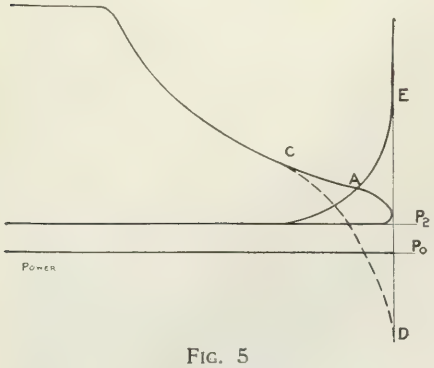


Fig. 5

not more than about 85 per cent. of the energy that is required to compress the steam can be recovered.

The claim is made that compression, by warming the cylinder, prevents to a certain extent the condensation of the live steam entering the cylinder, which is probably true, but this operation is effected by a double transformation: first, of the energy of the steam into the mechanical energy of the flywheel, and then from the flywheel to heat energy again, both transformations being made at low efficiencies. It would seem therefore that it would be better to take the heat directly from the boiler for this purpose.

So far as economy is concerned, it would appear that the greatest benefit of compression is due to its ability to regulate the ratio of expansion to a limited degree with single-valve engines. Wherever an engine is to operate under a steady load, some form of four-valve engine, with low compression, properly proportioned cylinders and tight valves and piston will give the best results, especially if it is to be run noncondensing If, however, the load is variable, with the engine running frequently much underloaded, it is probable that the single-valve automatic engine will do better than the four-valve type.

With unbalanced or only partially balanced valves when used at high pressures, considerable power is required to move them. This is noticeable in some of the late designs of four-valve engines, with Corliss-type valves and a valve-gear which gives a very marked nonharmonic motion to the valves. This effect is so great that some engines, when they are new, cannot be run with the throttle wide open with no load as there would be danger of pulling the valve-gear to pieces. It can be remedied to a certain extent by increasing the compression, for then at the time the valves open there is less unbalanced pressure on them and they move easier.

This same effect is noticed with Corliss engines working under comparatively high pressures. They must be given more compression than at low pressures or the amount of oil required to keep the valves running smoothly will be greater.

Most engine builders also desire a certain amount of compression as an aid to quiet running, particularly in engines running at comparatively high speeds. In the case of an engine running at

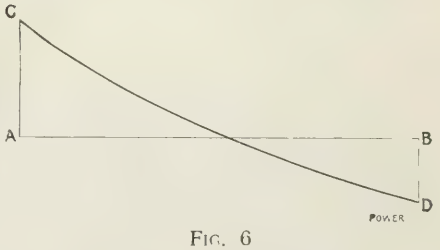


Fig. 6

high speed and with no compression, the stress at the end of the stroke must be taken on the bearings and also by the steam after admission occurs, but admission should not begin until very nearly the end of the stroke, and the reversal of pressures would be very sudden.

Effective Pressures

The ordinary indicator diagram is a combination of curves drawn during two strokes. The admission and expansion lines are drawn during one stroke and the exhaust and compression lines during the return stroke, so that the effective pressure is not measured at any instant by the vertical distance between the expansion and admission lines above and the exhaust and compression lines below, as is usually assumed. If the expansion curve for one end of the cylinder is combined with the exhaust line from the other end of the cylinder during the same stroke, then the vertical distance between these two lines would represent the true effective pressure. This makes no difference in the result if it is desired to find the power developed, but if the pressures on the pins and bearings are wanted, there is a difference. If the admission and expansion lines of one diagram are taken and combined with the back-pressure line for the other end of the cylinder, a diagram is had similar to Fig. 5.

At the point *A*, where the expansion and compression curves cross, the effective pressure on the piston is reversed; and if the cutoff is early and the compression high, the pressure will be reversed long before the end of the stroke. By laying off the effective pressure on the back-pressure line P_2 as a base, then, where compression starts, the line representing the effective pressure will leave the expansion line and extend downward, as shown by dotted line *CD*, the distance P_2D being equal to P_2E.

This gives an idea as to how the actual effective pressure on a piston varies throughout the stroke, but it does not show anything regarding the pressures on the pins and bearings. This is greatly modified by the inertia of the reciprocating parts, which may be very large. At the beginning of the stroke a certain amount of the effective pressure in the piston is used to give motion to the reciprocating parts, and while the piston

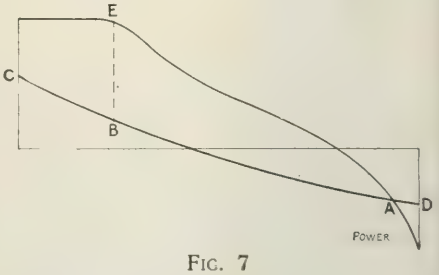


Fig. 7

is again coming to rest at the end of the stroke all this energy is given out. If the motion of the piston were the same when leaving each dead center, the inertia would affect both ends alike, but when the piston leaves the head end of the cylinder it must move faster than when leaving the crank end. This is due to the effect of the angularity of the connecting-rod. Let *AB*, in Fig. 6, represent the stroke; then the effect of inertia can be laid off vertically; if this is reduced to pressure per square inch of the piston to the same scale as that of the indicator diagram, the two diagrams showing the pressures on the crankpin due to the effects of the combined steam pressure and inertia can be combined. In Fig. 6, *AC* represents the effect of inertia at the head end and *BD* at the crank end of the stroke.

If an indicator diagram is drawn on this figure, a diagram similar to Fig. 7 will be obtained. In this diagram the vertical distances between the two lines representing the effective steam pressure and the effects of inertia would give the pressure on the crankpin at the corresponding part of the stroke. From this diagram it will be seen that the effect of

inertia is to reduce the pressure on the crankpin at the beginning of the stroke and increase it at the end.

If there were no compression, the pressure on the crank would become very large close to the end of the stroke and be suddenly reversed by the inrush of steam when the admission occurred, not only creating undue strain but causing the engine to pound if the bearings were not perfectly adjusted. The same result may be expected if the compression is very high and the load is light as the pressure is then reversed at some distance from the end of the stroke when the piston is still traveling fast.

While it can be shown that the change of pressure is comparatively gradual, the effect is the same as though the piston were traveling slowly and the pressure were suddenly reversed. This condition is seen in many single-valve automatic-cutoff engines; they will pound with light loads but will run smoothly when they have enough load to cut down the compression to the proper amount.

From the foregoing it may be inferred that the three principal advantages to be derived from the proper amount of compression are: keeping the ratio of expansion within proper limits when the load is very changeable and varies within wide limits; an aid in easily moving unbalanced valves with high pressures; and modifying the pressure on the crankpin, due to the steam pressure and the inertia of the reciprocating parts making them more uniform and preventing heavy strains and sudden reversals of pressure at the end of the stroke.

An added inference is that the benefit derived from filling the clearance space with compressed steam, so far as economy is concerned, is very small, if any, even though, as shown under theoretical conditions, it may fully compensate for the loss due to clearance.

Fan Engine Room Signals

Although the ventilating fans are of vital importance for the safe operation of a mine, the engines which drive the fans are, in many instances, unattended both night and day. Occasionally, the nearest hoisting engineer drops in to look at the engine which happens to be in his care, but at other times the engine takes care of itself.

In other instances, a fan engineer is on watch during the day-shift, but the engine is left in the care of a hoist engineer at night. There is a chance under such conditions that something may go wrong immediately following his inspection of the fan engine, and several devices have been rigged up by these engineers which will enable them to know that the engine is either running up to speed, slowed down or has stopped.

At one mine the engineer made an air pump out of a brass cylinder and fitted it with a piston and piston rod. The piston was connected to a link which was attached to a rocker-arm. This arm was pivoted at the bottom end and was made with two side arms. A carrier arm passed between them and engaged with an iron pin which ran through both arms. The carrier arm was bolted to the bottom of the engine eccentric, the motion of which actuated the air pump. The device is shown in Fig. 1.

When not in use a block is placed between the pin and the carrier arm, the block forming a riding surface for the arm.

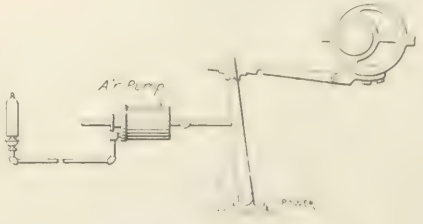


FIG. 1. AIR PUMP

When the day engineer goes off watch the carrier arm is engaged to the pin in the rocker-arm and the air pump is thus set in operation. The compressed air is carried through a small iron pipe to the hoisting-engine room where a small whistle is attached to the end of the air pipe. Thus, at every revolution of the fan engine the whistle is blown. In this case the shrill tone of the whistle had been muffled by wrapping cloth around it, so that the escaping air alone is heard by the engineer on watch, the secession of which would be instantly noted.

A second device is shown in Fig. 2. It consists of a recording gage attached to a pipe leading to the fan chamber.

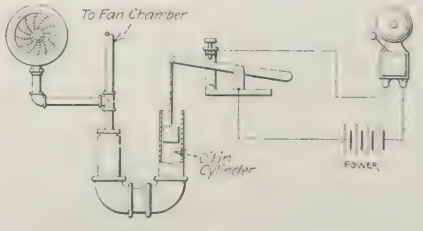


FIG. 2. DRAFT-GAGE ALARM

At the bottom of the tee connection a piece of 2-in. pipe is attached and a U-arrangement made, as shown. In the outer upright pipe a float is placed, which is connected to a contact lever by a small iron rod. The large pipe is then partly filled with oil. The contacts are connected to an electric bell terminal and batteries by wires. The outer contact block is fitted with an adjusting contact screw.

With the fan running at normal speed the pull in inches of water will lower the oil and float in the outer end of the U-connection. The adjusting screw is set so that no contact between it and the contact arm occurs while the fan is running at normal speed.

If for any reason, such as low steam pressure, the fan engine should slow down, the pull on the oil decreases and the oil rises in the outer leg. The float rises with it and moves the arm up against the contact screw, closing the circuit and ringing the bell.

A third alarm is made with a steam

FIG. 3. STEAM-GAGE ALARM

gage connected to the engine steam pipe in the usual manner. Two spring contact pieces fit over the face of the gage, as shown in Fig. 3. If the minimum pressure necessary to operate the fan engine is 80 lb. gage per square inch, the contacts are so arranged that the gage pointer will make contact between them at that pressure. This rings a bell in the hoist-engine room and the engineer looks into the cause of the drop in steam pressure.

Hydro-electric Development near Pittsburgh

Plans are under way for constructing a hydro-electric development near Pittsburgh by the American Water Works & Guarantee Co. and its allied interests. Acquisition of water-power sites on the Cheat River, within 50 miles of Pittsburgh, has been going on for some time and it is said that with the property and rights now held, more than 100,000 hp. can be developed when the installation is complete.

The company recently offered for sale $10,000,000 in 6 per cent. cumulative participating preferred stock and it is understood that part of the proceeds will be used for the hydro-electric enterprise.

Most of the power generated can be supplied to the railway, light and power systems of the West Penn Traction Co. alone, in addition to the local demand in the West Virginia and Pennsylvania markets. A majority of the West Penn company stock is owned by the American company.

The traction company has 209 miles of track and supplies light and power in 110 cities and towns in the Pittsburgh district within a radius of 30 to 60 miles of the city. The main power plant, in Connellsville, has a capacity of 42,000 hp., which will soon be increased to 55,000 hp. The demand from the West Penn system alone will probably exceed the capacity of the first of the hydro-electric company's development.

Large Steam Turbine Development

A discussion of the present development of large steam-turbine construction and of the field of the steam turbine both in this country and in Europe.

Steam turbines are now being used for driving alternating-current generators, turbo-compressors, turbo-blowers, pumps and marine propellers, and through gearing, to furnish power to direct-current generators, rolling mills and the propeller shafts of steamships. Reciprocating engines were formerly used for such purposes, but recently this class of engine has seldom been installed except for rolling-mill work, non-condensing service as on heating systems, rope and belt drives, hoists, and in certain combinations with low-pressure turbines in marine work. The high economy of the piston-pumping engines and also of some types of air compressors, has continued their popularity in spite of the increasing competition of steam-turbine units. The steam turbine has found favor principally on account of its low cost of installation, its small floor-space requirements, its continued good steam economy over a period of years and its small operating and repair charges.

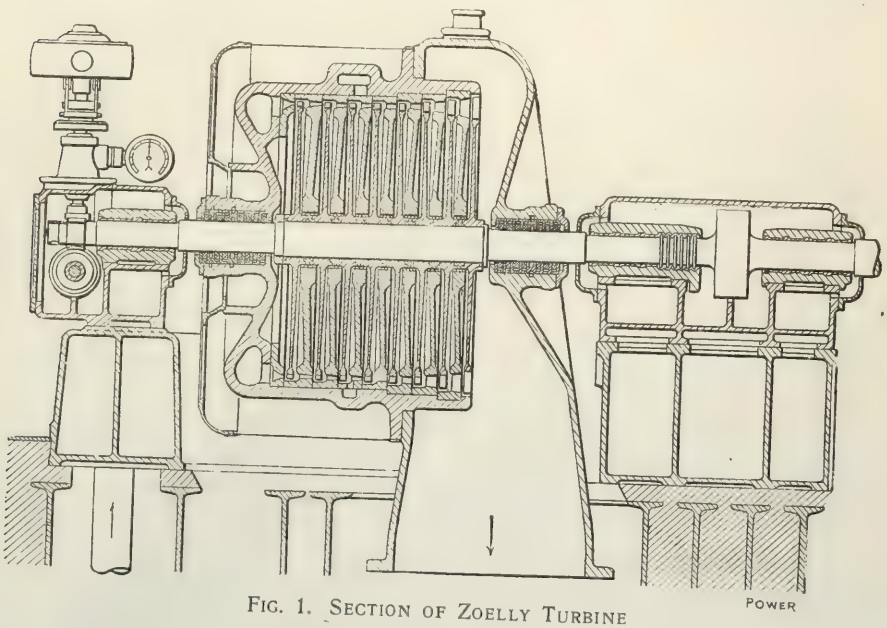


FIG. 1. SECTION OF ZOELLY TURBINE

The increased use of steam turbines in sizes up to 1000 hp. seems to have received at least a temporay check in Europe by the introduction of the new Stumpf direct-flow engine.

TYPES OF STEAM TURBINES

For the purpose of this paper, large commercial steam turbines will be divided into two classes: (*a*) fundamental types, and (*b*) modified or combined types.

The fundamental types of turbines are as follows:

a The Parsons type, which works on the so called "reaction" principle. In this type the heat energy of the steam is changed into kinetic energy, both in the stationary guide blades and in the moving blades. In other words, both sets of blades act as orifices expanding the steam through a small pressue drop. As nozzles and orifices usually have very high efficiencies, this turbine should, theoretically, prove the most economical of all types. The Parsons turbine consists of a drum, or a number of drums, carrying the blade rows which alternate with rows in the casing. The drums carry balance pistons to equalize the end thrust.

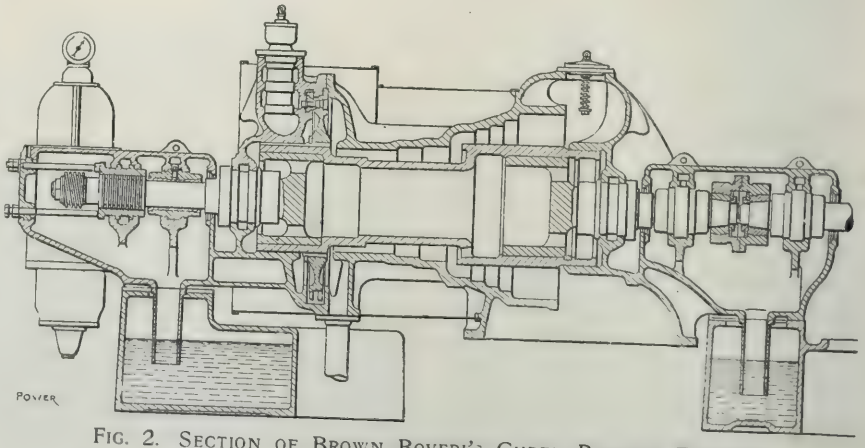


FIG. 2. SECTION OF BROWN BOVERI'S CURTIS-PARSONS TURBINE

b. The Curtis type, which works on the impulse principle with high steam velocities and few stages. Each stage, however, has two or more rows of revolving blades known as "velocity rows," with intermediate rows of guide blades. The steam velocity at the beginning of each stage is high. The revolving blades are carried on disks separated by diaphragms, which extend to the shaft and carry the orifices between stages. Curtis turbines are now usually built with horizontal shafts. In American practice some sizes over 7000 kw. still have vertical shafts.

c. The Rateau turbine, which consists of a number of simple impulse wheels in series on the same shaft and separated from one another by diaphragms carrying nozzles. It operates with lower steam velocities than the Curtis and consequently has many more stages. Each revolving element caries only one row of blades.

The type of turbine known as the Zoelly belongs to the same classification as the Rateau, from which it differs only in the use of higher steam velocities, in the number of stages and in certain constructional details. Fig. 1 shows a section of a Zoelly turbine. Though this type has been widely adopted abroad, it has not yet been placed on the American market.

Each of these fundamental types is based on sound theoretical principles. In the process of manufacture and in operation, certain features have not proved entirely satisfactory, hence far-reaching modifications have been made in the design of some types of turbines. Some manufacturers have combined the characteristics of two or more types to overcome the inherent limitations of each fundamental type. A discussion of the unsatisfactory operating conditions of each type will show the reasons which led up to the changes in recent turbines and will also aid in distinguishing the novel features of new designs.

The first row of spindle blades in a

*Abstract of paper presented by Prof. A. C. Christie at the spring meeting of the American Society of Mechanical Engineers.

standard Parsons turbine is placed on a drum of small diameter to make the blades as long as possible and to minimize the proportional leakage losses past the ends of the blades. A large number of rows is provided to keep the steam velocities low, as the blade velocities must be low with the small drum. The drop in pressure at each row is small and hence the leakage is correspondingly reduced. This construction results in a

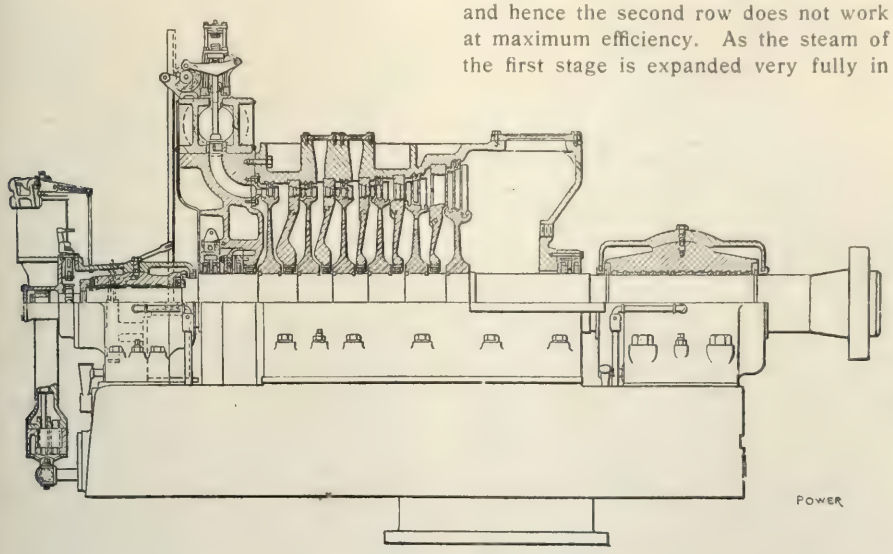


FIG. 3. HORIZONTAL SIX-STAGE CURTIS STEAM TURBINE

turbine with a long spindle and with great distance between bearings. High-pressure steam, frequently at high temperature, is admitted to the casing. Distortion of casing and spindle is thus easily conceivable in such construction, and to allow for this contingency the clearance on the ends of all blades is usually increased. This distortion may be due either to unequal heating or to the growth of the cast-iron casings. The fluid friction losses are large in this high-pressure section, for a large number of rows of blades must be revolved in steam of high density. The leakage losses and fluid friction losses in the high-pressure section, with the troubles due to distortion in the long shaft, have forced designers to modify this part of the turbine, either by new blading arangements, by dividing the total expansion in large sizes between two cylinders, or by introducing impulse blading.

The low-pressure sections of Parsons turbines have always shown high efficiencies. As low steam velocities are characteristic of this type, there is no cutting away of blade material, even with very wet steam, unless the feed water has injurious properties. This low-pressure section has therefore been altered only in details.

The presence of a large low-pressure balance piston close to the steam inlet has frequently caused serious distortions. Many builders now place this balance piston in the exhaust end, a construction due to Fullager.

The Curtis turbine utilizes high steam velocities in all stages. As steam becomes wet through expansion in the low-pressure stages, it frequently cuts the blade material considerably, although, contrary to first expectations, cutting in the high-pressure blades due to the high initial velocities is rare. The first row of velocity blades in a stage usually does the greater part of the work, and hence the second row does not work at maximum efficiency. As the steam of the first stage is expanded very fully in the nozzles, there is no high pressure or superheat in the turbine casing, or at the glands. The vertical type of unit is sometimes subject to electrical unbalancing and to other troubles peculiar to this

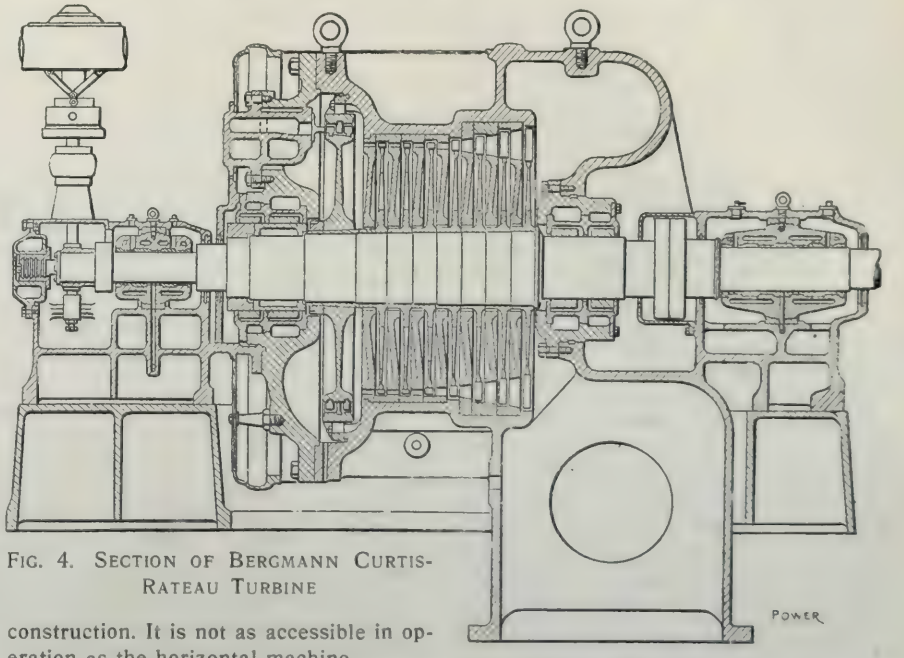


FIG. 4. SECTION OF BERGMANN CURTIS-RATEAU TURBINE

construction. It is not as accessible in operation as the horizontal machine.

Recent designs have provided for horizontal units and for the replacement of the low-pressure section by sections of other types.

The Rateau turbine has high pressures on the gland at one end. A large number of disks revolve in dense steam at the high-pressure end. It has lower steam velocities throughout than the Curtis, and consequently has no blade cutting in the low-pressure section. The clearances around the blades are large, but the shaft clearances of the diaphragms must be kept small. Some builders of this and the Curtis type have brought out new designs which employ the high-pressure section of the Curtis with the low-pressure section of the Rateau. These represent a compromise between efficiency and manufacturing costs.

Under the classification of modified or combined types, are turbines with modified Parsons, Curtis and Rateau construction, and turbines with combinations of Curtis-Parsons, Rateau-Parsons, Curtis-Rateau and Curtis-Zoelly construction, with a few special combinations. Some typical turbines of these classes are described in the following paragraphs:

The Allis-Chalmers Co. manufactures a modified Parsons turbine. High peripheral speeds are employed with a decreased number of rows. Part of the theoretical efficiency in the high-pressure end has been sacrificed by the use of fewer blade rows. Also a smaller proportion of work is done in this section than is usual in Parsons turbines. The blades all have a channel-shaped shroud. European experimenters have pointed out that better efficiencies are obtained with shrouded than with plain blades, as the so called spilling-over at the ends is prevented. Wing blades are fitted in the last low-pressure rows to care for the large volumes of low-pressure steam. The spindle is much shorter and stiffer than the standard Parsons, and hence smaller clearances can be

provided. The Fullager low-pressure balance piston is also used. The outstanding features of this design are, reaction principle with drum construction, few rows of blades with high steam velocities, short spindle construction, and employment of wing blades for high vacuum. The steam consumptions obtained show improved efficiency over standard construction.

A number of Parsons turbines have been built in which the total expansion is divided between two cylinders. One of the best known units of this type is

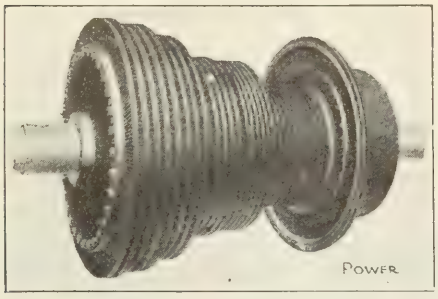


FIG. 5. SPINDLE OF CURTIS DRUM IMPULSE TURBINE

installed at the Dunstan Power Station, England. This was built by Richardsons, Westgarth & Co. on Brown-Boveri designs. This unit, however, employs a modified double-flow Parsons construction in the low-pressure cylinder.

C. A. Parsons & Co. have used cast-steel cylinders for the high-pressure part of their turbines to overcome the troubles due to deformation and growth of metal under high pressures and superheat experienced with cast-iron casings.

The Westinghouse Machine Co. has developed a double-flow machine which employs a Curtis high-pressure stage with Parsons intermediate and double-flow low-pressure sections. Its only balance piston is next to the Curtis ring. These turbines run at high speeds, have short shafts and small clearances. The steam is expanded in the nozzles, and hence there are no high pressures or temperatures in the casing. By providing two low-pressure sections, high vacuum can be economically utilized.

European builders of Parsons turbines, among whom are Brown, Boveri & Co., C. A. Parsons & Co., Franco Tosi, Sulzers, Willans & Robinson and Erste Brünner, have replaced the high-pressure sections of their Parsons turbines by a Curtis wheel with two or more velocity rows, but have retained the single-flow Parsons drum construction for the remaining portions. One of these units is shown in Fig. 2. The temperatures and pressures in the casing are low as the steam is expanded in the nozzles. The distance between bearings is decreased, the shaft is stiffer and clearances are smaller than in the standard Parsons turbine. The Fullager balance piston is used in the turbines of several of these builders. Turbines thus built have shown some exceptionally good efficiencies.

Compared with the Westinghouse machine, the leakage at the end of the low-pressure blading is less than in the double-flow section. On the other hand, the Westinghouse machine has smaller balance-piston losses and can also utilize the highest vacuum at better efficiency.

Melms & Pfenniger employ a drum impulse section of about five stages instead of a Curtis ring on some of their large turbines. They claim improved efficiency from this construction, though high-pressure and high-temperature steam are introduced into the casing.

The General Electric Co. now manufactures a horizontal Curtis turbine in all sizes up to 7000 kw., a section of which is shown in Fig. 3. This embodies all the essential features of the Curtis design. Compared with the vertical type, it provides easier access to all working parts, such as governor, bearings, valves, etc., and allows a better survey of the unit. The machine can also be dismantled and its internal parts examined with less trouble. The oiling problem is very simple compared with the vertical units, as there is no step bearing. It is possible that only horizontal units of all sizes will be built in the near future. The Algemeine Elektricitäts Gesellschaft also builds similar turbines up to 1000 kw. capacity.

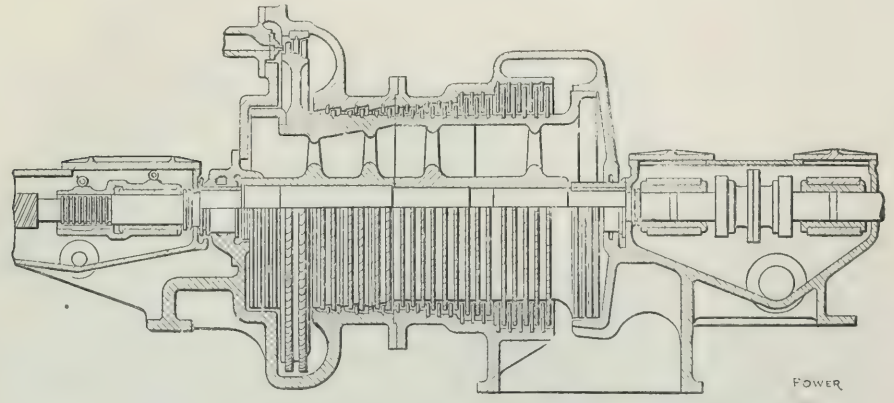


FIG. 6. SECTION OF CURTIS DRUM IMPULSE PARSONS TURBINE

Many manufacturers in Europe are now building a turbine of the type shown in Fig. 4. This consists of a Curtis high-pressure with Rateau or Zoelly low-pressure sections, the number of stages depending on the size of the machine, the speed and the steam conditions. The pressure and temperature in the casing are low, as a consequence of the expansion of the steam in the nozzles. The steam velocity is moderate in the low-pressure section. This machine is longer than the simple Curtis but shorter than the Rateau. Its builders claim improved efficiencies over either of the fundamental types.

Belliss & Morcom, of Birmingham, England, have introduced a turbine, the spindle of which is shown in Fig. 5. It consists of a high-pressure Curtis stage with a low-pressure drum construction like Parsons, but fitted with impulse blading on the spindle and blades forming expansion parallel wall nozzles in the casing. All other impulse turbines employ diaphragms extending to the shaft. A Belliss turbine has more rows than a corresponding Rateau turbine, and hence less pressure drop through each set of guide blades. The net leakage loss may thus be lower than the Rateau, even though the leakage area is greater. This turbine is essentially a Curtis-Parsons construction, with few rows of blades and with the expansion taking place in the stationary blades only.

A turbine, shown in Fig. 6, has recently been patented in England, which has some features of unusual interest. The high-pressure stage is Curtis, the intermediate section is of a drum-impulse type with three stages, while the low-pressure portion is Parsons, which may be also double-flow. This combines features of all three fundamental types, and it will be interesting to see what results are obtained in practice by such an arrangement.

(The paper continues with a discussion of turbine details, such as blading, nozzles, bearings and lubrication, spindle construction, packing glands, thrust bearings, governing devices, casings, bedplate and turbo-generators, and contains a voluminous table in which are given the characteristics of the turbines of different builders as well as the table of test results reproduced in POWER, June 11, page 832. It touches also upon low- and mixed-pressure turbines, turbo-compressors, turbine-driven pumps, geared turbines, and marine turbines.)

TREND OF TURBINE DEVELOPMENT

The cost of manufacture is very important in determining the future development of the steam turbine. Types such as the original Parsons and the Rateau, while inherently very efficient, cost too much to build to compete with the newer combined types.

The writer offers as his opinion that the combined types, such as the Curtis-Parsons, the Curtis-Rateau and also the

Curtis-Rateau-Parsons, will soon supersede the simple types. Probably the Curtis turbine will eventually be built only in horizontal units and will gradually be modified to a Rateau or even a drum-impulse construction in the low-pressure sections. The freedom from close adjustment in impulse turbines and the recent improvements in blading materials will greatly increase the use of this type, although Curtis-Parsons turbines are said to be cheaper to manufacture. In actual operation, it is an open question among engineers whether the reaction turbine has a higher commercial efficiency than the impulse type, and hence buyers usually consider first cost and personal preference only.

Turbines will probably be made shorter with very stiff shafts. With this construction many of the earlier blading troubles will disappear. But the peripheral speeds will also be increased and this will involve the development of suitable blading material and ways of holding blades that will satisfy these new requirements. Recent results seem to indicate that improved efficiency may be looked for with increased blade speeds.

Several impulse turbines have been built recently in Europe where the expansion was not complete in the nozzle, so that part of it took place in the first moving blades. Some large Curtis turbines recently installed in America are said to have Parsons blading in the last stage. These developments would indicate a movement to introduce reaction principles in impulse turbines, and further illustrate the tendency to merge types.

The hope of further improvement in efficiency lies in extensive study, particularly of the action of the steam during its passage through the moving and stationary blades, of the effect of form of blades, passages and casings, and of various forms of baffles and balance pistons to prevent leakage. Such research work has not been carried out up to the present time by most manufacturers, largely on account of the extreme care and heavy expense involved in such tests. The present state of development has been largely one of cut-and-try. The increasing competition of the gas engine and the possible development of a satisfactory gas turbine will force manufacturers to develop their turbines to the greatest degree of economy.

With regard to detail, simplicity will be the leading consideration. With the introduction of Curtis high-pressure stages, nozzle governing will undoubtedly be used to an increasing extent, though the results obtained by Westinghouse, Zoelly and Bergmann with simple throttling governors, raise a question as to whether the additional complication of nozzle governing will pay. Oil relays will probably replace all other systems of governing on account of their simplicity and reliability. The simple and efficient centrifugal oil-pump governor of Sulzer appears to be an improvement of considerable moment, and will probably be extensively used.

With the development of suitable gearing for steam turbines, their field of application has been greatly increased and turbines will shortly be used where engineers today would consider them utterly unfit. Low-pressure turbines will continue to be installed in plants where reciprocating engines are still in operation and also where large quantities of waste heat are available. The low- and mixed-pressure types of turbines will find extended use in connection with heating systems, evaporators, etc.

The development of the past ten years has been truly marvellous. No great gain in thermal efficiency seems possible, so that future improvements will be largely along the line of detail construction and modification.

Explosion of Air Receiver Due to Oil

By Frank Richards

At a temperature of about 500 deg. the oil which coated the inside of the head became ignited and heated the latter to a point at which it dished outward, the sudden shock shearing off the rivets.

A small air receiver connected with a portable gasoline-engine-driven air compressor employed on street gas-pipe work exploded on June 24, near the entrance to Greenwood Cemetery, Brooklyn. The occurrence exhibited a number of interesting features illustrative of the conditions under which such accidents may occur.

There seems to have been no sudden and enormous increase of pressure, such as would be caused by the ignition of an explosive mixture composed of volatilized constituents of the oil combined in suitable proportion with the compressed air; the normal working pressure which the receiver usually carried was probably not much if at all exceeded; nevertheless it seems to have been ultimately a simple pressure explosion, but accompanied by abnormal conditions which provided the opportunity.

The receiver was suspended horizontally under the frame which carries the gasoline engine and the compressor. It was 6 ft. long and 2 ft. in diameter, with dished heads; the convex side of the head was outward at one end and the concave side outward at the other end, which is the too common practice, just for convenience in riveting in the last head, and for no other good reason. The air entered the receiver from the compressor about 2 ft. from one end and half way up, and was taken out at the other side just opposite the inlet (see Fig. 1). A pop safety valve on the pipe from the compressor was set at 110 lb., and the usual working pressure was about 100 lb.

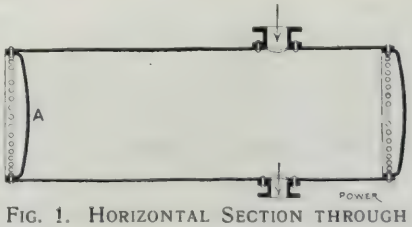


FIG. 1. HORIZONTAL SECTION THROUGH RECEIVER

As disclosed after the explosion, the entire interior surface of the receiver was covered apparently with all the oil it could carry. If the operator had been told to use all the oil he could, instead of as little as possible, the surface could not have been more oily. From time to time all the oil and water which might have collected at the bottom of the receiver should have been drained off; this may or may not have been done, but evidently it would not have made the surfaces clean, nor have freed them from the gummed or thickened oil with which they were coated.

The inner surface, *A*, Fig. 1, of the head which blew out, however, was dry and absolutely clean and its color and condition showed that it had been red hot. Evidently a local fire had raged, perhaps only for a minute or so, on the (then oil-coated) surface, burning the oil off and suddenly heating the plate; the fire did not have time to spread to the adjacent surface of the shell, when the explosion occurred.

In the explosion several curious things happened which it is worth while to note carefully. The head became so hot that, as it was under about all the air pressure it could stand, the dishing of the head was suddenly reversed, and the originally convex surface became concave, as shown in Fig. 2, this being a view of the inner surface. That this is the inner surface of the head is corroborated by the lip at the edge of the sheet. The ridge which is dented across the face was evidently

caused by something against which the head struck as it flew off.

It is easy to imagine that this reversal of the dishing of the head took place with a considerable snap, the shock and momentum, in addition to the aggregated air pressure of 45,000 lb., being sufficient to cause the head to let go. When it did, not a single rivet was sheared. It will be noticed from Fig. 2 that when the reversal of the dishing occurred, the rivet-carrying lip of the head was bent outward all the way around, so that the opposite sides, instead of being parallel, stood off 30 to 45 deg. from the normal position, and the end of the sheet to

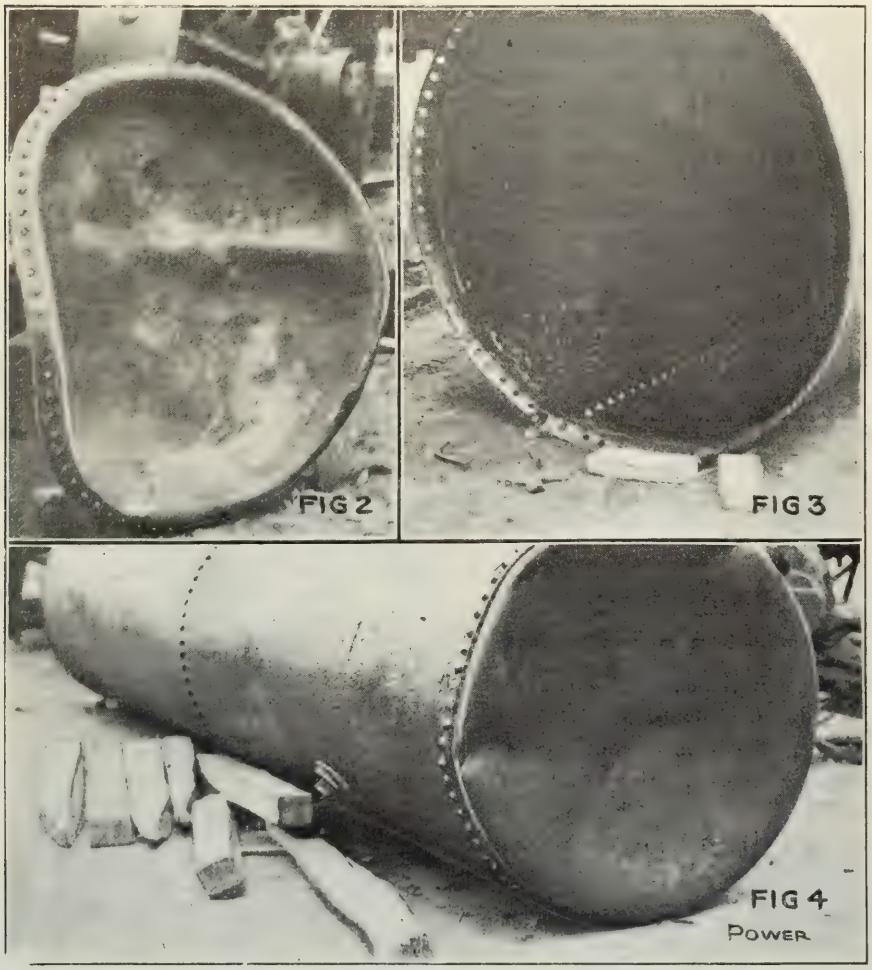


FIG. 2. INSIDE FACE OF HEAD AFTER EXPLOSION
FIG. 3. SHOWING FLARING END OF SHELL WITH RIVETS PULLED OUT
FIG. 4. OTHER END OF RECEIVER

which they were attached (see Fig. 3) bent outward in the same way. There were eighty ⅜-in. rivets, and when the lip was bent out their heads were pulled off, dropping outside the shell, and then the rivets pulled out of the holes. The fracture of the rivets was square across under the heads and flush with the outside of the shell. Of the 80 rivets, three retained their heads and those tore notches out of the shell.

When the head flew off, the body of the receiver was driven in the opposite direction, but could go only a few inches when it struck the axle of the machine, which dented the head as shown in Fig. 4. Other than this the shell seems to have received no damage.

Undoubtedly receivers have often taken fire internally, as indicated in this case, but having a sufficient factor of safety, in spite of the heating of the metal, they have been able to withstand the stress and no accident has resulted.

The interesting question is as to the conditions causing, or accompanying, these ignitions of the oily surfaces, and usually more or less mystery is made of this phenomenon when it occurs. Compare the oil tempering of steel springs. The spring is first heated in the fire to a bright red and then plunged into oil and cooled. This leaves it hard and brittle, and it is then drawn to the proper temper by "blazing off." To do this, the spring, while dripping with oil, is held over the fire, which must be without flame, and is slowly and carefully heated until the oil on the spring bursts into flame. The oil is not ignited by the fire, the flame merely appearing when the spring reaches the right temperature. The oil will keep burning until consumed, or the spring, while blazing, may be dipped into the oil and the flame will go out, but if quickly withdrawn the flame will appear again. At a certain temperature oil will ignite spontaneously, and this temperature is so fixed, differing, however, for different oils, that it has been taken for generations as an index of the heat required for the tempering of springs, swords, etc.

Precisely the same thing that happens in the oil-tempering of springs will happen to the oil-coated surfaces of air reservoirs, pipes, etc., when they get hot enough. The oil will take fire of itself without any spark, flame or other extraneous source of ignition. The oil-covered steel spring, when the oil upon it takes fire in the open air, is below a visible red heat, and that begins at about 700 deg. F. In the air receiver under consideration, the temperature due to compression may have been as high as 500 deg. With air at 60 deg. when compression begins, the theoretical terminal temperature, after adiabatic compression to 105 lb., is 496 deg. This compressor was running in bright sunshine on a hot day, and the intake air passed close to the already heated air receiver before entering the cylinder, so that its temperature at the beginning of compression was presumably above 100 deg., and its terminal temperature above 500 deg. There being nothing at the end of the receiver to cause any active movement of the air in contact with the oily surface, may have offered special opportunity for the atoms of carbon and oxygen to unite. Furthermore, since the air was compressed to 8 atmospheres the oil was exposed to 8 times the amount of oxygen that it would have at atmospheric pressure. This may partly account for the spontaneous ignition of the oil at a lower temperature and its burning more fiercely after ignition.

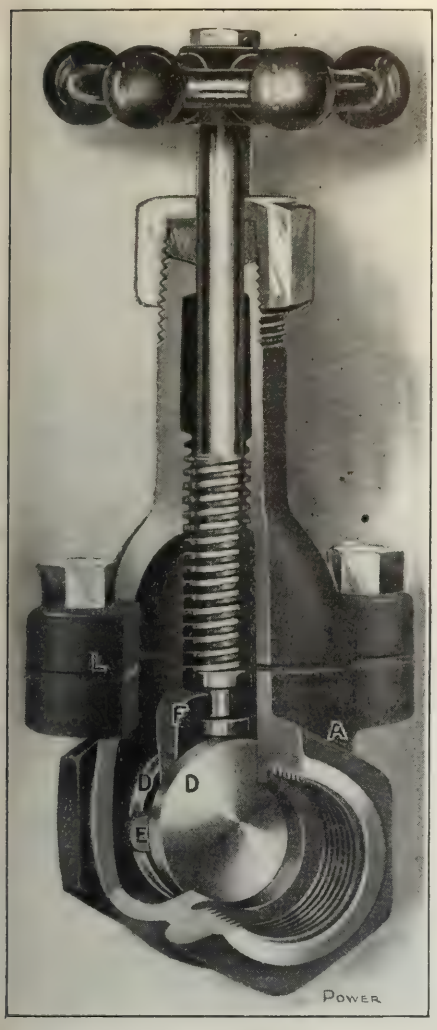
An account of an ignition of this character recently appeared in *Mines and Minerals*, and was reprinted in *Compressed Air Magazine*, June, 1912. No damaging explosion occurred, however. It was in connection with a four-stage compressor employed for charging locomotives. The ignition took place after the fourth stage of the compression, when the air was at a pressure of 65 atmospheres, and a recording thermometer, which was attached and in operation, showed that the ignition took place at 270 deg., which seems to corroborate the assumption that the higher the pressure the lower will be the temperature at which spontaneous ignition will occur.

This discussion distinctly avoids any consideration of the more disastrous class of compressed-air explosions caused by the ignition of an explosive mixture of air with some of the volatilized constituents of the oil, as there seems to have been nothing of that in this case.

"Cino" Iron Body Gate Valve

The new "Cino" iron-body gate valve is shown in section herewith. The bronze disks *D* have a ball-and-socket bearing at their center, suspended from and held in position by a malleable iron, wedge-shaped holder with lugs *E*, on either side which travel in corresponding grooves in the body.

In closing the valve, the disks automatically rock into position and adjust themselves to the taper of the seats. The



CINO GATE VALVE

spreading action of the ball-and-socket bearing is supplemented by the wedging action of the disk holder *F*, which affords any further tightening necessary after the disks have seated.

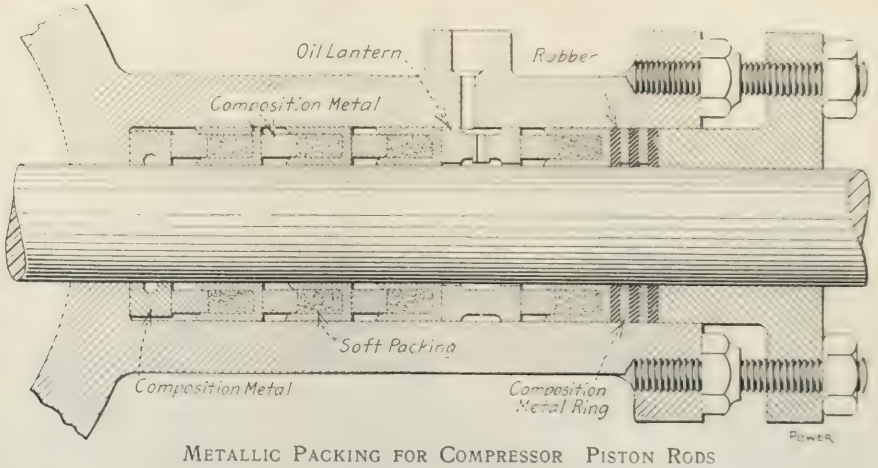
The flexibility of this double-disk mechanism compensates for any distortion or derangement of the angle of the seats, even though one disk may be temporarily obstructed by particles of dirt or scale. The valve takes pressure from either side and has a full and unobstructed opening.

Should anything accumulate in the pocket formed by the seats, it is crowded upward between the two disks when closed, and on the next opening is carried up and washed away. The two disks are coupled to the disk holder so that they are free to revolve, opening or closing and, actuated by the pressure, make almost a complete revolution, which automatically grinds in the disks and cleans the seating surfaces of dirt or scale.

It is not always necessary to replace worn disks with new ones, as they can be reground. Both disks are interchangeable. The valve is manufactured by the D. T. Williams Valve Co., Cincinnati, Ohio.

Goetze Compound Metallic Packing

This packing is constructed for compressor work and is made of composition metallic and rubber and soft packing rings. The rings are arranged as shown in the illustration. The composition metal rings are made U-shape, in the hollow



METALLIC PACKING FOR COMPRESSOR PISTON RODS

of which soft packing is placed. The packing is guaranteed for two years and is manufactured by the Goetze Gasket & Packing Co., New Brunswick, N. J.

Albany Sectional Dumping Grate

A new dumping grate recently gotten out by George H. Thacher & Co., Albany, N. Y., is illustrated herewith.

The grate frame, designed to come below the fire surface, is made with lugs which fit in grooves in the rocker bar where they are fastened in place. Each frame contains several sections of grates.

The side frame is made with lugs on the inner side against which projections on the rocker bars rest when the grates are locked in a level position. The grates assume an angle of 65 deg. when dumping, but can be adjusted to a dumping angle of 90 deg.

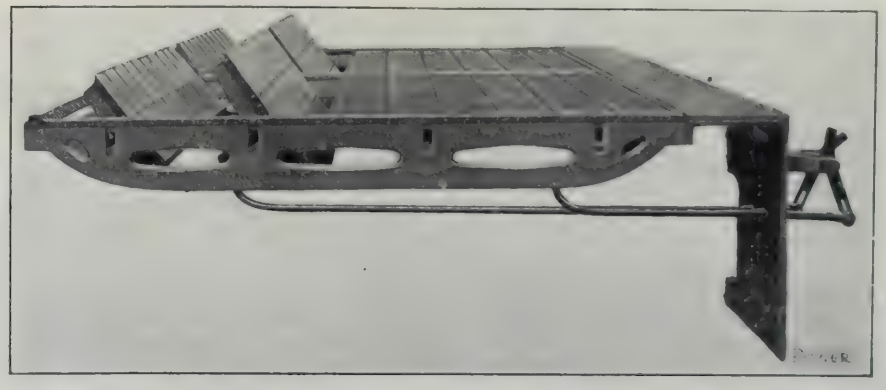
All parts are rugged in construction and the sectional fire surface is designed to prevent warping of the grate bars or side frames. Each grate overlaps the next one at the rear and the front grate overlaps the rear sectional end piece.

The Baume Hydrometer

BY LOUIS GROSSBAUM

A hydrometer is an instrument for determining the specific gravity of liquids, and is based on the principle that the weight of a floating object must always equal the weight of the liquid it displaces. The commonest is the Baumé hydrometer, of which there are two kinds, one for liquids heavier than water and one for liquids lighter than water.

The instrument consists simply of a glass float with a vertical cylindrical stem, graduated in units known as degrees. The float is immersed in the liquid, and the stem kept vertical by being weighted with a little mercury in the lower bulb. The depth to which the stem



ALBANY SECTIONAL DUMPING GRATE

sinks indicates the density of the liquid in degrees Baumé.

To convert degrees Baumé into specific gravity, the following formulas are used:

For liquids heavier than water,

$$Sp.gr. = \frac{145}{145 - deg.\ Baume}$$

For liquids lighter than water,

$$Sp.gr. = \frac{140}{130 + deg.\ Baume}$$

For instance, Fullerton oil, with a

FIG. 1. HEAVY-LIQUIDS HYDROMETER

density of 19 deg. Baumé has a specific gravity of

$$\frac{140}{130 + 19} = 0.9396 \text{ or } 0.94$$

To derive these formulas the method of calibration of the instruments must be understood. Consider first the "heavy-liquid" hydrometer, Fig. 1. This instrument is so constructed as to sink to a point near the top of the stem when immersed in distilled water at 12 deg. C. (53.6 deg. F.) This is the zero point of the scale. When immersed in a solution of 15 parts of common salt to 85 parts of water (sp. gr. 1.116) the instrument floats with more of the stem exposed than before, because less displaced liquid will equal the weight of the hydrometer. The point on the stem now at the level of the liquid, by convention is designated 15 deg. The space between this point and zero is divided into 15 equal parts, and the graduations are continued to the bottom of the stem.

To find the specific gravity corresponding to any degree on the scale, consider as unity the volume displaced when the instrument is immersed in water. This volume varies inversely with the specific gravity of the liquid, as is readily seen from the relation:

$$Sp.gr. \times volume = constant \times weight = constant$$

or

$$Volume = \frac{constant}{sp.gr.}$$

Consequently, when the instrument is placed in a liquid of specific gravity S, the volume immersed will be proportional to $\frac{1}{S}$, and the difference between this and the original volume $1 - \frac{1}{S}$. Assume that the hydrometer sinks in this liquid to a point h degrees below the original zero mark. Since the stem is cylindrical, this height h evidently must be proportional to the volume difference $1 - \frac{1}{S}$.

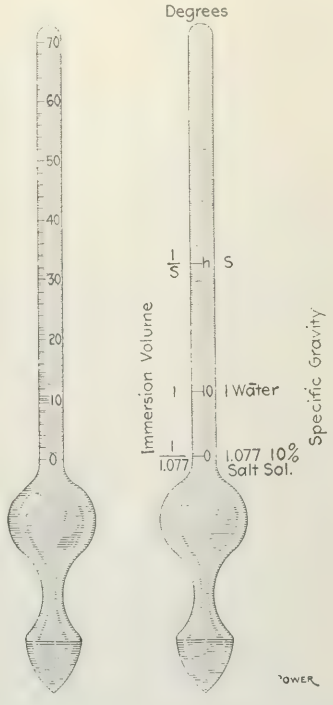


FIG. 2. LIGHT-LIQUIDS HYDROMETER

From the calibration it is known that when

$$h = 15 \text{ deg. } 1 - \frac{1}{S} = 1 - \frac{1}{1.116}$$

The general proportion therefore can be deduced

$$\frac{h}{1 - \frac{1}{S}} \quad \frac{15}{1 - \frac{1}{1.116}}$$

This proportionality may be readily seen from the diagram. The equation reduces to

$$S = \frac{145}{145 - h}$$

very nearly, where h is degrees Baumé.

The "light-liquids" hydrometer, Fig. 2, is similar, except that it is designed to sink to a point near the bottom of the stem when immersed in a solution of 90 parts of salt to 10 parts of water (sp.gr. 1.077), and this is taken as the zero point. The second calibration point is obtained by immersing in distilled water. The stem sinks deeper than before, and the point reached is designated as 10 deg. The graduations are continued to the top of the stem, the numbering being upward instead of downward as in the heavy-liquid instrument.

If h is measured from the zero point, the relation exists that h is proportional to the change in volume, which is now

$$\frac{1}{S} - \frac{1}{1.077}$$

For water,

$$h = 10 \text{ and } S = 1$$

so that

$$\frac{1}{S} - \frac{1}{1.077} = 0.0715$$

and the proportion obtains that

$$\frac{h}{1 - \frac{1}{S}} = \frac{10}{1 - \frac{1}{1.077}} = \frac{10}{0.0715}$$

From which

$$S = \frac{140}{130 + h}$$

very nearly, where h is degrees Baumé.

The above formulas are those generally used in America. Others have been worked out from different points of calibration, and using different (and usually more correct) values of the specific gravity for the calibration points. This accounts for discrepancies in different tables for specific gravities, corresponding to degrees Baumé. It should also be noticed that the formulas are used to apply to liquids at 60 deg. F. For other temperatures corrections must be made, as explained in POWER, Mar. 19, 1912.

New Generators for Keokuk

At the Schenectady works of the General Electric Co., the 7500-kw. generators for the Keokuk, Iowa, hydro-electric development are nearing completion.

Because of the slow speed of the low-head turbines, these generators are of great size for their rated capacity. They are of vertical-shaft type, 32 ft. in diameter and 12 ft. high, weighing over 600,000 lb. assembled. The turbines are of single-runner type with a 25-in. shaft revolving at 57.7 r.p.m. In spite of a low rotative speed, the peripheral velocity rises to about 4620 ft. per minute or 52.5 miles per hour. The turbine runners are about 15 ft. 8 in. in diameter and weigh 140,000 lb. each. The complete runner, including the generator rotor, weighs about 550,000 lb.

Canadian patent No. 138,432, says a contemporary, covers "an explosion and the method of producing it." The "explosion" probably took place in the printing plant, the editor furnishing the method.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

A Series of Grounds

By V. T. Kropidlowski

Two of our generators connected as in Fig. 1 and supplying current for shop lights and to a motor driving a locomotive transfer table, had been overheating considerably whenever the transfer table was used continuously for some time. One day generator No. 1 refused to carry its share of the load, and, moreover, when the motor was first started the pointer of the ammeter ran up even with that of the other machine, but then immediately dropped back to zero and remained there. I also noticed that the motor was taking more current than usual. From this it appeared that there was something wrong with the motor as well as the generator.

I decided to test the motor for grounds and having no special instruments for that purpose, used a 16-cp. lamp with sufficient cord to make the connections shown in Fig. 2. One end of a piece of wire was attached to one jaw of the main switch and the other end to the rail. One of the free ends of the lamp cord was connected to the other jaw of the switch, and with the other free end I began touching the contact rings of the controller, but no ground appeared. I then touched the contact fingers, and when on Nos. 1 and 2 the lamp lighted but burned brighter when touching finger No. 1. Of course, while this was being done the controller was in the off position, and the main switch was open.

Upon tracing the circuits from these fingers they proved to be the two armature leads. By a process of elimination, I continued disconnecting the leads from the brush-holders and touching the commutator segments, but the ground still showed. Upon removing the brushes from the holders and trying the commutator again no ground showed, but upon touching the brush-holders the ground again appeared. This led to an examination of the brush-holders and disclosed the fact that the nut and bolt securing one brush-holder to the frame was in contact with the motor casing—a good metal-to-metal contact.

Upon looking at the bolt of the second brush-holder I found it also grounded, but not a metal-to-metal contact, in which case a dead short-circuit would have existed. It developed that the brush-holder became warped, allowing No. 1 to come in contact with the frame, and at No. 2 dirt was lodged between the nut and the frame. By taking out the brush-holder frame and straightening it this trouble was eliminated.

Having the ground cleared at the motor, I returned to the engine room and found generator No. 1 still refusing to carry its load, although the motor was not exacting any more current than its load required. The machine acted as if its series field was not active, so I disconnected the series field terminals and passed a current through it, using the same incandescent lamp as that for locating the ground in the motor, but found that nothing was wrong there.

I then began looking for loose connections, and finally discovered that the negative cable of No. 1 machine could easily be pulled out of the connector where it connected with the fuse at the back of the switchboard (indicated at *C*, Fig. 1), the solder having melted out from around it. This accounts for the generator picking up its load at the instant the motor was started, but at the same instant the heavy current pulled away slightly the loose end in the socket and the effect was very much the same as opening the circuit by pulling its switch. After repairing this, the machine carried its share of the load as before.

But this was not all; another puzzle developed. After having apparently corrected everything, I decided to test the generators for grounds, using the same lamp for this purpose, and found a ground heavy enough to light the lamp brilliantly, although the frames were supposed to be insulated.

After trying for a week to locate this ground, I found it where the brush lead is soldered to the series field on No. 1 machine, as indicated at *A*, Fig. 1. There was a heavy lead thimble poured around the joint, but through carelessness in pouring, some of the lead had flowed onto the pole-piece, thus making a good contact with the frame. This had been covered by the tape and therefore had not been noticed before.

As to how the circuit was completed through the ground, when the machine was supposed to be insulated will be made clear by referring to Fig. 3. Water is used on the bearings of the machines, the piping being as shown, and the ground was evidently completed through

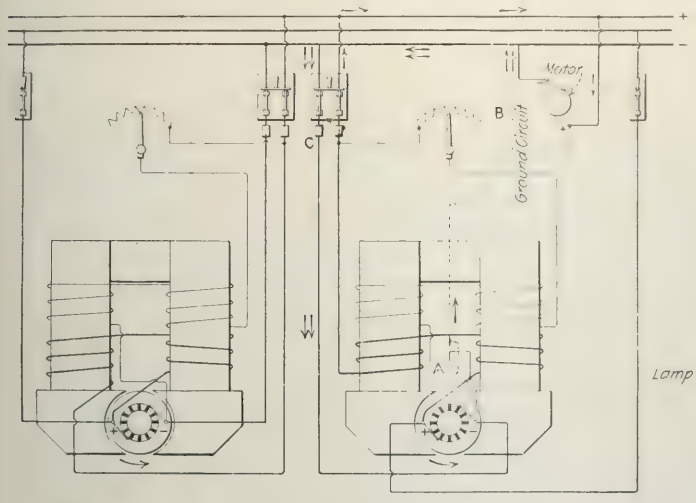


Fig. 1. Generator Circuits

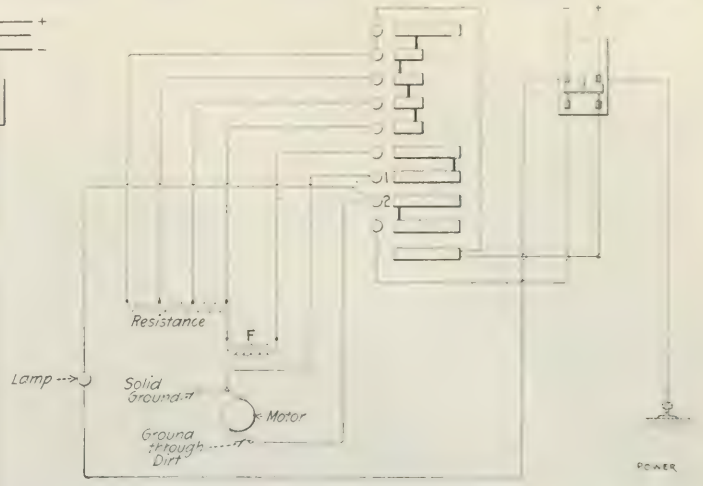


Fig. 2. Showing Method of Testing for Ground in Motor Circuit

the stream of water, the circuit being as indicated by the dotted lines.

The melting of the solder in the socket at *C*, Fig. 1, was due to the brush lead being grounded to the frame through the solder (indicated at *A*, Fig. 1), and to the grounds existing at the motor. The metal-to-metal contact with the ground existed at the negative brush of the motor, and the ground at the generator was at the positive brush. If it had not been for the ground current going through the stream of water (which had a comparatively high resistance) a dead short-circuit would have resulted and something would have burned out. As it was, however, there was a steady flow of current in addition to that supplied to the motor (which was heavier than normal, due to its double ground) but not enough to burn out anything in a short time. Eventually, however, it melted the solder in the connector of the negative lead.

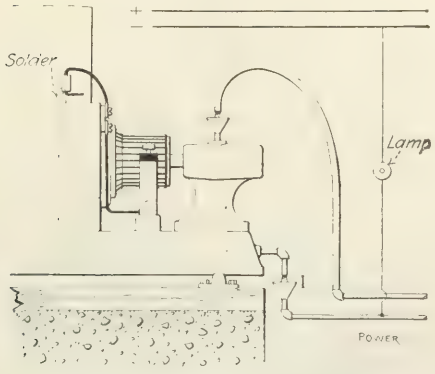


FIG. 3. SHOWING HOW GROUND WAS COMPLETED THROUGH SOLDER AND WATER

Referring to Fig. 1, the single arrows indicate the current in the positive leads and the double arrows the motor current plus the ground current, which did not go through the positive leads, but entered the ground at *A* and the negative leads at *B* at the grounded brush of the motor.

CORRESPONDENCE

Ground Caused Motor to Start

In Fig. 1 is shown the wiring diagram of the switchboard control for the steam turbine in the plant under consideration. It consists of a small electric motor having two field windings and mounted on the governor stand. Through the medium of a worm gear this motor varies the tension of the governor spring, thereby varying the steam admission and consequently the speed of the turbine. The two field windings are for running the motor in either direction—only one winding being in use at a time—and the direction of rotation is determined by the position of the single-pole double-throw switch shown.

This switch is on the switchboard and is normally held open by a spring at the motor. There are two limit switches which are opened automatically by the moving element of the control apparatus to prevent jamming the parts should the switch fail to open on the switchboard.

From the exciter busbars (to which the motor is connected) lights are also supplied about the station as indicated. One evening the motor suddenly started of its own accord and ran the moving element down; this opened the limit switch for that direction of rotation, but the motor kept running until the blade of the limit switch opened so wide that it jammed into the gear wheel of the control and stopped the motor. It was found in this position smoking hot with the current still passing through it.

The fuses were immediately pulled to cut off the current, and an investigation made. This developed that a ground had occurred in one of the field windings and another on the negative side of one of the lighting circuits, thus furnishing a complete circuit which was not broken by the opening of the limit switch. The path of the current was from the ground in the field coil to the ground in the light circuit and back to the negative busbars of the exciter. This is plainly indicated in the sketch, Fig. 1.

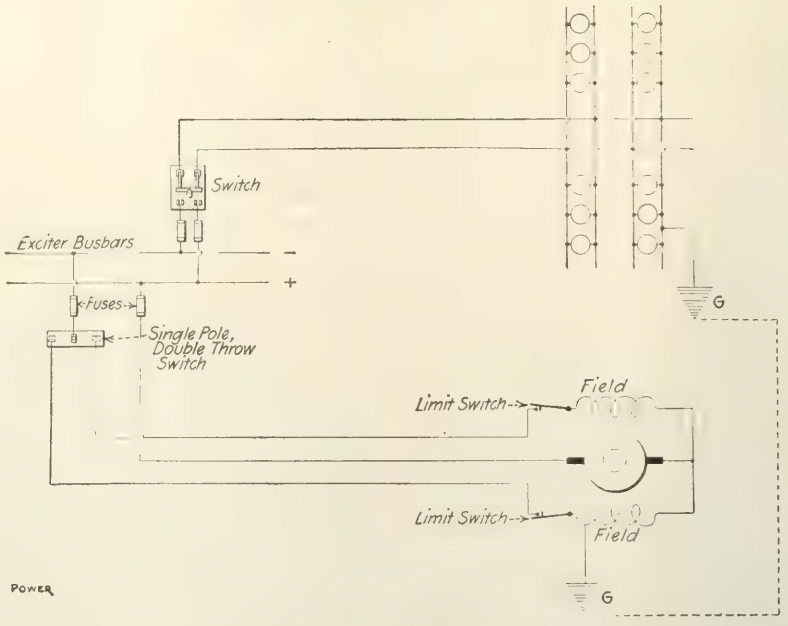


FIG. 1. CONNECTIONS WHICH CAUSED TROUBLE

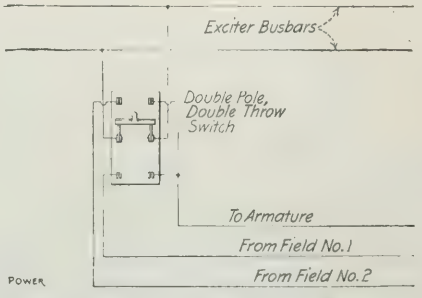


FIG. 2. WITH THESE CONNECTIONS THE TROUBLE COULD NOT HAVE OCCURRED

Had the wiring of the motor been as in Fig. 2, the trouble would not have happened.

C. L. GREER.

League City, Texas.

What Made Transformers Heat?

In answer to C. J. Brindley's inquiry in the July 2 issue as to why the insulation melted on his current transformers when the secondary coils were open-circuited, I would offer the following explanation:

For measuring instruments, transformers are designed to have small magnetizing currents and small coil-flux densities; the magnetomotive force of the secondary when closed is nearly equal to that of the primary and opposed to it. When the secondary is opened the magnetomotive force due to it disappears and therefore the core has acting upon it a large unopposed magnetomotive force, which causes the core flux to increase, thereby increasing the secondary potential.

If the resistance of the secondary circuit is fixed, the secondary current will increase in proportion to the secondary voltage and hence in proportion to the current in the main circuit. This increased current will burn out a transformer if the secondary coils are left open-circuited when disconnected from the instruments.

W. C. FISHER.

Brooklyn, N. Y.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

American Practice in Rating Internal Combustion Engines—II*

By T. C. Ulbricht and C. E. Torrance, Jr.

Mechanical Efficiency

From a large number of values obtained from American manufacturers and operators, the following average table was derived for the mechanical efficiencies of engines operating on various fuels:

Producer Gas	82.0 per cent.
Natural Gas	84.0 per cent.
Illuminating Gas	84.2 per cent.
Gasoline	87.7 per cent.
Oils	84.8 per cent.

These averages are on the basis of fuel used, instead of the engine type and erage thermal efficiencies of American engines, letters were sent to about ninety of the largest manufactures in the United States, requesting guarantees on the brake horsepower, thermal efficiency on this basis, kind and calorific value of fuel upon which the guarantee was based, and variation of guarantee, if any, with the size of engine.

The thermal efficiencies were mostly calculated from the guaranteed fuel consumption at full load, by the formula,

$$\text{Thermal efficiency} = \frac{2545}{\text{B.t.u. per b.hp.}}$$

the numerator being the B.t.u. equivalent of one horsepower-hour.

The curves, Figs. 13 to 20, show the results as obtained in most cases from actual guarantees given by the manufacturers, and since the tendency seems to be to under-rate the engines, or place the guarantee on the safe side, it would seem that these average curves represent good practice and are too low rather than too high.

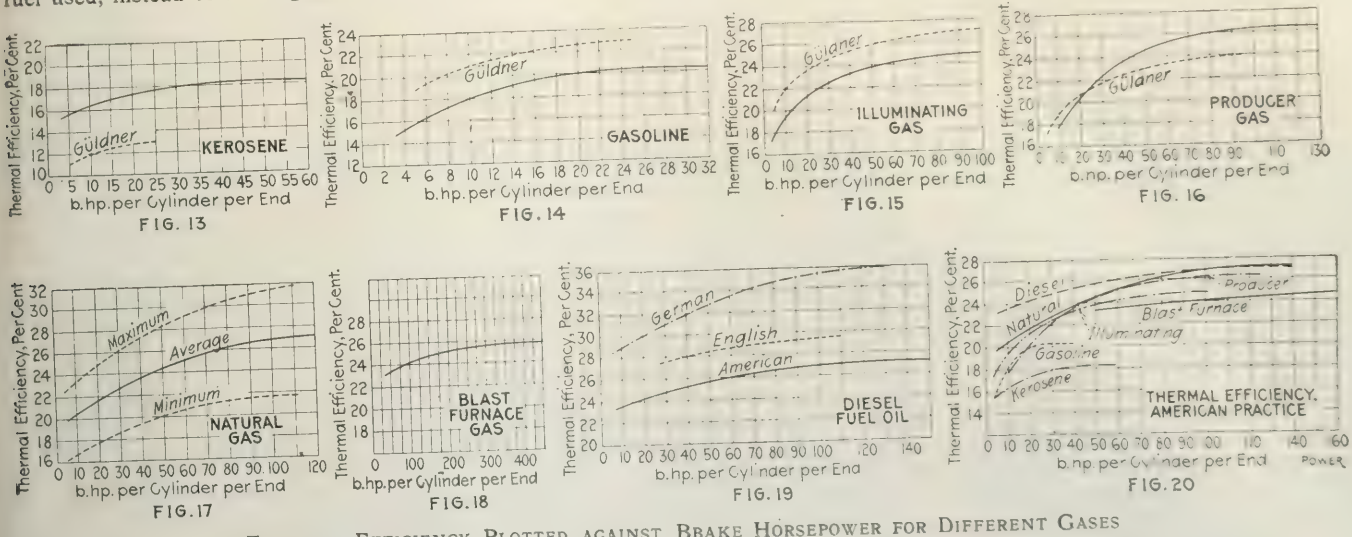
Fig. 13 shows the average thermal efficiency curve for engines using kerosene. Where possible, Güldner's values for German practice have been plotted on the same sheet with those representing American practice. For kerosene, Güldner's curve is found to be below the average for American practice. This discrepancy is probably due to the fact that when Güldner wrote, some ten years ago, very few oil engines had been developed, and the thermal efficiency was consequently low.

The thermal efficiency is seen to increase with an increasing brake-horsepower, approaching 18.4 per cent. above 40 b.hp. Fig. 14 shows the average thermal efficiencies for gasoline, which is lower than Güldner's curve for German practice. This would indicate either a higher development of the German gasoline engine, or a too conservative guarantee by the American builder. The efficiency is low for small horsepowers, but increases until above 22 b.hp., where it reaches a maximum value of 20.4 per cent. Güldner's value at 25 b.hp. is 23 per cent.

Fig. 15 also shows that for illuminating gas, Güldner's curve lies above that for American engines. At 100 b.hp., American practice shows 25 per cent., while German practice shows 27 per cent. For producer gas, American practice in general gives results higher than the German, as will be seen from Fig. 16.

As may be expected, due to the great variation in the analysis of natural gas, there is a wide range in the thermal efficiencies of engines using this fuel. (Fig. 19). Therefore the average, maximum and minimum curves are given. German practice does not include this gas. The average curve reaches a maximum of 27.3 per cent. at about 120 b.hp. Very few values were obtainable for blast-furnace gas, but the curve, Fig. 18, is what might be expected from good practice, showing a maximum of about 26 per cent. above 400 b.hp.

Special efforts were made to obtain values for the Diesel engine, and the result (Fig. 19) show the German values of thermal efficiency to be highest, the English somewhat lower and the American



Thermal Efficiency Plotted against Brake Horsepower for Different Gases

*Copyrighted July, 1912, by T. C. Ulbricht and C. E. Torrance, Jr.

still lower. This may be accounted for both by the longer period of development of this engine abroad and the higher quality of workmanship in Germany than in other countries.

Fig. 20 shows all the preceding average curves, for American practice only, reduced to the same scale for comparison.

Thermal Efficiency Drop with Increase of Compression

In discussing this phase of the subject, it will be assumed that the mixture can be regulated and cooling effected so that preignition will not occur. With each increase of compression pressure, there is a corresponding increase of maximum explosion pressure. At some point this pressure becomes such that further increase is impossible, as the friction in the bearings, pins, etc., increases rapidly and places undue wear on the material. This increase in friction decreases the mechanical efficiency, which directly affects the brake horsepower, resulting in a decrease of thermal efficiency above a certain point.

Also, with an increase of compression and explosion pressure, the maximum temperature becomes excessive, resulting in a large and rapid loss of heat along the expansion line to the jackets, in radiation, etc., so that above a certain point, the thermal efficiency decreases with any increase in compression pressure.

In the theoretical Otto cycle, it can be shown from the efficiency forumla,

$$E = 1 - \frac{Pd}{Pa}$$

that as the compression pressure Pa increases, the fraction Pd/Pa becomes smaller, giving a constantly increasing efficiency. Actually, however, as already explained, practical considerations so limit this increase, that above a given point (determined by Güldner to be 200 lb. per square inch) the thermal efficiency actually decreases.

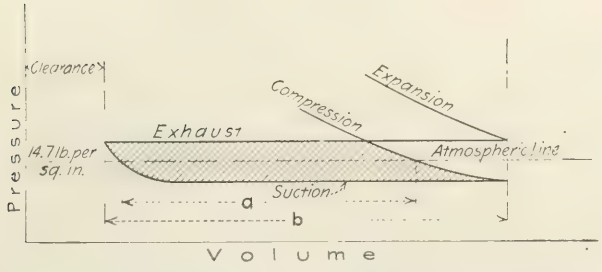


Fig. 21. "Lower Loop" Diagram

Volumetric Efficiency

The volumetric efficiency of a gas engine is the ratio of the actual volume of fuel mixture taken into the cylinder, to the piston displacement. It varies with the compression pressure, the piston speed, the suction resistance, the temperature during the suction stroke, etc.

It is usual in laboratory experiments to consider a "lower-loop" diagram, made by an indicator with a light spring, and actually measure the lengths a and b, shown in Fig. 21. Then the apparent volumetric efficiency is $a \div b$, but this does not take into account the temperature changes.

If a' is the volume of the charge at 14.7 lb. per square inch pressure and 62 deg. F. temperature, the actual volumetric efficiency is $a' \div b$. The method used in calculating the volumetric efficiency is illustrated in the following example for an actual engine using producer gas. Where the thermal efficiency was not given with the test, it was taken from the curves already given. In most cases, the correct ratio of fuel to air, as determined in the article on oils and gases (to appear later), were used in this calculation.

Example: The engine used for this determination was of the single-acting horizontal type, having a 21¼-in. cylinder, a 34½-in. stroke and developed 125 hp. at 145 r.p.m., corresponding to a piston speed of 834 ft. per minute. The following results were obtained:

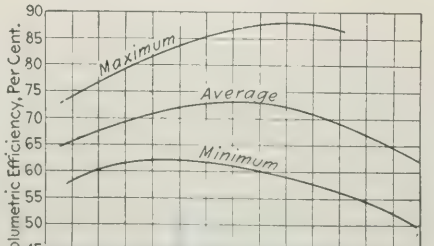


Fig. 22. Variation of Volumetric Efficiency with Piston Speed

Piston displacement = 7.08 *cu.ft.*

B.t.u. per hour = 2545 × *b.hp.* = 318,125

Cycles per minute = *r.p.m.* ÷ 2 = 72.5

$$B.t.u.\ per\ cycle = \frac{318{,}125}{72.5 \times 60} = 73.1$$

The thermal efficiency is given as 29 per cent.; hence each cycle requires in the cylinder

$$\frac{73.1}{0.29} = 252\ B.t.u.$$

One cubic foot of gas has an average heat value of 120 B.t.u., therefore the cubic feet of gas required are

$$252 \div 120 = 2.1\ cu.ft.$$

For producer gas the ratio of air to gas is from 1:1 to 1:1.25. With 1:1 there would be required 2.1 cu.ft. of air, giving a total mixture of 4.2 cu.ft. The volumetric efficiency is, therefore,

$$4.2 \div 7.08 = 52\ per\ cent.$$

With a ratio of air to gas of 1:1.25 there would be 2.63 cu.ft. of air required, making a total mixture of 4.73 cu.ft., and giving a volumetric efficiency of

$$4.73 \div 7.08 = 67\ per\ cent.$$

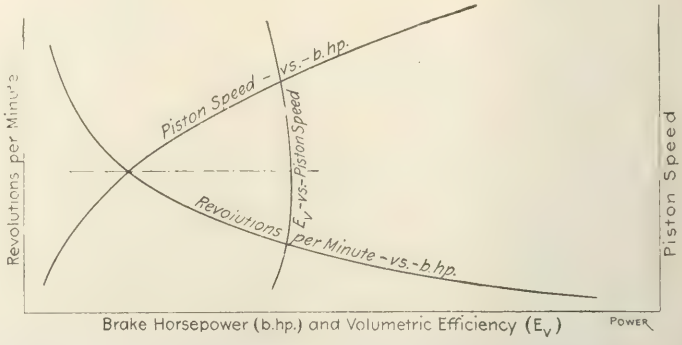

Piston Speed - vs.- b.hp.
Revolutions per Minute
E_v -vs.- Piston Speed
Revolutions per Minute - vs.- b.hp.
Piston Speed
Brake Horsepower (b.hp.) and Volumetric Efficiency (E_v)
Power

Fig. 23. Showing Point at which Maximum Charge Enters Cylinder

Fig. 22 shows the average, maximum and minimum values of volumetric efficiencies (E) plotted against piston speeds. The average curves show that engines of low piston speed have a low volumetric efficiency, but this increases with the piston speed until a maximum is reached at about 700 ft. per minute piston speed, after which it rapidly drops. This may be explained as follows:

For small engines of high rotative speed and short stroke in proportion to the cylinder diameter, the piston speed is low. The time for accelerating the charge volume is very short, resulting in a low final velocity. The curves plotted between cylinder diameter and valve diameter (to appear in a later article) show that they are proportional according to the following relation:

$d_v = 0.345\ d$ (*for the inlet valves*)

$d_v = 0.350\ d$ (*for the exhaust valves*)

d_v being the diameter of the valve and d the cylinder diameter.

Because of this, and the low final velocity and short time for acceleration, the charge volume is reduced, resulting in a correspondingly low volumetric efficiency.

On the other hand, as the sizes become large, the rotative speed is reduced, but the stroke becomes longer in proportion to the cylinder diameter, giving high piston speeds. The valve diameter is

proportional to the cylinder diameter in all sizes, but the piston speed has increased much more rapidly than the cylinder diameter, giving a very high maximum velocity to the incoming charge. The time for acceleration has been slightly increased, and the

However, in sizes intermediate between the limits cited, the volumetric efficiency is found to reach a maximum value. This is because in the intermediate sizes, there is a point where the rotative speed becomes proportional to the piston speeds (see Fig. 23), and

ciency is lowered, together with the developed horsepower.

Mean Effective Pressures

As the rating of a gas engine depends indirectly on the mean effective pressure, the accompanying mean effective pres-

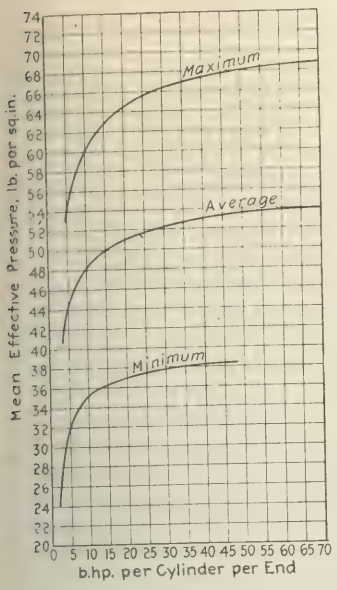


Fig. 24. Oils and Distillates

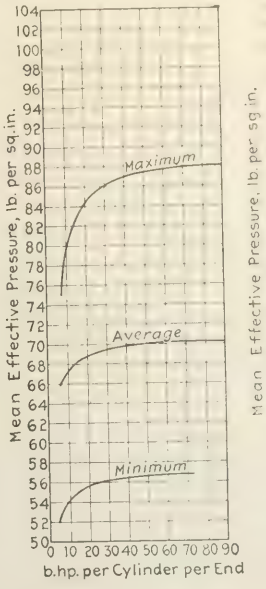


Fig. 25. Gasoline

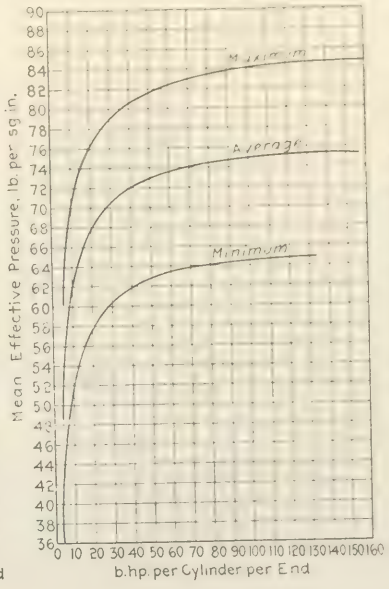


Fig. 26. Illuminating Gas

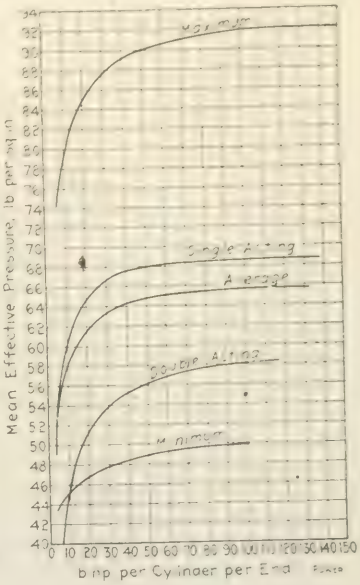


Fig. 27. Producer Gas

weights of gas entering a small and a large engine cylinder are proportional to two air columns, each of 1 sq.in. sectional area, which lengths represent the distances between the ends of the respective cylinders and their piston faces at the point of maximum acceleration for the small and large cylinders.

Because the valve diameter is still pro-

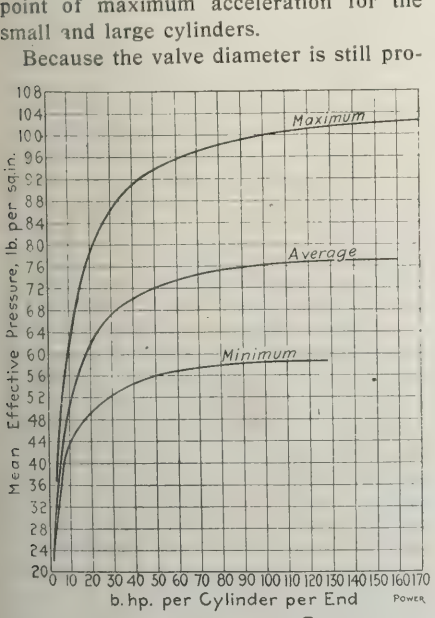


Fig. 28. Natural Gas

portional to the cylinder diameter, in all sizes, and also as the final velocity attained in the second case is very high, the air column, or charge volume is practically "wire drawn," or reduced in area, resulting in a charge volume which is too small in proportion to the cylinder volume and therefore giving a low volumetric efficiency.

therefore proportional to the cylinder volume. In this field, therefore, the velocity of the charge volume and the time allowed to acquire this velocity have just become balanced, allowing a maximum amount of charge to enter the cylinder, which results in a maximum volumetric efficiency. The volumetric efficiencies, as determined for American practice, are found to be somewhat lower than Güldner's values for German practice.

GULDNER'S VALUES OF VOLUMETRIC EFFICIENCIES

1. Slow speed, inlet valve mechanically operated	88 to 93 per cent.
2. Slow speed, automatic inlet valves	80 to 87 " "
3. High speed, inlet valves mechanically operated	78 to 85 " "
4. High speed, automatic inlet valves	65 to 75 " "
5. Very high speed automobile engines, with automatic inlet valves and air cooling	50 to 65 " "

This may be accounted for in several ways, chief of which is the universal tendency of American manufacturers to under-rate their engines. A careful study of the data received, shows that a majority of the engines will carry from 10 to 30 per cent. overload, which would mean a higher volumetric efficiency.

Another possible reason for the large variation between maximum and minimum values, is that many manufacturers fail to state the elevation for which a given engine will develop a given horsepower. This is important in the proper rating of gas engines, because at a high altitude an engine will not develop as much power as at sea level, due to the difference in pressures. Hence, the charge volume at the high elevation is less, and therefore the volumetric efficiency curves have been calculated and plotted between brake horsepower per cylinder per end, as abscissae and mean effective pressure in pounds per square inch as ordinates.

The mean effective pressure was found from the following formula:

$$i.hp. = \frac{plax}{33{,}000 \times 12}$$

in which

$i.hp.$ = Indicated horsepower per cylinder per end;
p = Mean effective pressure;
l = Length of stroke in inches;
a = Area of piston in square inches;
x = Number of explosions per minute.

Representing the cylinder diameter by d and the revolutions per minute by n,

$$a = \frac{\pi d^2}{4} \text{ and } x = \frac{n}{2}$$

Substituting these values

$$i.hp. = \frac{p \frac{\pi d^2}{4} n l}{33{,}000 \times 2 \times 12} = \frac{pd^2ln}{1{,}010{,}000}$$

Solving for p,

$$p = \frac{1{,}010{,}000\ i.hp.}{d^2ln}$$

Having already determined the mechanical efficiency for the different cases, the relation between indicated horsepower and d^2ln were found from the curves plotted between $b.hp.$ and d^2ln (Article I). Using this relation in connection with the foregoing formula, curves were plotted between mean effective pressure and brake horsepower per cylinder per end. These are shown in Figs. 24 to 28, inclusive.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Means for Accumulating Refrigeration—I

By Charles H. Herter

This article deals with means available for accumulating or storing up refrigeration, particularly in cold-storage rooms where the temperature must be kept as nearly uniform as possible. Reserve refrigerating capacity allows operating intermittently and provides for the contingency of accidental shutdowns, which is an advantage in plants equipped with but one machine.

In "Relative Costs of Continuously and Intermittently Operated Refrigerating Plants," in the issue of Sept. 19, 1911, the writer showed that, as a rule, plants with a capacity of 10 and more tons of refrigeration per 24 hours are most economical in yearly operating expense when run continuously in summer. Some of the following remarks apply especially to such plants. In the winter intermittent refrigeration is quite common, the low temperature outdoors making 24-hour operation unnecessary. In certain localities electric power may be had at favorable rates between peaks at the station load, and there intermittent refrigeration even for larger plants might prove as economical as continuous 24-hour operation. Labor, power cost, size of equipment, life of plant, kind and quantity of goods stored, etc., differ greatly with various installations, and it is therefore not possible to cover all conditions in a brief article.

The temperature rises in a cold-storage room, when refrigeration is shut off, mainly by heat penetrating the inclosing walls. Goods not yet cooled to the room temperature will also raise the temperature, but to simplify the calculations this will be ignored for the present. If the walls, floor and ceiling are not well insulated or the doors do not close tight, heat may get in so fast that the desired temperature can be maintained approximately uniform only by refrigerating continuously. Therefore, close attention must be paid to the fit of the doors, windows and other openings to see that they close air-tight with little effort.

A high-sill door is kept tight more easily than one with a low sill, because the joint is continuous around the opening and the high sill is also free from wear due to trucking. Some refrigerator-door manufacturers make a flexible gasket which is subject to great wear, and, as the cold, heavy air seeks to flow out at the bottom this source of leakage should receive constant attention. To test for air leakage, blow tobacco smoke against the cold side of the closed door; if any comes through the crevices the door should be refitted. A mistake frequently made is ordering doors unnecessarily large, because they cost little more than smaller ones, whereas the latter are more easily kept tight, displace less square feet of wall insulation, and allow less loss of cold air when the door is opened.

Windows in cold-storage rooms are not used by those aware of their low insulating value. For a temperature of 32 deg. F. a cold-storage room should be so insulated that the heat flow from the outside to the inside does not exceed 2 B.t.u. per square foot per 24 hours per degree Fahrenheit air temperature difference. Using two thicknesses of glass inclosing one air space for a window, the heat transmission coefficient is 13 B.t.u., while even with four thicknesses and three air spaces it is 6 B.t.u., with air-tight windows. Glass containing 1 to 2 per cent. of iron oxide, imparting a greenish tint, is preferable as it passes no radiant heat.

The amount of material stored also affects the maintenance of temperature. A meat room 20x40x15 ft. high, aggregating 12,000 cu.ft., can hold 50,000 lb. of meat. If the four walls are brick, 16 in. thick, insulated inside with ½ in. of mortar, 3 in. of corkboard, and ½ in. of portland cement finish, making a total thickness of 20 in., the heat-transmission coefficient is about 2 B.t.u. If the floor, built directly upon the ground, is 6 in. of concrete, 3 in. of corkboard coated bottom and top with hot asphalt, and a covering of 3 in. of concrete wearing floor, the heat-transmission coefficient is 2.3 B.t.u. If the ceiling (not a roof) is 6 in. of concrete with the insulation laid on top where its application is cheaper than underneath, the construction to be the same as for the floor, the heat loss will be 2.2 B.t.u. Thus, the heat entering this well insulated room per 24 hours will be

	Sq ft.	Deg.	B.t.u.
Walls,	1800	× (80–32) × 2	172,800
Floor,	800	× (60–32) × 2.3	51,520
Ceiling,	800	× (75–32) × 2.2	75,680
Total			300,000

equivalent to 12,500 B.t.u. per hour.

In large and important installations the heat leakage for the different walls should be calculated separately according to the method used by heating and ventilating engineers and a higher outside temperature should be assumed on the sunny side than on the north side, also allowances made for wind exposure, etc., but with an imaginary building such refinements are not necessary to the subject in hand. For the same reason, the varying amounts of refrigeration for day and night need not be considered, it being sufficiently accurate to assume mean temperatures to be constant during 24 hours.

Chilled meat is stored at between 31 and 36 deg. F., depending upon its kind, the degree of humidity, the duration of storage and individual opinion as to how it should look and taste. Siebel, in his "Compend of Mechanical Refrigeration," says: "Meat, kept constantly at 31 deg. F., can be kept fresh six months, but if allowed at times to go as high as even 33 deg. is likely to spoil in less than a month; however, if the ventilation and humidity are properly regulated it should keep about two months in good condition in the latter case." By operating intermittently and with the air temperature never below 32 deg., actual freezing of the meat is avoided and refrigeration is not burdened with the latent heat of freezing.

The foregoing 300,000 B.t.u. of heat will raise the temperature of the 50,000 lb. of meat, after 24 hours' shutdown

$$\frac{300{,}000\ B.t.u.}{50{,}000\ lb. \times 0.8\ spec.\ heat} = 7.5\ deg.$$

or 3 deg. in 9.6 hours. If the room contains only 25,000 lb., the meat temperature will rise 3 deg. after a stop of only 4.8 hours. These figures are merely approximations; in reality, much of the heat, as shown later, is retained by the walls, and little or none reaches the inner portions of the cooled meat. Freshly stored carcases with the inner temperature still above that of the room will appropriate the available refrigeration until the temperatures are equalized. By considering the total square feet of surface of the relatively warm meat and the coefficients of radiation and convection per degree of average temperature difference obtaining between meat surface and surrounding air during the time limit set, this quantity of heat may be determined.

As the foregoing shows, food stored in a well filled room will suffer less during a shutdown than when stored in a room practically empty; also the amount of insulation influences the temperature rise

during a shutdown. If, as is often true, the heat leakage averages 4 instead of 2 B.t.u., 600,000 B.t.u. will enter the room in 24 hours, not only increasing the work of the refrigerating machine but also causing a 3-deg. temperature rise to take place after 2.4 hours' stopping. The cold air confined in the room need not be included in these figures because its capacity for heat is only 0.24 B.t.u. per pound per degree, only

$$12{,}000 \times 0.0807 \times 0.24 = 232 \text{ B.t.u.}$$

being required to warm up all the air in the room 1 deg. when empty. With material in storage the amount of air will be much less.

Before calculating the heat the walls can absorb, several other factors may be considered which help to maintain temperatures; for instance, the frost collecting on the cooling pipes. On 1000 lineal feet of 1¼-in. pipe coated ¼ in. thick with snow weighing approximately 7 lb. per cubic foot, there would be 73 lb. of snow, its sensible and latent heat not amounting to more than 144 B.t.u. per pound, or a total of 10,512 B.t.u. For completeness might be included the heat absorbed by the metal of the 1¼-in. extra-heavy cooling pipes when warming from, say, 12 deg. to 32 deg., this being

$$3000 \text{ lb.} \times 20 \text{ deg.} \times 0.116 = 6960 \text{ B.t.u.}$$

The total for snow and pipes would be 17,472 B.t.u., an amount sufficient to lengthen the period of shutdown 1.4 hours.

The shut-down time can be longer when brine instead of ammonia is used in the cooling pipes. With the brine system the average temperature difference between circulating brine and air may be 16 deg., necessitating the use of

$$\frac{1000 \text{ ft.} \times 20 \text{ deg.}}{16 \text{ deg.}} = 1250 \text{ ft.}$$

of 1¼-in. extra-heavy brine pipe, to do the same work as 1000 ft. of 1¼-in. direct-expansion piping. This piping when filled contains 11 cu.ft. of brine. The heat capacity of a cubic foot of salt brine per degree Fahrenheit is about 58 B.t.u., and of chloride of calcium brine, 52 B.t.u. Using the latter figure, the 11 cu.ft. in rising 16 deg. absorbs

$$11 \times 52 \times 16 = 9152 \text{ B.t.u.}$$

Contrary to general opinion, the brine in the pipes delays the 3-deg. air temperature rise by only

$$\frac{9152 \times 24}{300{,}000} = 0.73 \text{ hour}$$

Altogether, then, the plant can be shut down 6.93 hours and not rise more than 3 deg. in temperature.

	Hours
25,000 lb. of meat	4.8
Snow and pipes	1.4
Calcium brine	0.73
Total	6.93

If the insulation used allows heat to enter twice as fast, the air temperature will be up to 35 deg. after a stop of only

$$\frac{6.93}{2} = 3.46 \text{ hours}$$

and if the room contains only 5000 lb. of meat, the stopping time dwindles to a total of 1.55 hours.

Good insulation not only permits operating the plant with the least refrigeration, power, time and cost, but also helps to reduce fluctuations in room temperature. After stopping refrigeration the inflow of heat continues, although at a constantly decreasing rate, until after days the inner air temperature will be within a few degrees of the atmospheric temperature.

The stopping period of 6.93 hours just calculated, representing a heat influx of 86,624 B.t.u., is further affected by the temperature variations of the walls, floor and ceiling. These in turn are the effect of temperature changes taking place outdoors. Hence before proceeding, a table giving the hourly temperatures outdoors must be decided upon as a basis. At the December, 1896, meeting of the American Society of Mechanical Engineers, F. H. Boyer read a paper containing an engineer's log of a large meat-packing plant at Boston, Mass. In this are recorded the outdoor air temperatures for July 18, 1896, the average for the 24 hours being 69 deg. F. An average of 80 deg., the temperature used herein, will represent the true condition for a larger territory than 69 deg. Boyer's hourly readings must therefore be raised about 11 deg. Then the hourly temperatures will be as in Table 1.

TABLE 1. HOURLY TEMPERATURES

Increasing Temperature Period	Deg.	Decreasing Temperature Period	Deg.
5 a.m.	70	2 p.m.	89
6 a.m.	73	3 p.m.	87
7 a.m.	77	4 p.m.	85
8 a.m.	80	5 p.m.	83
9 a.m.	82	6 p.m.	81
10 a.m.	84	7 p.m.	80
11 a.m.	86	8 p.m.	79
12 a.m.	88	9 p.m.	78
1 p.m.	90	10 p.m.	77
Average	81.1	11 p.m.	76
		12 p.m.	75
Average temperature for 24 hours, 79.6 deg.		1 a.m.	74
		2 a.m.	73
		3 a.m.	72
		4 a.m.	71
		Average	78.6

Table 1 shows that the lowest temperature obtains at 5 a.m. and the highest at 1 p.m., thus there is an increasing temperature period of eight hours and a decreasing temperature period of 16 hours. To add to the complication, at night the room temperature rises and the outdoor temperature falls, at which time the rate of heat flow will be correspondingly less per hour than during the day when the conditions are reversed.

To get an insight into these temperature fluctuations it will be sufficient to examine the four insulated walls and assume that these contribute 57,800 B.t.u., or three-fourths, toward the total 86,624 B.t.u. allowed to enter before the permissible 3-deg. temperature rise has taken place in all contents of the room. To analyze all surfaces separately is hardly justified here, particularly as the amount of goods stored, their initial temperature, the actual atmospheric temperature, and many other factors are so variable as to upset any fine calculations.

To determine the heat conducted through a solid wall of any material, the following formula can be used:

$$\text{B.t.u. conducted per hr.} = \frac{C}{\text{thickness in in.}}$$

where C is the coefficient of conductivity, or the heat conducted through 1 in. thickness, in heat units per square foot per degree (Fahrenheit) difference in wall temperatures per hour.

The coefficient of conductivity of brick with a specific gravity of 1.98, or 123.5 lb. per cubic foot, is, according to Peclet, 5.56 B.t.u. Then for a 16-in. brick wall the heat transmitted per degree difference of temperature between the two wall faces is

$$\frac{5.56}{16} = 0.3475 \text{ B.t.u.}$$

per square foot per hour. If this wall, as in the meat room under investigation, is covered with insulating material—namely, with a ½-in. layer of cement mortar, 3 in. of corkboard, and a ½-in. finishing coat of portland cement plaster, total thickness 20 in.—the heat conducted from the outer to the inner face per degree wall temperature difference can be calculated according to the following formula:

$$\text{B.t.u. per sq.ft. per hr.} = \frac{1}{\left(\frac{d}{c} + \frac{d_1}{c_1} + \frac{d_2}{c_2} + \frac{d_3}{c_3}\right)}$$

where d, d_1, d_2 and d_3 indicate the respective thicknesses and c, c_1, c_2 and c_3 the corresponding coefficients of conductivity.

For fine plaster of 1.25 specific gravity (78 lb. per cubic foot), Peclet gives $c = 4.2$ B.t.u. per hour for 1 in. thickness. The specific gravity should always be stated because the conductivity varies with the density. It varies also with the temperature, but not sufficiently to affect these calculations. The heat transmission through some good grades of American compressed corkboard 2 in. thick varies, according to recent tests, from 3.2 to 3.5 B.t.u., depending upon the quality or make, per square foot per 24 hours per degree difference in temperature between the air on the two sides

From personal research, the writer has found that he must distinguish between the amount of heat passed through a test piece per degree air temperature difference and that passed through per degree wall-surface temperature difference. The necessity for such distinction has never been emphasized; nevertheless, so long as radiation and convection are to be reckoned with, there must be a difference or drop in temperature between a wall sur-

face and the surrounding air. To calculate the heat passing through a compound wall according to the accepted formula, the conductivity must be known of each separate layer as referred to its wall-temperature difference.

With practically still air the heat radiated per square foot per degree difference per 24 hours by smooth steam-heating pipes is 30 B.t.u. The radiation from a corkboard up to a certain thickness at least will not differ materially from this value, for H. Recknagel, by actual tests, found it to be correct for rough stone walls. Under constant conditions as much heat as enters a wall, passes through and leaves it. Accordingly the following formula applies:

$$H = \frac{c}{d}(t_2 - t_3) = a_1(t_1 - t_2) = a_2(t_3 - t_4)$$

where

H = B.t.u.;
c = Coefficient of inner heat conductivity (per degree wall-temperature difference);
d = Thickness of wall (over all);
a_1 = Total coefficient of radiation (including convection) at the warm side of the wall;
a_2 = Total coefficient of radiation (including convection) at the cold side of the wall;
t_1 = Temperature of warm air;
t_2 = Temperature of warm wall surface;
t_3 = Temperature of cold wall surface;
t_4 = Temperature of cold air.

An analytical treatment of conditions met with in testing insulating materials is not really pertinent to the subject of this article; still, as some readers may desire additional information regarding the effect air agitation has upon K, the coefficient per degree air-temperature difference $t_1 - t_4$, this subject will be discussed in a later article.

Troubles in an Absorption System and Remedies—I

By W. S. Luckenbach

As the season for producing ice and refrigeration to the full capacity is now upon us, every man in charge of an ice plant needs to be resourceful to maintain it at full capacity. The writer, therefore, believes the following comments from his experience will be timely. Some apply to both the absorption and compression system, but most of them to the absorption system.

First in importance is a full charge of aqua ammonia at not less than 26 deg. density, 28 is better, if the ammonia pump is in first-class order, as less heat is required to liberate the gas. It is essential to notice at what pressure steam in the generator will do the best work, viz., to produce the most cold with the least fuel consumption, because, the lower the steam pressure giving the desired results the less cooling water must be pumped and consequently less fuel consumed.

To get best results the gas liberated in the generator should be not less than 14 per cent., and by referring to the standard table we find that 25.7 deg. (Baumé scale) aqua ammonia contains 28 per cent. ammonia and at 18 deg. 14 per cent.; hence with the rich liquor at, say, 26 deg. and the poor liquor at 18 deg., the proper distillation is accomplished; should, however, the rich liquor be 24 deg., then the poor should be 16 deg., to obtain the 14 per cent. of gas. This, however, can only be accomplished from the same amount pumped into the generator by the application of more steam heat. Hence the importance of maintaining a high standard of aqua.

It may happen that on one day the rich liquor is found to be 26 deg., and the poor liquor 18 deg., while the next day the rich liquor is 24 deg. and the poor liquor 20 deg. It will puzzle some as to what caused the change and what is the remedy.

This change can only take place in the exchanger. The hot poor liquor passes from the generator through the exchanger in one direction and the comparatively cool rich liquor passes in the other direction, thereby exchanging temperatures, and if one of the pipes, or coils leaks, the density of the poor and rich liquor will change. The remedy is to make a bypass and cut the exchanger out at once, and have the pipes removed as speedily as possible.

Sometimes the liquor in the machine gets low, while running in the ordinary way, and, after adding more ammonia, gets low again in a few days, with no leaks of any kind. This may occur when charging or recharging and distilling; considerable air enters the system and will occupy space in the condensing and cooling coils, finally finding its way to the absorber from which it is purged. It, however, frequently happens that the liquor lies dormant in the bath coils; especially may this be the case if the bath in the freezing tank is on a level with the machine room, instead of being above the liquor level in the absorber, thereby establishing a gravity system. The first move is to produce a partial vacuum so that any liquor lying in the bath coils will flow to the absorber. To do this, the liquor in the generator should be allowed to get low and the liquor in the absorber correspondingly high, then the poor liquor valve should be closed at the absorber and the generator filled as full as possible to avoid a boil-over. This will lower the liquor in the absorber and produce a partial vacuum.

CORRESPONDENCE

Brine Leak in Ice Tank

It is expensive to repair leaks in an iron ice tank when the tank has to be jacked up in order to get under it to drive the rivets. In fact, while letting the tank back into position, other rivets are loosened and new leaks are made. I stopped the leaks in a 30x40-ft. iron ice tank in a much quicker and less expensive manner.

The brine was pumped out of the tank into ice cans and the coils and tank were thoroughly washed. The coils were then set over to one side, leaving half of the tank to be worked on. This half was again washed and mopped thoroughly dry; all the seams and rivets were heated with a blow torch and brushed while hot with a stiff fine wire brush until all the scale and dirt were removed, leaving a bright surface. The seams and loose rivets were calked, using air. Where the heads of the loose rivets would not stand calking, the rivet was driven out and a tap rivet put in. The seams and rivets were then heated as before, and while hot they were painted with a mixture of hot coal-tar pitch and Ferrubron paint. This mixture was liberally applied, spreading the paint 4 or 5 in. on each side of the lap of the seam.

In making this mixture 75 lb. of Ferrubron paint was used to the barrel (300 lb.) of pitch; this paint gives elasticity to the pitch and thus tends to prevent cracking. The other side of the tank was treated in a like manner. This tank has been in service for two seasons, and, so far, has shown no sign of leak.

Gay A. Robertson.

Louisville, Ky.

Wear in Ammonia-Compressor Cylinder

I would like to offer the following as a possible explanation why horizontal ammonia-compressor cylinders wear more at the top than on the bottom surface of the cylinder, as related by Mr. Anderson in the June 18 issue. When a compressor is fitted with a spherical piston head the pressure tends to centralize the piston, the wet gases, and oil coming in on the rod will flow to the bottom surface of the cylinder and lubricate it, while the top surface remains comparatively dry, and consequently will show wear before other parts.

As stated, this condition applies to the spherical piston. Will Mr. Anderson give us his experience with the flat piston head, that is, do both types show the same wear?

J. H. O'Hern.

Denver, Colo.

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POWER

Temperance

In our general campaign for the welfare of those engaged in the field which we serve, we have felt that we could not ignore the importance of temperance in the use of intoxicants—hence the foreword in this issue. It is one of POWER's functions to point directions in which the engineer may improve himself or his conditions. Another is to suggest ways for increasing the efficiency of the plant, which naturally depends in a great measure upon the efficiency of the operator and that is never at its highest if he is a heavy drinker.

Two things that brought the subject of temperance prominently to mind at this time are alluded to in the foreword, a wreck which occurred on the Lackawanna R.R., July 4, in which 46 were killed, and the engineer was alleged to be under the influence of liquor, and the statements made by John L. Sullivan in a recent interview, concerning his career of hard drinking. His advice to others does not stop with temperance, for he recommends leaving liquor entirely alone. He claims that among all the men he knew, which was a very great number, he could not recall more than a dozen abstemious drinkers.

Time may have been, as some have claimed, when a man was not considered a gentleman unless he drank to excess, but happily that day has long since passed. Could we now but banish the treating habit we would remove the most subtle evil of the whole thing. In the several social clubs that have prohibited treating, drunkeness is now unknown.

Still, even with treating persisting, the strong minded man can refuse alcoholic drinks when he has had as much as is good for him, and take a cigar or a soft drink. Those who always do this can do it with better grace than formerly, for the tendency to ridicule the man who has back-bone enough to stick to his principles, or who knows what is best for him, is steadily growing less. Even habitual drinkers very generally now respect a man's desire to stay "on the water-wagon."

Where safety to human life requires temperance, no further argument is necessary. All railroads now require it, and more and more are other industries, if not demanding, at least putting a premium on men for responsible positions who do not over-indulge.

Boiler Accident Prevention

The stays in the dome of one of two boilers, connected together by a line containing a slip expansion joint having no studs and check nuts, were to be removed from a boiler in a brewery in Nanaimo, British Columbia, recently. One of the boilers was under pressure at the time of removing the bolts from the flange connection at the idle boiler. When the engineer removed the last bolt from the flange the pressure separated the expansion joint and the escaping steam killed him.

If the facts are as stated it would seem that the engineer was either careless or thoughtless. While the last few bolts were being removed he should have noticed the strain on them caused by the separating of the expansion joint. Surely a heavy blow on the last bolt must have been required to remove it. It is pitiable that he did not heed the warning.

The government inspector of boilers and machinery testified that he inspected the plant nine months previous to the accident. We understand that the laws of British Columbia require studs and check-nuts on such joints, but the inspector considered it safe and did not compel their use. Even though he acted in good faith, he is, in a measure responsible according to our understanding of the law.

Furthermore we cannot undestand how a competent jury could do no more than recommend a more rigid enforcement of the law as stated in *The Nanaimo Free Press* of June 25.

Equally responsible are the engineer in charge of the plant and the man who erected the faulty joint, if the owners did not refuse to equip the joint with

studs and check-nuts. This case is one of many and should serve to caution those having to do with boilers.

A boiler inspector's duties impose such grave responsibilities that they should not be entrusted to any who are not extremely cautious and thorough. Besides extensive knowledge of pressures, strength of materials, etc., a boiler inspector should have good judgment and an unrelenting vigilance upon which so much depends in safety to life and property.

Because a condition is permitted by law does not prove that it is safe. An inanimate license or certificate hanging on a wall will not prevent a disastrous flywheel explosion if an engine governor has been grossly neglected by the holder of the license. A license is indicative of good character, competency and scrupulous care on the part of the holder and he should aim to deserve its endorsement.

No amount of technical knowledge means much if unaccompanied by common sense. The Nanaimo accident and countless others are due to a deplorable lack of it.

Gasoline and Kerosene Supply

The oil companies seem to have taken advantage of the increased demand for gasoline, occasioned by the large number of motor vehicles and boats, by abnormally boosting the price. This is particularly noticeable in certain cities where the price has advanced 25 per cent. over that of last year.

In defense of their action the dealers and producers argue that the demand is fast exceeding the supply. Unfortunately the figures of the present year, with which to refute or substantiate this claim, are not yet available. However, while this economic theory may suffice as an explanation in the absence of other proof, it must be admitted that it costs no more to produce a gallon of gasoline now than it did a year ago; in fact, if we are to consider the usual effect of producing in quantities, it should cost less. Hence, it would appear that some are reaping enormous profits.

Let us consider how these conditions affect stationary practice, which is concerned not so much with gasoline as with the other products of crude petroleum. For every gallon of gasoline there is produced two or more gallons of kerosene and considerable fuel oil. We are told that the oil companies regard the kerosene as a byproduct and are shipping large quantities to foreign markets, where it is disposed of at a low figure, over twenty million barrels being exported last year alone.

Why not keep more of this kerosene at home and utilize it for power purposes in oil engines? Surely it would be more profitable to sell this to home trade, if a market could be created, than to compete in foreign markets, with the additional cost of shipping.

The "Principe di Piedmonte" Accident

The accident on the Italian liner "Principe di Piedmonte," an account of which appears on page 172, is of much concern to all steam users. This accident and that on the "North Dakota" in 1909 are identical, and many similar ones have happened in stationary practice.

These cases well illustrate the importance of non-return stop valves. Had the boilers of these vessels, and those in stationary practice where tubes have let go, been provided with them, the results would have been less serious. Even with several exits in a boiler room, and in most of them there are not, the chances of escape at such times are small enough, but to be entrapped in a space quickly filled with scalding steam is horrible beyond imagination.

Deep in the consciousness of every power-plant man, however competent he may be, there lurks the fear of a boiler under ordinary working pressures, and though he has never experienced a serious burst, a broken gage glass is indicative of what is possible.

However bad it may be when the steam from one boiler is suddenly liberated in a room, it is far worse when several connected to a common header belch steam and water that flashes into steam at the extreme pressure reduction. The fires cannot be drawn, and fortunately there is no need of it, as the dying fires would not produce enough heat to evaporate the water left in the boilers. A man's first thought, on such an occasion, is justifiably his own preservation.

It does not cost much to equip boilers or supply pipes with non-return valves or other means to prevent a discharge from all the boilers when only one is affected. The accidents that have occurred where no such provisions are made should be a warning to those building new plants or operating old ones. Desire to safeguard human life should be strong enough to bring about these reforms voluntarily without waiting for law to make them compulsory.

More Publicity Needed

Usually, when a central station succeeds in taking over an isolated plant, it is announced through the daily press, or literature is mailed to the business man who may be a possible purchaser. The central station, knowing the commercial value of publicity, seldom misses a chance to advertise its service. The isolated plant needs more publicity.

It is pertinently urged by *The Isolated Plant* that the manufacturer also address himself directly to the business man if he would hope to induce him to buy power-plant equipment and generate current instead of buying it. Now, he appears to be playing into the hands of the central station.

Very often home-made power would be cheaper than purchased current, but a campaign of education and publicity is badly needed to make known under what conditions the isolated plant is to be preferred. The business man listens to the louder voice, and at present the central-station people are making all the noise.

A wiseacre has recently gathered figures, according to *Harper's Weekly*, in which he attempts to show that in firing one ton of coal an hour (eight tons a day) a fireman expends but little more energy than the young-girl typist in operating a typewriter that length of time. He expends about 20,000 ft.-lb., she, 15,000 ft.-lb. He must add the shovel's weight to get his figure. She must produce 12,000 more or less cruel words to get hers. He (meaning the statistician this time) will probably "get his" if he starts this line of argument with a fireman.

Has the site of the original power plant been discovered? A British engineer claims that the Garden of Eden was located in Egypt, and it is proposed to restore the home of Chief Adam to its pristine glory. What a chance this offers to the efficiency engineer!

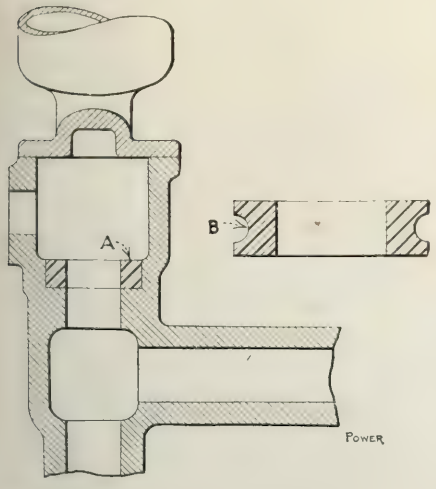
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Valve Seat Repair

The sketch shows one of the discharge chambers of a high-pressure pump, with the steel valve seat *A* in place. This pump works against 1500 lb. pressure on an accumulator for hydraulic work. The seat, which is supposed to be a press fit, came loose, allowing the fluid to pass between it and the cylinder walls, soon enlarging the walls.

Examination showed no alternative but to take the cylinder off, rebore and fit



VALVE CHAMBER AND STEEL VALVE SEAT

it with a new seat, valve, etc. As we could not spare the pump long enough for such a job, I took the seat out and turned the edge off, as shown at *B*. The cylinder was then heated to dispel all moisture and expand it as much as possible. After replacing the seat, the space between it and the cylinder wall was filled with good, hard babbitt and then well calked to make it tight. This job was done two years ago and the seat has shown no signs of coming loose.

H. H. DELBERT.

Titusville, Penn.

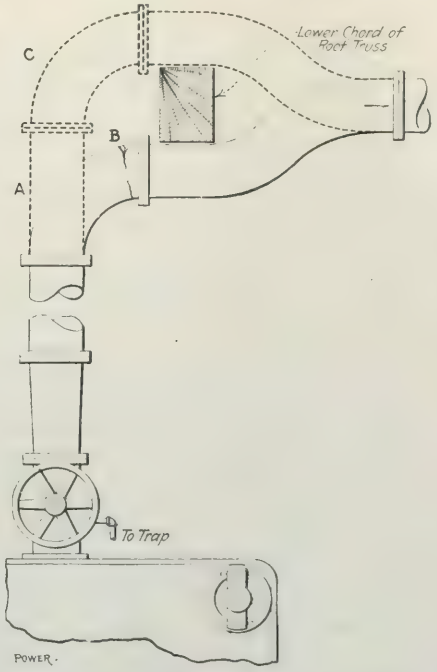
Damaged Steam Pipe Due to Faulty Design

At a cotton factory where I was employed the 10-in. pipe that conducted steam to the engine was arranged as shown by the full lines in the illustration.

The intention was to run the pipe in a straight line, but owing to a mistake, the lower chord of the truss that supported the roof of the engine house was so placed as to prevent it.

A special offset section of pipe was therefore made to go under the truss and the vertical piece shortened accordingly. In the elbow at the bottom of the vertical pipe is a ½-in. drip pipe, connected to a trap that is supposed to carry off the condensation accumulating in the pipe when the engine is not running.

Occasionally the drip arrangement fails to keep the pipe free of water, and if it is not noticed by the watchman, the result is a severe water hammer after the water gets above the line *A*. The offset section of pipe forms an ideal water pocket which, if permitted to go on filling, is sure to do damage. On one occasion the elbow was cracked at *B* and displaced by a new one, causing considerable expense and attendant interruption of service. Before the new elbow



PROPOSED CHANGE IN PIPING TO PREVENT ACCUMULATION OF WATER

was put in, I suggested turning the offset piece of pipe up, to go over the truss beam and lengthening the vertical piece to meet it, as shown by the dotted lines.

I explained that if the trap failed to work, no harm would result while the vertical section is filling with water, because the action is slow and the steam cannot mix with the water, therefore no water hammer could occur, and after the water reached the level of line *C*, it would trickle back to the boilers.

My suggestion was not approved by the company's representative, who said the pipe was designed by an expert and was all right. This answer was about as sensible as the mother's when a doctor asked about the milk she was feeding the sick child, and she replied: "Oh, I guess the milk is all right, because it says 'Fresh Country Milk' on the milk wagon."

Later on, either the trap failed to work or the valve in the connecting pipe may have been shut, for the 10-in. pipe broke and the steam rushed into the engine room until the boiler stop valves were closed. The cost of this break must have been considerable, because aside from the expense of replacing the pipe, the steam spoiled three large leather belts, and the mill was shut down until new belts arrived and were adjusted to place.

After this experience one would suppose that the company would surmise that something was wrong and not take further chances. The fact is, the pipe was replaced just as originally installed, which is evidence of their confidence in the judgment of their expert.

Under such management, when the cost of maintenance and operation are figured, they must indeed be suffi iently high to give the solicitors for the central stations and hydro-electric transmission companies an opportunity to put up a good argument in favor of stopping the local plant and buying power from the companies they represent.

J. W. PARKER.

Clinton, Mass.

Stopping Leaky Rubber Diaphragms

Frequently, after rubber diaghragms have been used for some time, they become slightly porous and give considerable trouble in the form of mysterious little leaks. This is especially noticeable in heating systems, controlled by electric thermostats and diaphragms operated by low-pressure air.

Recently I learned of a cheap and effective way of overcoming this difficulty. Whenever it became difficult to maintain the air pressure by a small hydraulically operated air compressor, a helper was sent out to try the rubber diaphragms with a brush and soapy water to detect leaks. If any were found, as evidenced by bubbles forming over the surface, the diaphragms were removed and the inside

surface given a good coat of ordinary molasses, after which they were replaced. Many of these treated diaphragms gave longer and better service than new ones put in at the same time.

It is now this concern's practice to treat all of its leaky diaphragms with molasses and mark them with the date to keep a record of the service they give. It has been found that diaphragms made of canvas indurated with rubber and treated with molasses when signs of leaks begin to show, are the cheapest and most satisfactory.

The idea was borrowed from a similar treatment of old porous bicycle tires and found to be even more satisfactory. The molasses seems to have no detrimental effect on the rubber, but rather tends to keep it pliable without rotting it.

M. S. WEIL.

Philadelphia, Penn.

Firebrick Walls Prevent Smoke

We have had much trouble with the smoke inspector while using South Fork scalp level bituminous coal of the following properties in a 250-hp. water-tube boiler with 0.2-in. draft in the uptake.

	Per Cent.
Fixed carbon	72.56
Volatile matter	15.39
Sulphur	0.9970
Ash	9.15
Moisture	2.09
B.t.u. commercial	13,825

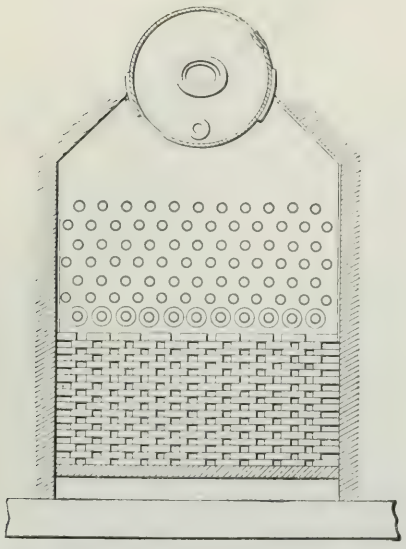


FIG. 1. END SECTION SHOWING BRICK SETTING

After obtaining prices on various devices for lessening smoke, we concluded that it would be cheaper to install more hot surface of such character that it might be easily repaired, and, if possible, of cheap material. We planned the surfaces shown in Fig. 2, using No. 1 firebrick at $20 per 1000. Four walls, *A B C D*, were built up dry to within 2 to 4 ft. of the tile under the tubes, and as shown in Fig. 1, the bricks were laid with spaces between making a checkerwork. In this way we added 84,825 sq.in. of surface to store the heat of the flames, which considerably lessened the amount of smoke.

FRANK MARTIN.

College Point, N. Y.

Taking Care of Drips

In manufacturing plants, when the area of the building is not over 100 sq.ft. and is run up a few stories, taking care of the drips is not simple, but in a large manufactory where the buildings are scattered over three or more acres, the drip problem becomes very much more complicated. I have found, during several years' experience, that engineers generally do not pay the attention to drips that their importance would seem to require. Often the sewer seems to be the favorite way to get rid of them, and as long as they are out of sight they are out of mind. Unless they blow back, no further attention is paid to them. Then usually further additions are made to the plant and again the drips go to the sewer, until in a few years the amount of condensation running to the sewer becomes considerable. Drips can generally be taken care of properly if a little thought is given them.

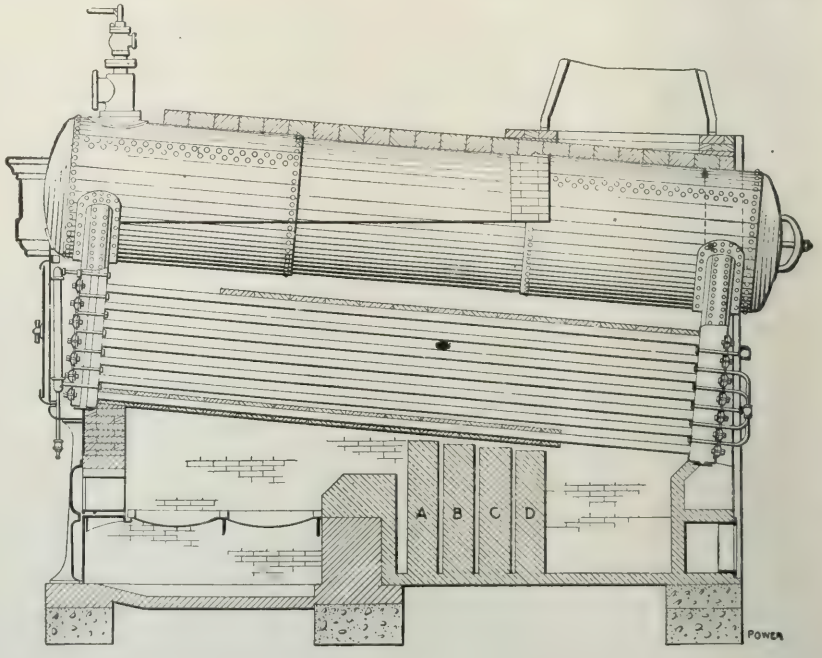


FIG. 2. SHOWING HOT WALLS FOR PREVENTING SMOKE

Plants occasionally grow beyond all expectations and, while the steam and exhaust lines will be enlarged, the same drip main is left to take care of the additional load. Sometimes it does, but often at the expense of half flooding the heating and drying apparatus, thereby not giving the results intended and possibly holding back the production, or putting more back-pressure on the engines. In such cases, a vacuum pump on the end of the lines will do the work nicely and will give quick results in circulation, and better drying, and the drip lines will be less likely to leak.

At some places the drips are not collected at all and cold water is fed through the heater. There generally is some waste in large plants, regardless of efforts to prevent it. Some managements go to the other extreme and spend more to save a few pounds of condensation than they will ever gain. Where the condensation amounts to a dozen or so pounds an hour, I have known long lines of drips to be run both over and underground, the cost and upkeep of which was out of all proportion to the saving effected.

Where the condensation eats the pipes rapidly it is occasionally cheaper to run a drip to the sewer; but it does not pay to forget about it. An inspection of traps at regular intervals is profitable in plants, large or small, for they always need looking after. I have seen traps that were in active service for about 15 years discharging into a river. Whether they were ever overhauled in all that time I could not find out, but when I saw them, they were certainly not holding back any steam.

If the condensation can be returned to the boiler that is the best disposition of it, unless there is considerable use for hot water when drips may be so utilized. Dripping the high-pressure into the exhaust is one way of saving long runs of pipe, but this cannot be done where much low-pressure steam is required, as

considerable of the discharge from the high-pressure main will not reëvaporate. I have found that a well in the exhaust, where a nest of high-pressure traps discharge, having a large low-pressure trap, answers the purpose very nicely.

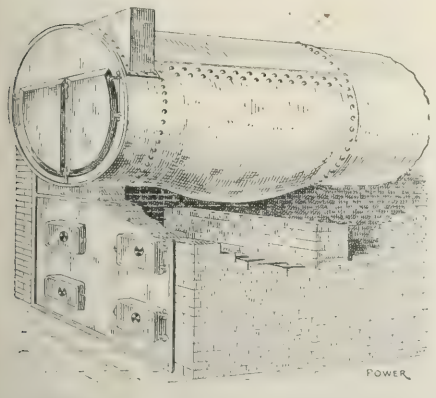
It is false economy to use an old trap on work for which it is not intended, rather than buy a proper one. How to care for drips is usually an individual problem, as conditions vary in different plants, and what would be suitable in one would not do in another. Hence no hard-and-fast rules can be given. The main trouble is that the drips are often neglected, whereas they should receive careful attention if the plant is to be run economically.

JAMES ROONEY.

Providence, R. I.

Badly Bagged Boiler

Herewith is shown a bagged boiler sheet extraordinary for the size and cause of the bag. I was not engineer at the plant when the boiler was burned, Feb. 12, 1912, but I took charge Feb. 27.



A 60x42x9 IN. BAG IN A BOILER PLATE

It is a return-tubular boiler 60 in. in diameter and 16 ft. long, and has fifty-six 3½-in. tubes. The bottom is made up of two 8-ft. sheets extending half way up on each side, having two double-riveted longitudinal seams and one girth seam. The bag was in the half sheet directly over the grates. It commenced about 3 in. from the girth seam, extended toward the front end of the boiler for 60 in., was 42 in. wide and dropped down 9 in., as illustrated.

No scale was in the boiler, but the plate was covered with about ⅛ in. of baked cylinder oil, due to letting the water get low in the heater (an old return-tubular boiler used as an open heater), so that cylinder oil found its way into the boiler. As near as I can find out, the boiler always had plenty of water.

On Monday morning, Feb. 12, after the engineer had started the engine and gone back to the boiler to throw a fire, he discovered the bag. The gage glass was nearly full of water and the pressure was 60 lb. No other reason being apparent, the oil must have caused the bag. The plant has two boilers, both the same size, but only one is used at a time.

The other boiler was in operation and it made me rather uneasy to know that I was firing a boiler undoubtedly coated with oil as the other had been. I had no chance to clean it until the other was ready for use, but I managed to keep the heater full of water and let some overflow to remove the oil from the surface of the water, and prevent more oil getting into the boiler. I commenced using soda ash, using two quarts a day for the first ten days, then feeding it every other day for the next ten days, and twice a week after that. By frequently blowing down and washing out the boilers, I have got rid of all the oil, and the boilers are now in good condition.

This shows what oil will do to a boiler, and also suggests a remedy for anyone that may happen to experience the same trouble.

WILLIAM DEBO.

Mansfield, Ohio.

Blowing Safety Valves Waste Steam

An employer once said to his engineer, when remonstrating about an unexpected discharge of the safety valve: "It was the most terrible, unearthly noise I ever heard, and if it occurs again you may consider yourself discharged."

"Well," replied the engineer, "there's this much about it; if you hadn't heard that noise you might 'a'heard a dern sight bigger one."

These expressions are perhaps typical of the general ideas regarding the purpose of the safety valve: a necessary nuisance, from the owner's standpoint, because custom, or the inspector, compels attaching it to the boiler. The engineer knows that it is to act as a relief for an unexpected or unavoidable increase of pressure beyond the safe limit of the boiler.

In many plants the pressure carried is but little lower than the pressure allowed, and a slight decrease in the demand for steam will cause the safety valve to open; if no objection is made this may be repeated at short intervals while the plant is being operated. It may be that the engineer is demonstrating his skill as a fireman, foolishly thinking that each blast proves his ability to master the steam-generating equipment, or, possibly the working pressure is carried as near the blowing-off point as possible to effect a short cutoff on the engine, for the sake of economy. The cost of the fuel wasted by the safety valve will generally offset the gain that may be credited to the shorter cutoff produced by the few pounds added to the initial pressure.

By an actual test on a boiler blowing off steam for four consecutive minutes, it was demonstrated that 6 cu.ft. of water, or 374 lb., was converted into steam and wasted. Taking 8 lb. of water as converted into steam per pound of coal burned, the amount of coal wasted in the four minutes was 46.7 lb.

CHARLES F. ADAMS.

Lynn, Mass.

Feed Pump Troubles and Repairs

When our feed pumps were run fast, one side would short stroke, the feed line shake and the check valves on the feed line slam continually.

I first examined the discharge pipe inside the horizontal tubular boilers and found it broken off in No. 2 boiler 2 in. from the reinforcing bushing inside the shell. This caused the slamming of the check valves as the feed water entered the steam space at 208 deg., while the temperature of steam at 120 lb. is 350 deg., therefore, causing a partial water-hammer. I then ground the check valves to a good seat on each boiler.

I next examined the water end of the pumps and found the plungers badly scored, and the suction and discharge chambers contained scale and fine pieces of concrete which caused the scored plungers. The pumps got their water from a tank which received the returns from the heating systems in 14 buildings.

The return tank was 12x10x8 ft., built of brick and lined with concrete ½ in. thick. The continual expansion and contraction eventually cracked the concrete and pieces of it then worked their way to the pump as the suction pipe had no strainer and the suction was taken from the bottom of the tank. I changed this so that the suction would be about 6 in. from the bottom and put on a strainer. The suction valves were in fair condition, but the discharge valves were too soft and the pressure had pushed them down between the spider arms of the valve seat. I ordered composition valves of the same material that is used in making disks for globe valves. I then had the valves faced and a groove made in them where they rest over the arms of the seat.

My reason for facing them was that after the valves are in use a short time the arms of the valve seats wear a ridge in the valve, and when it is turned by the continual opening and closing, this ridge keeps the discharge valves from closing or seating perfectly. Two new plungers finished the job and we had two good running pumps.

JOHN J. MAHONEY.

North Cambridge, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Water in the Low Pressure Cylinder

I have read the article "Water in the Low Pressure Cylinder" on page 898 of your issue of June 25.

I do not believe that this can be the explanation of damage from water in the low-pressure cylinder, because, the admission being only 0.3 of the stroke, the water cannot enter except during this period, but it has at least 0.8 of the stroke to get out of the cylinder. How, therefore, can it stay there to make trouble on the return stroke?

The case cited could produce damage to the high-pressure cylinder, because the water in the receiver could be drawn into this cylinder just at the moment when the exhaust valve closes for compression, when the engine was operating with so small a load that the pressure in the high-pressure cylinder fell below the pressure in the receiver.

H. Bollinckx.

Brussels, Belgium.

Improving a Neglected Plant

Mr. King's letter under this heading in the May 21 issue appeals to me as I have succeeded after much effort and with comparatively little expense in improving my plant, which is a central station.

When I took charge I found a most incompetent management. The manager acted as chief engineer with a first assistant who was too inexperienced for his responsibility. My coming was not pleasant to them as my contract called for a salary unusually high in that locality, and entire supervision of the plant. I was also to receive a 10 per cent. increase in salary every year until a stipulated amount had been reached. Should I fail to produce good results I was to resign. In six months my salary was increased 20 per cent.

I soon gained the support of the crew, as I had insisted on their highest efficiency and promised additional wages to each man in proportion to his worth. The men were instructed that those not meeting the requirements would have to go.

After putting the engines in good condition, I installed oil burners in the boiler furnaces, thus doing away with coal, and the service of two men. The wages of those retained were then raised. The feed-water heater only heated the water to 90 deg. and it was changed so that the water was heated to 210 deg. without using live steam. There were no feed pumps, injectors being the only means of feed, but I ordered pumps the first day I took charge. The oil-soaked wooden floor of the engine room was replaced by one of concrete. The swampy ground around the plant was filled in, and the drainage pipes repaired. A new foundation was also built under one engine.

The service has so improved that our output is greatly increasing and new machinery is being installed to meet the demands. Two large generating sets have recently been added and an order has been placed for a large Diesel oil engine.

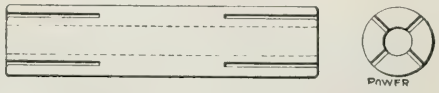
My experiences teach me that if the management will coöperate fully with its engineer, he will furnish the highest kind of service at a minimum cost. Most chief engineers will amply reward their assistants if they prove competent and progressive.

John Pierce.

Chicago, Ill.

Emergency Expansion Bolt

The expansion bolt mentioned by Mr. Daly in the May 28 issue would answer its purpose admirably, but most expansion bolts when once put into a wall



Sleeve for Expansion Bolt

cannot be removed without cutting them out, thus defacing the wall. For this reason the ordinary expansion bolt is losing favor and going out of use while one that can be removed and will allow the fixtures or machines to rest against the wall without having to be sprung out over the ends of the bolts is most desirable.

I put a bracket back of a large pipe, which extended about 6 in. from the wall, to support the pipe and also to keep it from vibrating. The bracket, which was made U-shape, could not be sprung out over the ends of bolts, and other means had to be resorted to. As the wall was very thick, and long bolts could not be passed through, some scrap lead was melted into sleeves 3/8x7/8x4 in., as shown, and sawed down from each end in quarters for about 1¼ in., leaving about 1½ in. of solid tube in the center.

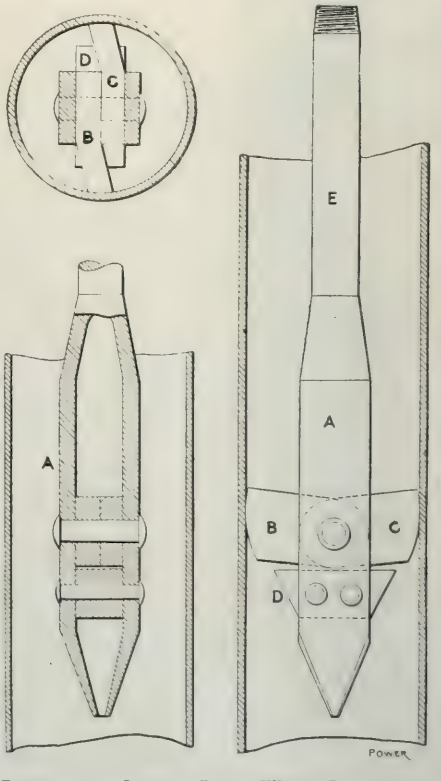
These were driven in the holes in the brickwork flush with the face of the wall and ½-in. lag bolts were then screwed into these lead sleeves, expanding them in the brickwork and making a very rigid, quick and neat job. The taper of the lag-bolt thread is sufficient to expand the lead sleeve and the coarse wood-screw thread gives it a very firm hold. Care must be taken not to have the center hole in the lead sleeve too small for the bolts or there is risk of breaking off the bolt before it can be screwed in tight.

R. A. Cultra.

Cambridge, Mass.

Lifting a Broken Plunger

F. H. Cordner's letter under this heading in the May 7 issue was particularly interesting to me as I have had much experience in boring wells and lifting broken plungers, tubing, pipes, etc., from them.



Device for Liting Deep Well Plungers

The tool illustrated is the one with which I have had the greatest success. It is made by welding together two pieces of 1½x3/8-in. iron *A*, shaped so as to re-

ceive the toggle members *B* and *C*. A square piece of iron *D* is riveted on, as shown, so as to act as a stop for the pieces *B* and *C*. By welding a short length of round iron *E* to the piece *A* and threading it to receive a length of pipe, additional weight is given the tool, which is desirable.

The tool is lowered into the well with a rope and the toggle pieces bind so hard that there is no slipping. The tool is very easily disengaged from the pipe or tubing.

W. B. Odell.

Birmingham, Ala.

Putting in Crankpins

Mr. Gilson's method of putting in crankpins, as described in the June 11 issue, would be rather slow. By heating the hole with pieces of hot shafting, I do not think he could get the disk hot enough.

With a pin of the proper size and the disk sufficiently hot, the pin will go in without any driving, but one must work in a hurry, for once the pin has entered the hole it will soon heat and expand enough to become tight. To be on the safe side, a ram should be made so that the pin could be driven in if it stuck. I prefer pulling a pin in with a bolt. Before the pin is taken out of the lathe, bore and tap the end for the pulling bolt.

For a 5-in. or a 6-in. pin use about a 1¼-in. machine-steel bolt. When all is ready give the pin a coat of white lead, square the pin with the disk and pull up on the nut. Be sure the bolt is a snug fit in the pin; otherwise it may pull out. Lightly tap the end of the pin as the bolt is pulled up; usually swing a piece of shafting from over head, but if there is not room, a hammer may be used. After the pin is pulled up, saw off the pulling bolt. I have put in several pins this way and have never heard of any giving trouble.

P. P. Majorbanks.

Bedlich, La.

Mr. Gilson might have saved fully 50 per cent. of the drilling work if, instead of drilling the holes near the circumference of the pin, he had drilled on the diameter, as shown in the illustration. As the ratio of one-half the circumference to the diameter is 1.5708 to 1, the effect in loosening the pin would have been the same.

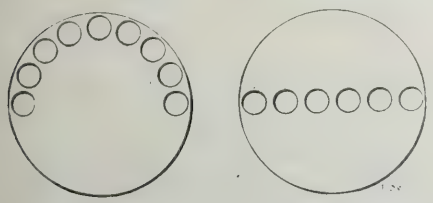


Fig. 1 Fig. 2

Position of Holes for Removal of Pin

Fig. 1 shows nine ⅞-in. holes on a 6-in. pin which are necessary, according to Mr. Gilson's way of removing the pin. In Fig. 2 is shown six ⅞-in. holes on the same size pin, which are enough to allow it to be readily removed.

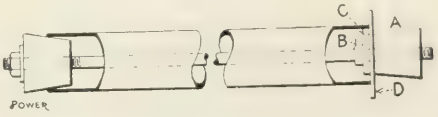
Charles F. Prescott.

Los Angeles, Calif.

Unnecessary Risks

In commenting on the editorial in the Apr. 30 issue, Charles J. Mason, under the above heading, in the June 4 issue, writes of a risk which he deems necessary.

I have plugged the tubes of Scotch marine boilers, but never deemed it necessary to toast myself over a bed of hot coals to do so. We used a cast-iron plug *A*, as illustrated, instead of a pine one, and the rod, plug, washer, etc., were pushed through the tube from the front end. The greatest diameter of the plug was slightly less than that of the tube. A shoulder *B* was made on the rod at the end of the threads, and the washer *C* was put on over the threads and against this shoulder. A piece of asbestos packing *D*, the diameter of which should be larger than that of the tube, was then slipped on over the rod and against the washer. The threaded plug was then screwed tight on the rod and the whole thing pushed through to the rear end of the tube with the aid of an extension handle.



Plugs in Boiler Tube Ready for Tightening

The rear plug should be pushed a little beyond the end of the tube to allow the asbestos packing to straighten out. The rod should then be pulled hard forward and tightened with the nut at the front end. The purpose of the packing is to enlarge the diameter of the plug and seal the joint.

Leighton Johnson.

Exeter, N. H.

How the Cylinder is Cut

The cutting of the top of the cylinder bore, described by William E. Dixon, in the June 11 issue, brings to mind a similar experience I had with a small engine. The first indication of trouble was a pound in the cylinder when running at about one-fourth rated load, which continually increased. When the cylinder head was removed it was found that the top of the bore was badly cut and the piston rings were very rough.

A good grade of oil was used in the plant, and other engines using it gave no trouble. Nevertheless, the cut cylinder appeared not to be receiving proper lubrication. Steam at 105 lb. pressure without superheat was used. I believe that even some of the high-grade oils will not mix with steam at this pressure, but will travel along the steam pipe and steam chest in drops as it is fed, and upon entering the cylinder very little of it comes in contact with the top, which would account for the wear at that point.

After repairing the cylinder and fitting new piston rings, it was decided to feed oil directly into the top of the cylinder at the middle of its length. A ⅜-in. hole was drilled into the cylinder and enlarged and tapped for a ¼-in. pipe, on which was connected a lubricator. Although little oil reached the top of the cylinder, it stopped the wear on both rings and cylinder.

That the other engines were not affected seems to be due to a difference in the iron in the cylinders, which is much harder and the stroke is also shorter. Since then a cheaper grade of oil has been used, and it seems to mix with the steam and reaches all parts with which the steam comes in contact.

P. L. Werner.

McKeesport, Penn.

All the symptoms described by Mr. Dixon tally closely with those I once experienced in a similar case, and the whole trouble was corrected by squaring off the ring, i.e., making the ring freer in the groove.

Some engineers are inclined to blame things that go wrong to almost any cause but the right one. For instance, superheated steam, boiler compounds, improper methods of lubrication or poor lubricating oil are often blamed, while sometimes the whole fault lies in lack of care to see that the piston rings can expand and contract freely before being placed in the cylinder. This very same cause, and its effect, that is, cutting a cylinder out at the top, once cost a well known engine builder a new cylinder.

Steam engineering is a profession from which all possible "ifs" and doubts should be eliminated. The surest way to settle any doubt about a ring being in running condition is to put the piston together complete, securing the follower as tightly as though it were in the cylinder, and then try a "feeler" all around the ring to make sure that it is somewhere near parallel. A light pressure, or a tap with a hammer handle, or block of wood at any point on the circumference of the ring, should move it to the side opposite. If it does not move freely, it should be

taken out, and the marks made by the contact of the surfaces will show where it should be scraped to clear the ring. It is well also to slightly round the edges with a file, as this removes all sharp corners and stops the tendency to scrape the lubricant from the cylinder surfaces.

One engine builder makes a piston with one ring, such as is described by Mr. Dixon. To prevent the ring from wearing a shoulder in the cylinder, a groove is cut at each end of the bore for the edge of the ring to come into, when the piston is at the end of each stroke.

H. R. Low.

Moosup, Conn.

The cause of the excessive ring wear is too much spring pressure. To overcome this, Mr. Dixon should straighten out the springs until they are all in place under the ring and the ring held flush with the face of the piston at a point opposite the cut. He can force the two ends together with a strong thumb pressure. If the springs will not straighten out by hitting them with a hammer, heat them to a cherry red and quench in ordinary engine oil. To get a uniform curve in each spring, lay one on the floor and mark its curvature with a soap-stone pencil, then heat the spring and straighten it to suit conditions.

The reason all the cylinders are not affected the same is due to several causes: variation in the hardness and set of the springs, also difference in the hardness and density of the metal in the rings. Very often it is due to the manner of casting the cylinders. Some builders cast all cylinders vertically with a 6- or 8-in. riser all around the cylinder bore. This allows sand, cinder and also the plumbago wash that becomes detached with other impurities to float to the top. This is then cut off in the shop and thrown away. Invariably such cylinders have metal of a uniform hardness and density in the bore. Others cast their cylinders horizontally, with the result that the impurities are occluded in the iron, either in the top of the cylinder, if cast with the steam side up, or on the bottom, if cast exhaust side up. Usually cylinders made this way have a soft, spongy spot somewhere in the bore.

Rings with too much spring pressure on them soon wear to a sharp edge that will act like a planer tool and cut even the hardest iron. The top of the cylinder bore gets little lubrication, unless the oil is thoroughly vaporized, or special provision is made to put it there, as in gas-engine practice. I would suggest that Mr. Dixon fit the pistons tighter in the cylinder bore and apply minimum spring pressure behind the rings, and also put a special oil pipe in the top of the cylinder.

His particular conditions will readily suggest how best to do it. Until he gets over his trouble, he should take out the piston rings at regular intervals and file off the sharp edge. If he follows the above suggestions, he will save the cylinders that have not yet been affected.

J. F. Nagle.

Cos-Cob, Conn.

That Red Hot Blowoff Pipe

In the issue of June 18, Mr. Hannemann inquires why his blowoff pipe became red hot if it was full of water. The natural reply would be that it did not. If the pipe was full of water it did not become red hot; if it became red hot it was certainly clear of water at that point.

Most probably mortar, or other rubbish in the boiler, got into the blowoff pipe and obstructed it; this sometimes happens with boilers just erected.

So much for Mr. Hannemann's troubles. I will mention some of my own experiences which run along a similar line. I used to be "head fireman" in a plant equipped with vertical water-tube boilers, fired with underfeed stokers and using mechanical draft. We had as nearly perfect feed water as I ever saw, making practically no scale. In those days, generation of steam was my specialty, and I took pride in keeping the safety valves just hissing all day.

Sometimes the load would increase and only an incandescent fire would maintain the pressure. Imagine my astonishment to find that the fire side of the tubes would bulge out with little bags about as large as half an egg. I had always been taught by precept and common sense that a clean sheet or tube could not bag, but here was a contradiction, for the tubes were as free of scale as when they were put in, and no oil got in our boilers. The only explanation we could give was that the intense heat impinging against the tubes would generate steam, making a bubble inside the tube, holding the water away for an instant, allowing the tube to become red hot and the pressure to bulge it. I never knew of a burned blowoff pipe which was not obstructed.

Recently a horizontal boiler in a large sawmill plant, fired with dry shavings, had the rear head among the tubes burned without the water being low in the boiler. We accounted for that only by the possibility of steam bubbles inside and by the fires being pushed to the limit.

C. R. Summers.

Chattanooga, Tenn.

Mr. Hannemann's trouble is a fair sample of power-plant happenings. I do not deny that a blowoff pipe could get red hot, but my experience with men in plants leads me to believe that he entertained a false impression.

With a boiler subject to an intense fire, one is apt to imagine when looking at a soot-covered sheet over the fire, that the sheet is bagged. I know of an engineer of 30 years' experience who, on looking under his boiler, became excited and declared it was red hot, while two gages of water showed in the glass. He pulled the fire and sent for an inspector, only to be told that he must have imagined it.

Mr. Hannemann is burning shavings which would cause a flame to travel the entire length of the boiler. On opening the rear door and looking at the boiler under these circumstances, one is apt to think the blowoff pipe and everything else is red hot. This is especially probable since the boiler was being put into service for the first time, and on such occasions an excess of anxiety conduces to such delusions.

Mr. Hannemann inquires as to the best way of drawing a sawdust fire; I would not draw one even in an emergency. Closing the ashpit and opening the fire-doors, and then covering the fire with about 8 in. of sawdust is best. The dampers should be left open.

A. Waldron.

Revere, Mass.

Soon after a fire was built under Mr. Hannemann's boiler, heat bubbles began to gather and move out through the open end of this pipe into the boiler until, as the heat became more and more intense, the amount of steam generated in this pipe was sufficient to keep the water driven back out of the pipe. Consequently, it is very plain why the pipe became red hot. The powder substance mentioned by Mr. Hannemann, was probably lime scale.

A careful perusal of the underlying principles upon which the Solignac-Grille boiler is based, as described on page 867 of the June 18 issue, would help Mr. Hannemann to understand why his pipe became red hot, since what they are proposing to overcome is analagous to this blowoff-pipe problem.

Lloyd V. Beets.

Nashville, Tenn.

Flue Gas Analysis

Commenting on the editorial under the above heading in the June 11 issue, I would state that six months ago I knew practically nothing of flue-gas analysis. Our evaporation per pound of coal from and at 212 deg. F. was 5.5 lb. Now it is over 10 lb., almost 100 per cent. gain.

The engineer who does not give flue-gas analysis his attention may some day find himself at the end of a wheel-barrow.

George C. J. Spangenberg.

Buffalo, N. Y.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Connecting Indicator to Cylinder Drip Cocks

Can true indicator diagrams be obtained of an engine by connecting the indicator to the cylinder drip cocks?

E. W. H.

Yes, if the indicator is connected at right angles to the drip connection, or otherwise, so as to prevent impact of the drip water against the indicator piston, and there is thorough drainage of the connections, as by a drip tee placed in the engine side of the indicator to drain the connection and leave unobstructed the full area of steam passage required for proper action of the indicator piston. When all other conditions are constant under which a set of diagrams are taken, if the diagrams are the same, it should be evidence that they have been unimpaired by presence of water in the connections, though such evidence is not entirely conclusive.

Coal Saved by Warmer Feed Water

Where 2400 lb. of coal is used per day for making steam at 100 lb. pressure, which is used for a hoisting engine and for pumping, the temperature of the feed water being 45 deg. F., how much coal should be used for the same work with feed water at 100 deg. F.?

S. T.

A pound of water raised from 32 deg. F. to steam at 100 lb. boiler pressure requires 1188.8 British thermal units. With feed water at 45 deg. F. each pound requires 45 — 32 = 13 B.t.u. less than at 32 deg., i.e., each pound requires 1188.8 — 13 = 1175.8 B.t.u. With feed water at 100 deg. F. each pound would require 100 — 32 = 68 B.t.u. less than at 32 deg., i.e., each pound would require 1188.8 — 68 = 1120.8 B.t.u. Therefore, the amount of coal required with feed water at 100 deg. should be

$$\frac{1120.8}{1175.8} \times 2400 = 2287.736 \text{ lb.}$$

Capacity of Water Tank

The inside dimensions of a circular tank are 12 ft. diameter at the top, 15 ft. diameter at the bottom and 15 ft. deep. What is its capacity?

T. S.

The cubical content of a circular tank of the form described would be found in cubic feet, by adding together the area at the top, the area of the bottom, and four times the area of cross-section taken at the mid-depth; then multiply the sum by the whole depth and take ⅙ of the product. In the example the areas to be added would be:

Area at top = area of 12 ft. diameter = 113.10 sq.ft.

Area at bottom = area of 15 ft. diameter = 176.71 sq.ft.

The mid-section being 13½ ft. diameter four times its area would be 4 × 143.14 = 572.56 sq.ft. and their sum = 862.37 sq.ft.

Then 862.37 × the depth, 15 ÷ 6 = 2155.925 cu.ft., equivalent to 2155.925 × 7.48 = 16,126.319 gal.

Slip of Water Meter

Why should the registration of a water meter in the feed line to a boiler be very much less for a given quantity of water fed through it by a double acting, single-cylinder 4½x2½x6-in. pump than when fed through a meter by a 6x4x6-in. duplex pump?

H. H. B.

Intermittent discharge from the single-cylinder pump would cause most forms of meters to register less than when the same quantity is discharged at a steadier rate of flow as would be the case when feeding the boiler with the duplex pump.

An additional cause for greater slippage of the meter when using the smaller pump may be that it is used at a rate of discharge which is very much lower than the rate of discharge of the duplex pump is lower than the rated capacity of the meter.

Chimney Size and Construction

What should be the size of a chimney suitable for 300-hp. boiler plant using coal as fuel with natural draft? Would vitrified brick be a good material for such a chimney?

T. W.

If the chimney is built 85 ft. high it should have a round flue 64 in. inside diameter or if square 44in.x44in. at the smallest part. Vitrified brick would be good for the outer shell if a mortar is used which will set up hard and with good adhesion to the brick; but the interior of the shell should be protected from heat by an independent core wall of fire brick or ordinary unvitrified brick. The protecting core should be built not less than 8 in. thick for a height of 10 ft., and 4 in. thick an additional 10 ft., and so constructed that it may expand independently of the main wall. For this purpose about ½ in. space should be left for lateral expansion of the core in every direction and there should be plenty of room for independent vertical expansion.

Hoisting Engine Link Motion

Why is it that, when the link is down or in the forward motion, the valve of a hoisting engine should travel farther than when the link is up, or in the backward motion.

A. C. Z.

An ordinary link motion valve gear cannot be so designed that the same travel will take place for forward as for backward motion without sacrificing the equilization of cutoff.

Using Averaging Planimeter

Taking an indicator diagram from an engine with the reducing motion so adjusted that the length of the diagram is 4 in. and applying a direct-reading planimeter, it is found that the engine develops 250-hp. Then changing the reducing motion, but with all other conditions the same as when the first diagram was taken, another diagram is taken and upon applying the planimeter, still adjusted for a diagram 4 in. long, a lower figure than the first is obtained owing to the fact that the second diagram has smaller area than the first. How are the calculations to be made correctly?

K. W. K.

By using a planimeter which measures the area, the mean effective pressure is obtained by dividing the area by the length and multiply by the scale of the spring, i. e.

$$\frac{Area}{Length} \times scale = m.e.p.$$

For the same scale the same m.e.p. could not be obtained, unless the value of the fraction remained constant. If the diagram is 2½ in. long in place of 4 in. long the area would be reduced in the same proportion so that if the adjustment of planimeter is employed which is proper for 4 in. diagrams, multiply the reading by 4 and divide by the actual length of diagram.

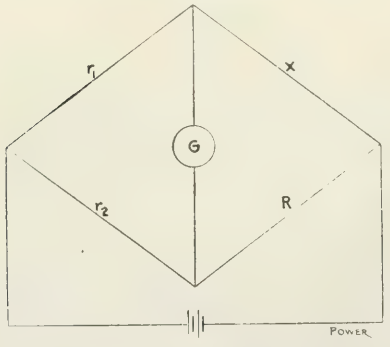
If the planimeter used gives direct readings for different lengths of diagrams then with a shorter arm set for a shorter diagram, the wheel movement will be proportionately greater for the same area.

Study Questions

This Week's Questions
Last Week's Answers

(56) A man painting a steel stack is sitting on a sling seat hung by a single rop running through a pulley at the top of the stack. The free end of the rope is fastened to a ring in the ground 54 ft. from the base of the stack. When the man is 62 ft. from the ground the length of rope from the ring through the pulley to his seat is 100 ft. How high is the the stack?

(57) In the accompanying diagram of the connections for a measurement of resistance by the Wheatstone bridge, the



resistance r_1 is 10 ohms, r_2 100 ohms, and when no current flows through the galvanometer G, the resistance R is 343 ohms. What is the unknown resistance x?

(58) If a steam engine had thermal and mechanical efficiencies of 100 per cent., how much heat would it consume in raising a load of 1 ton to a height of 500 ft.?

(59) A beam 15 ft. long, supported at both ends, has a uniform load of 5000 lb., a concentrated load of 3000 lb. 5 ft. from one end and another of 4000 lb. 5 ft. from the other end. What are the weights on the two supports?

(60) Shot vertically with a velocity of 1000 ft. per second, what height will a cannon ball reach, neglecting air resistance?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(51) The work done per minute is 15,000 × 200 = 3,000,000 ft.-lb. corresponding to

$$\frac{3{,}000{,}000}{33{,}000} = 90.909\ hp.$$

Dividing by the mechanical efficiency gives

$$\frac{90.909}{0.70} = 129.87\ i.hp.$$

In the formula

$$I.hp. = \frac{plan}{33{,}000}$$

or

$$p = \frac{33{,}000 \times i.hp.}{lan}$$

where

p = Mean effective pressure, pound;
l = Length of stroke, feet;
a = Area of piston, square inches;

l is given as 3 ft.; n is twice the 60 r.p.m., or 120; p is to be determined and a the average area of the two sides of the piston can be found from the formula.

$$a = \pi\left(\frac{D^2}{4} - \frac{d^2}{8}\right)$$

where

$\pi = 3.1416$;
D = Diameter of piston;
d = Diameter of piston rod.

Hence

$$a = 3.1416\left(\frac{20^2}{4} - \frac{3.5^2}{8}\right) = 309.35\ sq.in.$$

Then

$$p = \frac{33{,}000 \times 129.87}{3 \times 309.35 \times 120} = 38.48\ lb.$$

(52) The consumption per hour will be

6000 × 1.75 = 10,500 lb.

and for 24 hr.

10,500 × 24 = 252,000 lb.

or

$$\frac{252{,}000}{2000} = 126\ short\ tons$$

or

$$\frac{252{,}000}{2240} = 112.5\ long\ tons$$

(53) The current flowing, by Ohms' law, is equal to the electromotive force divided by the resistance

$$I = \frac{E}{R}$$

As it is the same throughout the circuit

$$I = \frac{50}{300} = \frac{35}{x}$$

$$x = \frac{35 \times 300}{50} = 210\ ohms$$

(54) The volume of a sphere is

$\frac{1}{6}\ \pi D^3$ or $0.5236\ D^3$

where

$\pi = 3.1416$;
D = Diameter of sphere.

Then the volume of the 2-ft. sphere is

$0.5236 \times 2^3 = 4.1888$ cu. ft.

This sphere with 1 cu.ft. added uniformly will become a sphere of a volume of

4.1888 + 1 = 5.1888 cu.ft.

and

$$5.1888 = 0.5236 \times D^3$$

$D^3 = 9.9099$;
$D = 2.148$;

is the diameter of the new sphere. Half the difference of the diameters will be the thickness of the covering or

$$\frac{2.148 - 2}{2} = 0.074\ ft. = 0.888\ in.$$

or closely ⅞ in.

(55) The pressure corresponding to a head of water of 185 ft. is

185 × 0.433 = 80.105 lb.

On a diameter of 20 ft., or 240 in., in 1 in. length of cylinder, the pressure tending to rupture will be

$$\frac{240 \times 80.105}{2} = 9612.6\ lb.$$

This divided by the allowable unit stress gives

$$\frac{9612.6}{16{,}000} = 0.6008\ sq.in.$$

as the required section per inch of length, therefore the thickness for no joints, and for joints of 75 per cent. efficiency

$$\frac{0.6008}{0.75} = 0.801\ in.$$

is the required thickness.

Dallas Plant Enlarged

Dallas, Tex., is rejoicing over the success of its electric power plant. In the past three years the population has grown so rapidly that $600,000 have been spent in enlarging the plant to meet the increased demands.

Chief Engineer J. J. Markham, formerly of Pawtucket, R. I., has practically rebuilt the plant during his administration and has brought it to a high efficiency.

The plant now has 19 boilers, mostly of the water-tube, oil-burning type, having a total of about 8000 hp. General Electric Curtis turbine generating units ars used. The 7500-hp. turbine set recently installed is the largest in Texas. Probably more additions will be required in the next two years.

Loss by Pittsburgh Flood

On July 21, states a press dispatch, a terrific rain storm flooded Pittsburgh manufacturing plants, impaired the trolleys and railroad service and inundated many business houses. The plants of the Westinghouse Air Brake Co., Wilmerding, and the Westinghouse Electric & Manufacturing Co., East Pittsburgh, were flooded by several feet of water. The Westinghouse interests are said to estimate their loss at $175,000.

Electric Plant Purchase Approved

Judge Hollister, of the United States District Court, recently approved the sale by the trustees of the Hamilton Gas & Electric Co., of Hamilton, Ohio, to Frank L. Perin, who represents the majority of the bond holders, for $430,000.

Annual Convention of Ohio Electric Light Association

Cedar Point, one of Ohio's famous watering places, was the scene of the eighteenth annual convention of the Ohio Electric Light Association on July 16 to 19. The attendance was the best ever reaching almost 500, and with only one professional session daily, all had ample opportunity to enjoy the entertainment provided by the committee, the bathing or the attractions at the "point."

The sessions were held on the second floor of the convention hall and a number of interesting exhibits occupied practically all of the first floor. The papers were of practical interest, and a talk on "Rate Making" by Hon. Halford Erickson, chairman Railroad Commission of Wisconsin, was especially appreciated.

Tuesday afternoon at 2 o'clock the first session opened with the usual routine business, including the secretary's report covering the progress made by the society, financial conditions and some recommendations on matters of current interest.

President W. C. Anderson, of the Canton Electric Co., in his address referred to the work of the association and highly complimented the secretary on his efficient and energetic services. The president spoke of the improvements made in the Mazda lamp, increasing the efficiency, and of the new sizes as small as 15 watts with early prospects for a 10-watt lamp. The introduction of these lamps reduces the amount of current used and makes the question of minimum charge for house service of the greatest importance.

Fresh air and the danger of poisoning from carbon monoxide when using natural gas for lighting in poorly ventilated rooms was a topic touched upon with the conclusion that electrical energy was the only illuminant which did not contaminate or impoverish the air. After referring to the process of sterilizing water by means of electricity and the advancement made in electrical heating devices, the president closed with a few remarks on municipal vs. private ownership. In his opinion, the inefficiency and waste of municipal control far exceeded the profit obtained under private ownership.

Committee Reports and Papers

The material presented by the committee on electrical transmission was of a general character, dealing principally with construction data on local transmission systems and lightning protection. Figures on maximum and minimum spacing of conductors for various voltages were given and a good general rule on transmission of 1000 volts per mile of line suggested. The discussion centered on lightning protection by means of multigap and electrolytic arresters and an overhead ground wire. This material will be presented in a later issue.

On Wednesday morning, the Hon. Halford Erickson, chairman of the Railroad Commission of Wisconsin, told why rate regulation was necessary and pointed out the duties of the commission in regulating rates for electric-lignting plants. The rates must be based on a fair valuation of the plant. The cost of reproduction new, less depreciation, rather than original values is now favored by the courts. Mr. Erickson told how the original cost and cost of reproduction were computed and also gave considerable time to depreciation and its determination. In making rates, he considered the demand factor of the greatest importance. There was a big difference between the cost of production for a customer taking energy only one hour per day and another drawing on the supply for 10 or perhaps the full 24 hours. A fair profit over operating expenses and depreciation should be permitted. In other words, the rates should be such as would allow a free flow of capital and executive ability into the field. For electric lighting 8 per cent. on a fair cost value should be a reasonable return on the investment. An animated discussion followed and a great many questions were asked about the relative cost of serving one customer with 1000 kw. and 1000 customers with 1 kw. each, on raising and lowering rates by the commission and on discrimination in rates.

On the subject last mentioned O. P. Gothlin, chairman of the Ohio commission, emphasized the importance of putting honest values into the property. The promoter has no right to give away stock or sell it at nominal values and make the company pay dividends on the "water." In estimating the worth of a property the going value as well as the actual should be taken into account. A great deal of time, ability and money are required to work up the business to a profitable basis.

Secretary D. L. Gaskill followed with a paper on "The Public vs. the Utility," which was full of common sense on the proper relations between the public and the electric-lighting companies. For low rates and good service, protection from unwise and sometimes malicious competition must be assured by the state. The companies will then be able to command the capital necessary and the public-service commission can control and adjust any point of dispute on service or rates.

The committee on Costs, J. D. Lyon, chairman, submitted a report closely related to the two previous papers. All items entering into the valuation of a public utility were enumerated and some little space given to the following subjects: going value, depreciation and diversity factor. The money spent in building up the business and securing new customers should be included in the going value, and conditions determine whether these expenditures should be charged to construction, operating expenses or to cost of plant. The committee urged the importance of securing reliable data on the diversity factor and its effect upon the individual cost of serving. According to the character of the service required or perhaps the time of serving each customer should pay a certain proportion of the fixed charges. What this proportion shall be in each case or in each class of service is the question.

A committee was appointed to investigate the desirability of the society affiliating with the National Electric Light Association. Ohio has the largest state engineering association in the country so that a favorable report by the committee should prove of mutual advantage.

Meter accuracy and testing was the burden of the report by the meter committee. This was on Thursday morning and immediately following a paper on "Electrolytic Purification of Sewage," by Prof. F. C. Caldwell, closed the session.

"Electricity in Rural Districts" was the first paper at the Friday session. The author, J. C. Matthieu, spoke of the opportunities in this field and the problem of educating the farmer to the use of electrical energy for power and lighting. At present rural business does not pay unless the service is taken from a line installed between two urban centers.

In his paper on "Joint Pole Line Construction," J. L. Spore advanced some original ideas on the arrangement of wires on a joint pole line, and cited a particular joint-pole agreement covering all necessary restrictions and regulations governing expenses and maintenance. The discussion disclosed a joint construction-work agreement about to be made standard by the Bell company. An abstract will be included in the proceedings.

Entertainment

One professional session daily gave the visitors time to enjoy the bathing and not miss any of the features provided for their amusement. On Tuesday evening all were guests of the association at a banquet enlivened by popular songs between the courses. Dancing followed, and on the next evening another banquet at which "Walden," the magician, entertained. On Wednesday afternoon a water ball game between the supplymen and the engineers aroused a great deal of interest. The result was a defeat for the supplymen and the same thing happened in the ball game on Thursday. Musicales, card parties, a launch ride and bowling amused the ladies.

Officers Elected

For 1912-13 the following officers were elected: President, J. C. Martin, Wilmington; vice-president, J. D. Lyon, Cincinnati; secretary-treasurer, J. D. Gaskill, Greenville; executive committee, W. C. Anderson, Canton; O. H. Hutchings, Dayton; W. Parsons, Springfield; W. R. Griffen, East Liverpool and W. E. Richards, Toledo. The usual committees were also elected at this time.

Exhibits

Among the exhibitors were the usual companies making electrical products such as lamps, transformers, motors and cooking utensils, and in addition electrically operated washing machines were shown. The list of firms follows: Allis-Chalmers Co., Milwaukee, Wis.; The Brokaw Eden Mfg. Co., St. Louis, Mo.; The Canton Rubber Co., Canton, Ohio; Chicago Fuse Manufacturing Co., Chicago; The Copeman Electric Stove Co., Flint, Mich.; Crocker-Wheeler Co., Ampere, N. J.; The Eclipse Electric Manufacturing Co., Chicago; A. D. W. Electric Sign Co., Cleveland; The Enterprise Electric Co., Warren, Ohio; Fort Wayne Electric Works, Fort Wayne, Ind.; General Electric Co., Schenectady; The John Dietz Mfg. Co., Cincinnati; W. N. Matthews & Bro., St. Louis; Nelite Works of General Electric Co., Cleveland; The Ohio Blower Co., Cleveland; The Packard Electric Co., Warren, Ohio; The Premier Vacuum Cleaner Co., Cleveland; The Sterling Electric Mfg. Co., Warren, Ohio; The A. L. Sykes Mfg. Co., Cincinnati; The Triumph Electric Co., Cincinnati; Wagner Electrical Mfg. Co., St. Louis; Westinghouse Electric Manufacturing Co., Pittsburgh.

Graphite as a Surfacer

Lubrication is the very life of a moving mechanical part. If you neglect lubrication or do it on a hit-and-miss principle, you pay for it by broken parts, shutdowns and other kindred troubles. The road to the junk heap is made an easy one if insufficient or inferior lubricants are used.

No matter how well you may think a piece of machinery is lubricated, unless provision is made for the time when oil or grease fail to do their duty (caused by an obstructed oil channel, overload, leaks, etc.), very serious abrasion is likely to take place and it is then the lubricant which can stand up under adverse conditions, takes its knocks and is not thin skinned, that protects you against losses.

Prof. Goss strikes the keynote when he says, "Tests show that with no other lubrication than graphite the journal and brass may be run together for hours at a time, heating up to a high temperature, and this may be done repeatedly, and yet the rubbing surfaces of the bearings remain unscored. As a protection against abrasion of rubbing surfaces, I know of nothing that can equal the record of graphite."

Boiler Explosions in 1911

A summary of boiler explosions for 1911 in the United States has been carefully compiled by *The Locomotive*. The figures follow:

SUMMARY OF BOILER EXPLOSIONS FOR 1911

Month	Number of Explosions	Persons Killed	Persons Injured	Total of Killed and Injured
Jan.	76	22	43	65
Feb.	44	31	52	83
March	39	27	38	65
April	39	14	22	36
May	43	18	39	57
June	31	30	17	47
July	42	18	34	52
August	32	13	29	42
Sept.	29	7	20	27
Oct.	48	14	53	67
Nov.	39	11	35	46
Dec.	37	17	34	51
Totals...	499	222	416	638

Smoke Problem Lectures

The department of industrial research, University of Pittsburgh, was recently provided with a fund for thoroughly investigating the smoke nuisance. The investigation is being conducted by 25 specialists, seven giving it their entire attention.

Recognizing the interest in the smoke problem manifested by many American cities, the department announces that its staff is prepared to lecture on the following subjects: The smoke nuisance (a general presentation of its main phases); smoke and the public health; smoke and the cost of living; smoke and plant life; methods and means of smoke abatement; the effect of smoke on buildings and building materials; the psychology of smoke; the smoke nuisance and the housekeeper. For further particulars apply to Dr. R. C. Benner, Department of Industrial Research, University of Pittsburgh, Pittsburgh, Penn.

Alternating Current Generators

Bruce Peebles & Co., Ltd., engineers of Edinburgh, have recently issued pamphlet 13B, which gives very complete ratings of the European standard 50- and 25-cycle machines for a range of speeds of from 1000 to 125 r.p.m. at pressures from 200 to 6600 volts. Rules are given for selecting the correct machine for other cycles not listed. The pamphlet also contains full lists of approximate weights and dimensions of alternators with exciters. The Peebles machines are manufactured to meet all special requirements.

Safety Valves

Among the interesting recommendations in the report of the committee on safety valves submitted at the recent convention of the Association of Master Mechanics and Master Car Builders at Atlantic City, is the following:

"Manufacturers should be required to stamp on the valve the lift in inches, as determined by actual test, with valve in working condition and set for a blow-back of not to exceed 3 lb. The committee recognizes the fundamental importance of valve lift. It therefore recommends that in design and manufacture valve lift receive proper consideration."

This opinion is interesting in its bearing on the discussion of this point in safety-valve design before the American Society of Mechanical Engineers. While advanced by those concerned primarily with safety valves for locomotives, what may be said for them applies with equal force to safety valves on stationary boilers.

Bureau of Mines Publications

The Bureau of Mines has the following publications for distribution. Only one copy of each will be sent to a person, and they should be ordered by number and title, addressing the Director of Bureau of Mines, Washington, D. C.

Bulletin No. 41: "Government Coal Purchases under Specifications," with analyses for the fiscal year 1909-10, by G. S. Pope, with a chapter on the fuel-inspection laboratory of the Bureau of Mines, by J. D. Davis.

Technical paper No. 5: "Constituents of Coal Soluble in Phenol," J. C. W. Frazer and E. J. Hoffman. No. 9: "Status of the Gas Producer and of the Internal-Combustion Engine in the Utilization of Fuels," R. H. Fernald. No. 19: "Factor of Safety in Mine Electrical Installations," H. H. Clark. No. 20: "The Slagging type of Gas Producer," with brief report of preliminary tests; C. D. Smith.

Bulletin No. 39: "The Smoke Problem at Boiler Plants," a preliminary report; D. T. Randall; a reprint of United States Geological Survey Bulletin No. 334, revised by S. B. Flagg.

J. Paul Clayton's "New Analysis of the Cylinder Performance of Reciprocating Engines" has been issued as Bulletin No. 58 by the engineering-experiment station of the University of Illinois. The presentation in the bulletin is rather more complete than in the paper submitted to the American Society of Mechanical Engineers, and has appendices, describing in detail the construction of the logarithmic diagram, the experimental plant, the methods of test followed, and a bibliography of the subject. It may be had upon application to W. F. M. Goss, Director, University of Illinois, Urbana, Ill. Price, 60c.

BOOKS RECEIVED

CENTURY LECTURES. American Gas Institute, New York. Cloth; 174 pages 6x9 in.; illustrated; indexed. Price, $2.

THE PRINCIPLES OF HEATING. By William G. Snow. David Williams Co., New York. Cloth; 224 pages, 5¾x9 in.; 60 illustrations; tables; indexed. Price, $2.

ANALYSIS OF METALLURGICAL AND ENGINEERING MATERIALS. By Henry Wysor. The Chemical Publishing Co., Easton, Penn. Cloth; 82 pages, 8½x10½ in.; indexed. Price, $2.

CHEMICAL ARITHMETIC AND CALCULATION OF FURNACE CHARGES. By Regis Chauvenet. J. B. Lippincott Co., Philadelphia, Penn. Cloth; 302 pages, 5¾x9 in.; tables; indexed. Price, $4.

NEW PUBLICATIONS

ASSOCIATION EDUCATIONAL WORK. By George B. Hodge. Published by the Association Press, New York City, 1912. Cloth, 256 pages; 5¼x8 in.; illustrated. Price, $1.

This book is the result of much painstaking effort by Mr. Hodge and represents 20 years of experience in Y. M. C. A. endeavor to organize associations, give men and boys occupational training and decrease the alarming growth of illiteracy in the United States, due to immigration.

Reliable statistics are given to show that there are over 370 different kinds of professions, occupations and trades in America, only a small portion having definite facilities to help men and boys prepare for their life work.

Strong emphasis is laid on the necessity of a closer relation between the shop and the school, and in all the day courses recommended for boys between 14 and 22 years of age half of the time must be spent in the shop. Special attention is directed toward helping a young man to choose that vocation for which he is mentally and physically fitted, and the association claims to have been very successful along this line. "The boy without vocational guidance," says the author, "is father of the bench-warmer and the unemployed."

To those interested in vocational and moral uplift this book will prove valuable in practical suggestions of field study, financial problems, short cuts to educational development and making good citizens of boys who are unable to help themselves. A complete cross-reference index of over 500 topics treated is placed at the end of the book.

A PRACTICAL MANUAL OF STEAM AND HOT WATER HEATING. By Edward R. Pierce. Domestic Engineering Co., Chicago. Cloth; 339 pages; 5½x8½ in.; 30 illustrations, tables. Price, $2.50.

The book is primarily for the practical man. The mathematical treatment is simple, involving only arithmetic. It should be appreciated by steamfitters and practical men engaged in house heating. In the steam tables the older values are used. The book would have been more uptodate with the new Marks and Davis tables.

Historically, the book is interesting for its material on the early forms of radiation, cast-iron boilers and their progenitors. Considerable space is given to the changed conditions of the house-heating business whereby the steamfitter instead of the boiler manufacturer has become responsible for the proper installation of heating systems. Mr. Pierce mentions the fact that cast-iron boiler manufacturers have raised their ratings due to competition and furnish practically no information to enable the steamfitter to select the proper size of boiler for any given condition. He recommends that engineering societies interest themselves in the establishing of a standard for information to be furnished by cast-iron boiler manufacturers in their catalogs. It is proposed that radiation be sold on a guaranteed B.t.u. basis due to the widely ranging efficiency of the different types of cast-iron heating surface. The book is commended to society members interested in standard ratings for cast-iron boilers.

The use of Wolff's rule is recommended for determining the necessary radiation instead of rule-of-thumb methods. The element of air change in this method makes an uncertainty, giving an excess of radiation for large rooms while generally correct for small spaces. The method, however, is recommended in place of an inexperienced guess.

One section is devoted to explaining how the barometrical difference in altitudes would affect the boiling point of water and possibly the boiler capacity. This should have nothing to do with the capacity or boiler output, although steam would be generated earlier and at somewhat lower temperature at higher altitudes. A pressure of 2 lb. at the boiler in the catalog should mean 2-lb. gage pressure with the barometer 29.9 in. and 218.5 deg. F. This is an absolute pressure and temperature within the system whether on Mount Shasta or New York. Some open vapor systems might have to be modified by closing the circuit entirely, but this would not affect the rating or capacity of the boiler.

A table given shows a difference of 10 deg. between the steam temperature of the boiler and the radiation. If this means the steam temperature in the radiator, there would be a drop of 3.2 lb. from 219 deg. to 208 deg. in the radiator and 4 lb. from 227 deg. in the boiler to 215 deg. in the radiator. This would require a fall of 9 ft. between the lowest radiator and the water line of the boiler to prevent flooding, and would be difficult to provide for in many house-heating systems. These low radiator temperatures may be another way of making sure of sufficient radiation. A temperature of 185 deg. is given for the boiler and 180 deg. for the radiator while the items either side show a difference of 10 or more degrees.

The discussion on continuous operation of heating systems is commended to all who may be interested in large or small plants but it applies most forcibly to the former. The author indicates the effect on the heating plant of cooling the masonry walls and the increased cost of intermittent operation as well as the increased capacity necessary.

On another page are three arrangements of heating surface at different temperatures, the highest temperature costing the least, and the author says that the higher pressure will be the most expensive to operate. In view of the statement previously made that pressure is only a measure of intensity and not of capacity, this is not so if the pressure and radiation are designed for zero weather in each case and a boiler of the same capacity is installed. One arrangement will be just as economical as the others in proportion to the ease with which the pressure and temperature of the steam can be changed to meet outside conditions of temperature. If any of the arrangements were maintained at the maximum pressure stated, the rooms would overheat in moderate weather and fuel would be wasted just as ordinarily happens where the maximum is 2 lb. pressure unless automatic heat control is provided. If the boiler was selected by the catalog rating of surface as called for at the higher pressure, it would be too small and Mr. Pierce's statement would be true.

Vapor systems, it is declared, are not well fitted to operate on indirect heating due to low temperatures and, it might be added, large steam volumes as well. As it is generally desirous to heat living rooms with fresh air supply in this class of work the higher pressures will be found more effective.

Section XIV discusses the arrangement of the main return with reference to the water line of the boiler. This is mentioned as Table FF cannot be reconciled with the conditions given.

Another section outlines the information necessary in boiler catalogs so that a proper selection of boiler may be made for any given case. Years ago it was customary to guarantee that heating plants would maintain a room temperature of 70 deg. in zero weather without attention to the fire for eight or ten hours, and nothing was said about the pressure. Table G2 gives some interesting data as to the effect on the rating if this kind of guarantee were required.

Comparing two boilers, one with large flues and small fire surface and one with the reverse, both having the same total heating surface, the author reports tests under like conditions which showed that the second transmitted almost twice as many B.t.u. in a given time. The grate area, amount of fuel and duration of test

are not stated, but would have been interesting so as to determine the relative efficiencies.

The book is written as though the author were conversing with the reader, which relieves the monotony of what otherwise might prove dry reading and maintains the interest. In addition to the useful information and simple instruction, the book contains some pointed cartoons on false boiler ratings. It is to be hoped that the book will have a decided influence in placing bolier ratings on a rational basis.

EFFICIENCY AS A BASIS FOR OPERATION AND WAGES. By Harrington Emerson. Third edition, revised. Published by the Engineering Magazine Co., New York. Cloth; 171 pages; 7½x5 in. Price, $2.

For the most part this is purely a philospohical treatment of the application of efficiency methods in their relation to the various phases of modern industry.

The first half of the book is devoted to the philosophy of efficiency; the remainder, with the exception of the last two chapters, is more specific and deals with the actual work of the line and staff organization; cost reduction, accounting and standards; waste elimination, and efficiency-system operations. Standard time and bonus systems are also well treated.

In the first chapter appears the assertion: "The actual and potential wastes in each year amount to as much as the total accumulation of wealth." This is supported by the statement that by proper management the pig-iron output of one foundry was increased 500 per cent., and that $300,000,000 per year is the cost of preventable wastes in American railway practice.

Americans are severely criticized for lack of "patience, persistence and things enduring." Mr. Emerson is not borne out by history, so far as our lack of the second mentioned quality is concerned, and if he is right in the rest, it is not to be overlooked that it is this very spirit that is responsible for our rapid, though perhaps extravagant progress, and for our international prestige. Our further success depends on our adopting corrective measures to offset the evils of prodigality.

The need of complete parallelism between line and staff is admirably brought out. Power costs, it is said, may possibly be reduced 28 per cent. and power consumption 30 to 33 per cent. This applies to many plants, but by no means to all.

As an example of time losses in labor operations it is stated that: "Standard times may be anywhere from 10 to 90 per cent. less than actual times." The importance of making it pleasant for the worker operating under standardized time is emphasized. Unfortunately the worker is more often regarded as a unit machine whose psychological attributes are an unavoidable impediment to the more rapid increase of profits. This is not the fault of the efficiency officer, but rather the management.

Chapter 10 dwells on the necessity of equity in connection with bonus systems. The inefficiencies of overtime work are well pointed out. The remaining two chapters are philosophical.

The book is well arranged and commendably carries out the author's intention to supply those seeking a knowledge of the fundamentals of "efficiency."

GOOD ENGINEERING LITERATURE; What to Read and How to Write. By Harwood Frost. The Chicago Book Co., Chicago, Ill. Cloth; 422 pages; 5¼x7¾ in. Price, $1.

The purpose of this book is fourfold: to prove the value of good English and literary expression; to show how to collect and arrange engineering subjects for the press; to so aid the engineer in reading literature that his time and money may be profitably spent, and to collect and preserve information which is unattainable to the average engineer.

While Mr. Frost's book is mainly written for the professional engineer, it can be profitably studied by operating engineers who hesitate to write for their trade and technical papers because they fear that their productions are not properly constructed or lack an "elegant flow of language."

The chapter, "What to Write About," urges the contributor to aim to make his contribution "practically useful to his brethren; its merits depend on his success in doing this." The technical paper is "always looking for information on new and better tools, devices and processes, and then makes it public for the general benefit."

Chapters are also devoted to the preparation of manuscripts for publication, grammar, punctuation, card indexing and filing, and the engineer's library.

Principe di Piedmonte Explosion

Contrary to some reports, the accident on the Lloyd Sabaudo liner "Principe di Piedmonte" while at sea on July 17, in which five men were killed, was not due to an exploded boiler tube or valve, but to a bursted supply pipe from a boiler to the header.

The five boilers are connected to the header by bends of 6-in. copper pipe. The break occurred in the copper pipe at the flange connection to the header, and between the header and the stop valve. A close examination of the break showed it to be fresh and more clean than jagged.

Fourteen men were in the boiler room at the time, nine being at one end near the engine room door. The remaining five, who were killed, were trapped between the affected pipe and the door, and as the steam rushed out of all five boilers they were immediately overcome.

No cause is assigned for the break. Nothing is said to have hit the pipe and, unless something was radically wrong, water-hammer could not have caused it.

Those killed were the first engineer, the chief fireman and three stokers. It was the chief fireman's first trip on the vessel, he having transferred to her in order to visit his son in New York.

The accident is identical to that of the "North Dakota," which burst a 7-in. pipe during an endurance test in 1909. Editorial reference is made to the accident on page 160.

The Ruppert Accident

In the haste of going to press with the report of the accident at the plant of the Jacob Ruppert brewery, New York City, in the issue of July 16, proper verification of the facts was omitted. It appears that the report as given contained certain inaccuracies.

Instead of a cylinder head being blown out of the ammonia end of one of the compressors, four bypass valves were blown out. Soon after the escape of the ammonia fumes an explosion occurred which killed one man instantly and afterward resulted in the death of two others.

The direct cause of the explosion or the bursting of the bypass valves is not known. It is thought that the ammonia mixed with the air to form an inflammable gas which was ignited by the flaming arc lamp in the engine room. Although damage to the woodwork and electrical fittings was quite serious, the plant was again put in operation soon after the ammonia fumes cleared away.

Electric Shock Fatal

Attempting to fight a fire in the tower of the City Hall, Atlantic City, on July 17, Capt. Edward Barnett, of the fire department, was instantly killed when a lantern in his hand completed a high-tension electric circuit. Chief George Profatt, of the City Electric Bureau, was knocked unconscious in an effort to rescue Barnett, and several firemen were hurled from the tower to a landing 15 ft. below when the metal trimmings of the tower on which they were standing became grounded.

Elevator in 70-ft. Drop Seriously Injures Six Men

Six men were seriously hurt when the cable of a freight elevator in the Puck Bldg., Lafayette and East Houston Sts., New York City, snapped and dropped the car from the third floor to the subcellar 70 ft. below. The safeties were evidently not in working order.

Engineer Killed by Escaping Steam

A serious boiler accident recently occurred at the plant of the Union Brewing Co., Nanaimo, B. C. Preparatory to removing the stays from the dome of one of two boilers connected together by a line containing a slip expansion joint, the bolts were removed from the flange connection of the line at the idle boiler. As the other boiler was under full pressure, the expansion joint, which evidently was not provided with studs and checknuts, let go as soon as the last bolt was removed. The escaping steam overcame and killed Albert Randle, one of the assistant engineers. Editorial reference is made to this accident on page 159.

New River Development Inspection

General Manager H. W. Fuller, of the Appalachian Power Co., Bluefield, W. Va., was host recently to over a hundred visitors on a trip to developments Nos. 2 and 4 of the company on New River. A special train left Bluefield early in the morning and returned in the evening. The party was taken over the developments and the work described.

It was announced that development No. 4 will be ready to furnish power, Aug. 1, and development No. 2, Oct. 1. The former will have a capacity of 9000 hp., and the latter 20,000 hp. A handsome booklet giving the company's history and photographs and statistics showing the progress of the work to date was given each guest as a souvenir.

N. A. S. E. Convention Program

The local committee of the National Association of Stationary Engineers, which has charge of the arrangements for the coming national convention, to be held in Kansas City, Mo., during the week of Sept. 9, has prepared an excellent program, and unless all signs fail, this year's convention will be particularly notable.

Monday, Sept. 9, will be given over to the usual receptions of delegates, members, visitors and their ladies, registration and the distribution of badges at the headquarters, the Coates House, where the local Ladies' Auxiliary will tender an informal reception to the visiting ladies. In the evening the national officers will receive the delegates and visitors in the hotel parlors; an informal musical program will be offered.

At 9:30 a.m., Tuesday, W. F. Groom, chairman of the local committee, will call the convention to order in the New Casino Hall, at 1023 Broadway. An imposing list of speakers has been prepared for this session. Governor Herbert S. Hadley, of Missouri, will give the address of welcome and National President E. H. Kearney will respond. Governor W. R. Stubbs will welcome the visitors to Missouri's sister state, Kansas, and National Vice-President J. F. McGrath will then respond. Mayor Henry L. Jost, of Kansas City, Mo., will welcome the delegates to the convention city, Past National President Joseph F. Carney replying. John H. Atwood will speak on "The West" and Past President Carl S. Pearse will respond. At high noon the convention will be officially opened by National President Kearney, who will appoint the various committees and make the formal announcements. The official photograph will then be taken at the south front of the convention hall. At 12:30, the mechanical exhibit in Convention Hall, Thirteenth and Central Sts., will be formally opened by National Vice-president McGrath. At 2 p.m., an official session will be held, and in the evening the visiting delegates and members will visit the city's fraternal organizations.

Wednesday's official session will be held at 9 a.m. At 1:30, special cars will leave the Coates Hotel for Electric Park, where a complimentary entertainment will be given the visitors during the afternoon and evening by the owners, J. J. and M. G. Heim, under the personal supervision of C. H. Mann. Admission will be given only to wearers of the official badge and their families. The annual baseball game between the engineers and exhibitors for a silver cup will be played at the Gorden Koppel stadium, near the park, at 3 p.m.

On Thursday, the official session will be held at 9 a.m., followed by the annual memorial services at 11:30. At 2 p.m. the fourth session will be held. In the evening the usual entertainment by the National Exhibitors' Association will be given in Casino Hall.

On Friday there will be two business sessions and the election of officers. In the evening the officers will be installed, followed by a grand ball in New Casino Hall.

On Saturday the annual meeting of the officers will be held at the Coates House. Visitors will be escorted to the various places of interest about the city upon request to the information committee.

During the convention the Ladies' Auxiliary will hold several sessions at the Coates House, interspersed with an attractive program of entertainment, including an automobile ride on Thursday at 2:30 p.m. Automobiles will leave the hotel at 2:30, proceeding to the Swope Park Zoo by way of Penn Valley Park and sections of Kansas City's splendid boulevard and parkway system. The return route will be along the famous Cliff Drive.

National Association of Cotton Manufacturers Meeting

The ninety-third meeting of the National Association of Cotton Manufacturers is to be held at the Griswold, Eastern Point, New London, Conn., beginning on the evening of Sept. 9, 1912, and continuing through the two following days.

The delightful scenery of this location and the attractions of the house are such that an attendance of over 400 is expected.

The program of the meeting is in an advanced state of preparation and will not be issued until shortly before the meeting. The association desires from the members further papers on subjects connected with the cotton manufacture or tributary to it.

SOCIETY NOTES

The second annual meeting of the Institute of Operating Engineers will be held on Friday and Saturday, Sept. 6 and 7 in the Engineering Societies Building, 29 West Thirty-ninth St., New York. Friday morning will be devoted to the reunion of members and friends and the reports of committees. The main features will be the award of diplomas on Friday evening and the report of the Constitution Revision Committee on Saturday. Friday afternoon will be devoted to the reading and discussion of papers.

PERSONAL

President R. C. Maclaurin, of Massachusetts Institute of Technology, has sailed for Europe to spend his vacation largely in inspecting the buildings and equipment of technical institutions at Birmingham, Manchester, Glasgow, Munich, Charlottenburg and elsewhere, preparatory to the building of the "New Technology by the Charles."

It is not unlikely that Joseph McNeil may resume his connection with the Massachusetts Board of Boiler Rules, but in a different capacity. Robert J. Dunkle, who represented the boiler-insurance interests upon the Board, is connected with the Casualty Company of America, which company has given up boiler and flywheel insurance. This will probably involve the substitution for Mr. Dunkle of the representative of a company which is actively interested in that branch, and it is entirely likely that Mr. McNeil, who resigned as chairman of the Board of Rules to accept the chief inspectorship of the Eastern district for the Hartford Steam Boiler Inspection & Insurance Co. may be the man selected.

Moments with the Ad. Editor

Some weeks ago we asked readers to tell us something of their experiences with Power advertisers.

We wanted to know *definitely* what they had bought or recommended the buying of that was installed in their plants and that was first called to their attention through Power.

We offered to pay for the letters published and here is number one, from A. R. Kummerer, of Hazleton, Pa.:

"Correct advertising is the best way for the manufacturer to put his goods before his public. For these reasons:

"First—By advertising in a paper like Power with a good live circulation, he can state the merits of his goods to thousands of people who are looking for things to save them money and who would not find them with the goods themselves stuck away on the shelves of supply houses. He can do this without the loss of time and without the inconvenience required to go the supply house. And in *one issue* of Power.

"Second—Seeing goods advertised continuously sets a fellow to thinking that there must be more than talk back of the goods or they could not stay because papers with live circulations don't carry advertising free of charge.

"Third—The illustrations of advertised articles coupled with the manufacturers' generous offers of catalogs make it convenient for the engineer to compare the articles with the conditions in his plant.

"I have purchased Cling Surface Belt Dressing, Schieren Duxbak Belting, Manzel Bros. Oil Pump, Keystone Grease, Smooth-On Iron Cement, McCord M'f'g Co., Gaskets, National Feed Water Heater, Kewanee Unions of National Tube Co. Sheet Rubber of Quaker City Rubber Co., and have an order placed with the Coppus Engineering Co., for a turbo blower of their make.

"Personally, I have purchased books on engineering work, namely, Engine Room Chemistry and Steam Turbine Practice and will continue to buy such books as are of value to me.

"I never omit looking over the advertising matter in Power and often pick up old issues and look over the ads to make sure that I have not missed something that would be of value and service to me."

Now here is *your* opportunity to tell us something about *your* experiences.

Every letter published is paid for in real money, as are all contributions to Power. Letters on this page command from $2.00 to $5.00.

For several years the ad. editor has been preaching that it pays to read advertisements and buy advertised goods.

Somewhere back in the dust another power-plant paper has come out of the tall grass and is following in father's footsteps.

It has made a very careful study of father's language, too, so that if past performance argues for future action, it will, within the next few weeks, ask *its* readers this self-same question—What are you getting out of advertising in *that* paper?

Now, since the ad. editor holds no patent-right, copyright or moral right to the English language, all he can do is to ask the question *first* and then leave it to Power readers to make a *quick* response.

If you will tell him *now* what you have gotten out of Power advertising you will be paid for every line printed.

Let us know definitely what you have bought or influenced the buying of for your plant that was first brought to your notice through Power.

Address the Ad. Editor, Power, 505 Pearl Street, New York.

POWER

Vol. 36 NEW YORK, AUGUST 6, 1912 No. 6

MR. EMPLOYER, have you been down into your engine room lately?

Do you know what the temperature is there these hot days?

Did it ever occur to you that the men who must live and work—do hard physical work—in that temperature, must suffer?

Think, the next time that you are sweltering and gasping because it is 90 deg. in your parlor car or library, of the poor fellows who work for you twelve hours or more every day in a cellar where it is 110 deg. or more.

You wouldn't condemn men—your men—to such discomfort unnecessarily, would you?

Have you done all that you can to better their condition? You cannot put the engine and boilers on the roof, but have you done what you can, in reason, to make the place where they must be, livable? Are the heat radiating surfaces covered as thoroughly as they might be? Are there ample ducts to supply cool fresh air? Are there means for taking off the air which has been heated by contact with and radiation from hot surfaces?

A few connections from the top of such a room to the ash pits and exhaust fan will pull out the heated air, reduce the temperature a number of degrees and increase the comfort in a greater proportion. A desk motor fan such as you use yourself would help some.

Drop down and see how your power plant men are standing the hot spell; how long you would stand it in their places; and what you can do to make it easier for them.

Prize Turbine Article

Running a Steam Turbine

By Paul Hoffman

In the last issue of 1911 POWER offered a prize of $50 in gold for the best article on the practical operation of a steam turbine. Of the 46 manuscripts received in response to this offer this was unanimously chosen by the judges as the best. Others of the articles, treating the operations more in detail and with illustrations, will be published in later issues.

The repeated call for contributions to the contest which the editors of POWER proposed to their subscribers, on running a steam turbine, encourages me, although rather a stranger to American practice and in no way proficient in English composition, to submit a few ideas on the subject.

My experience in operating steam turbines has been largely in Germany, where these engines are in high favor; as American types, however, do not differ in any essential feature from their European relatives, the following remarks will apply to them as well:

To begin with I must apologize for finding some fault with the statement of the question, as it was put before the readers of POWER: How can a man, *who never saw a turbine except in a picture book*, be led alongside such a machine and told how to run and to attend it?

Frankly, I would not like such a man to experiment with a turbine, and, in fact, I believe that, except in cases of emergency, this practice will not be followed by any turbine owner. Before taking charge of a turbine the novice engineer—although he may be a perfect reciprocating man, should be thoroughly instructed as to the necessary manipulations by the contractor's erecting man, if a new installation is concerned, or by his predecessor if he is to take care of an existing plant.

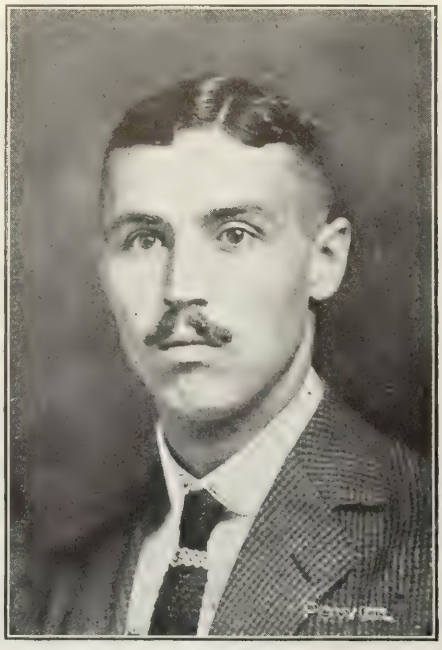
It would be of great assistance if every builder of turbines would furnish complete and definite instructions with each engine erected and give some practical lessons to the man who is to operate it. This policy has been adopted with excellent results in Germany; it does not yet seem to be general in this country. Do we ask too much if we want our new engineer to have also some inside information about his machine, and would it not be desirable that he had a certain knowledge of the fundamental principles of a turbine?

POWER itself has set a very high standard for the engineering profession. According to its views, and in order to be more than a mere stopper and starter, every uptodate turbine engineer should aspire to understand the theoretical side of his machine too. He should possess more than the fundamentals of heat and thermo-dynamics; he should know something about the conversion of pressure into velocity, and he ought to be acquainted with the ways of reaching a compromise between the tremendous velocity of spouting steam and the demands of the driven machinery, the methods of pressure- and velocity-staging and the combination of the two, the relation between speed and efficiency, and a dozen items more.

The means of knowledge are close at hand; a few good books and engineering journals will furnish the required theoretical information.

All his science, it has been objected, will entirely fail to make him a successful engineer. Science alone, indeed; for the most important part of engineering practice is empirical. But knowledge added to experience and observation will help a good deal to answer all the whys and wherefores which may be encountered in a turbine plant.

Now, the preceding considerations by no means imply that it takes a specially educated or experienced man to run a turbine, as long as everything goes right. On the contrary, the turbine builders have so successfully endeavored to simplify the task of attendance, to make everything mechanical, automatic and foolproof, that the job may seem easy. But, let any trouble arise, and the difference between the ignorance and lack of judgment of the mere engine-driver and the resourcefulness and competence of the "man who knows" will come into evidence. A turbine is a piece of fine machinery which should receive intelligent attention; the higher the class of machine the more this is true.



PAUL HOFFMAN

Paul Hoffman, the winner of the turbine article prize, was born in 1882. He has been engaged in engineering since the age of 17 and for about 8 years in steam turbine work. His theoretical training was at the Polytechnicum, Zurich, Switzerland, with Professor Stodola, and his practical experience was obtained in the shops of the Elsassische Maschinenbau, Mülhausen, Alsace; Escher-Wyss, Zurich, and in the German Navy.

For the last three years Mr. Hoffman has specialized in turbo-compressors and took part in the designing of the first Zoelly turbo-compressors.

He came to this country one year ago and is now with the Ingersoll-Rand Co., at Phillipsburg, New Jersey.

SMALL TURBINES

The turbines built and operated in this country vary largely in size and design. Their principal use, up to now, has been to generate electric power. In recent times the field for turbines, and for small turbines especially, has much enlarged; they have been applied as prime movers to all sorts of auxiliary machinery, to pumps, blowers, compressors. The small-capacity turbines used also for excitation service, for light and power purposes of small plants, up to about 200 kw., constitute a class by themselves. They are designed on different principles than the large units; simplicity, ruggedness and reliability are the factors first considered in their construction.

Built with a single wheel, of the pure impulse type, as in the De Laval turbines, or with velocity stages as in the Kerr, Terry, etc., turbines with the governor acting directly on the throttling valve, with all attached mechanism reduced to their simplest expression, have nothing to get out of order and their operation needs a minimum of care. A man looks at them once or twice a watch, and puts his hands on the bearings; oil is filled in once a month, and they keep

running for years without interruption, without repairs, with practically no maintenance cost. This is the reason of their success, in spite of their lower efficiency, compared with reciprocating engines.

Large Turbines

From about 400 kw. upward, however, the turbine enters the competition with the reciprocating engine on a new basis, for it shows, with increasing capacity, an advantage in economy, which becomes quite conspicuous in the very large units installed in recent times in different power stations. Large turbines are built essentially on the principle of high efficiency and their design and mechanical construction is accordingly more elaborate and complicated.

The present remarks will apply to this class of turbines only, as requiring naturally a more careful attendance and being subject to more disturbances.

The three well known manufacturers of large turbines in the United States, the General Electric, Westinghouse and Allis-Chalmers companies, have developed two different types of turbines.

The General Electric turbine is an impulse turbine, with a reduced number of velocity-stage, or so-called Curtis wheels. The Westinghouse and Allis-Chalmers types are based upon the reaction principle (Parsons) with multiple pressure stages. Besides these, the impulse or action turbine with multiple pressure stages, as of the Rateau or Zoelly type, has been constructed, but is as yet of limited, though increasing importance.

Clearance

Without discussing the design of any of these turbines let us only point to a few features, common to all, which will particularly strike the converted reciprocating engineer. Among the conflicting stories which he may have heard about steam turbines, a special emphasis is laid on the tremendous speed of the rotating parts and great stress is put upon the disastrous effects of an encounter between blades traveling with a velocity nearly that of a cannon-ball, and any obstacle. A certain uneasiness on this behalf may seem warranted, for it must be conceded that, until recently, the clearances between moving and stationary blades were often made too small, so that the least vibration of the shaft would cause both parts to touch, often resulting in blades being torn off, and wheels and shaft distorted.

With regard to this point, impulse and reaction turbines show somewhat different requirements. Whereas the radial clearance in impulse turbines is practically unlimited and the axial distance can be made 0.1 to 0.2 of an inch and more without appreciable loss in efficiency, the reaction turbine demands reducing the radial clearances to the lowest practical amount. The blades need not run close together, but, to prevent excessive leakage over their tops, the distance between the ends of the moving blades and the casing and between the ends of the fixed blades and the rotor, must be kept down to 0.025 to 0.125 in., according to the size of the machine and the length of the blades. These figures demand evidently the highest grade of workmanship on the part of the builder and an accurate balancing of the rotor. The arrangement of the dummy rings on the balancing pistons adds to the difficulty. At this point, too, most careful adjustment is required. To prevent steam leakage to the condenser the projecting rings turned out in the rotor must run as closely as possible to the corresponding strips in the casing, the allowed clearance being about 0.01 of an inch.

The axial adjustment is controlled by the thrust bearing which can be moved in either direction by bolts acting on the casing. This adjustment being made with minute care in the erecting shop, it is advisable for the operating engineer to leave it undisturbed, except in case of necessity. After the machine has been taken apart and put together again, a new adjustment must be made. This will be done with the machine slowly running. The thrust bearing is moved one way until a slight contact is noticed (the noise will be more easily detected by a proper piece of wood applied to the casing and the ear) then brought back the desired amount of clearance.

The dummy rings and balancing pistons can be dispensed with in double-flow turbines (Westinghouse) and in impulse turbines. For the latter type the leakage losses take place through the clearance left between the shaft and the diaphragms, which are fixed disks bearing the nozzles. A labyrinth groove packing is generally provided at this point and requires a narrow radial fit of similar accuracy to those above mentioned.

The steam consumption of a turbine depends largely upon the maintenance of the initial clearances; the greatest care must be taken to avoid their reduction or destruction, as by vibration.

Gland Packings

A few types of turbines, especially the original Parsons turbine, apply the principle of the labyrinth to the stuffing-boxes, too. This has the advantage that friction losses are absolutely avoided, at the cost, it is true, of a small steam loss.

The leading firms in this country have adopted devices of another kind; the General Electric Co. uses a carbon packing, the Westinghouse and Allis-Chalmers companies a water-gland packing. With the former the engineer will have to remember that the carbon rings should have a loose fit on the cold shaft; as carbon does not expand but contracts through heat, the expansion of the shaft would open the segments of an exactly fitting ring when warming up, and leakage would occur. The carbon rings do not need lubricating, but are likely to break and wear out. In the long run they have been seen to cut regular grooves in the shaft; the latter, therefore, often has an interchangeable bushing.

The water-gland packing insures a perfect closing of the turbine, but the runner, forming a little centrifugal pump which forces the water in a circumferential sealing space, absorbs a certain amount of power. Besides, if the water is not perfectly clean, it will deposit scale in the glands which has to be removed, sometimes frequently. Finally the water which passes into the turbine will evaporate there, and may cause the formation of a considerable layer of scale on the blades, with a very detrimental effect on the steam consumption. The water supply must be carefully regulated to keep the leaking quantity as small as possible. To avoid evaporation in the glands the temperature should be the lowest obtainable. The water should come to the glands with a few pounds pressure.

Bearings

The next important parts to be considered are the bearings. They constitute with a few gears (worm-gear for governor shaft, tachometer drive, oil-pump gear) the only parts of the turbine where actual metallic friction takes place. As no other acting forces are transmitted by the shaft besides its own weight, the wear of the babbitt shells will be extremely little. The bearings are flooded with oil under pressure; this enters between the babbitt and spindle and forms a continuous film, acting like an elastic cushion. The oil pressure is furnished either by a special oil pump driven from the turbine shaft, or by gravity. Both systems have their merits. The self-contained system is independent of outside help; the oil is stored in a compartment of the bedplate, is thence pumped through a cooling coil in an adjacent cooling chamber of the bedplate, and conveyed to the bearings by a piping system. Once the oil-regulating valves on the bearings are adjusted, they need not be touched again, the oil supply being an automatic function of the turbine shaft's revolution. A disadvantage of this system is the necessity for an auxiliary steam- or hand-driven oil pump, as the main pump does not deliver enough pressure right from the start.

Oil pumps of various designs have been used; plungers, gears, centrifugal or helicoidal, any of these constructions can be depended upon. For gear and helicoidal pumps the decrease of delivery due to the wear must be considered.

In a gravity system the oil is supplied to the bearings from an elevated tank to which it is returned after use by a separate oil pump. The circuit in this case is longer, and the stored quantity of oil can be much larger. This has been found by experience to be favorable, the deterioration of the oil being less rapid. It is claimed as another advantage that the oil-piping system is more accessible for inspection and repair.

With either system the oil consumption of the bearings should be correctly adjusted, so that the rise of temperature between inlet and outlet will be approximately the same for all bearings.

The Governor

A part of great importance in every turbine is the governing mechanism. The governor is of the centrifugal spring-weight type, and either attached directly to the turbine shaft or mounted on a special shaft, driven from the turbine shaft by cut gears (worm-gear). The gear works in an oil bath.

The governor has to move only a small pilot valve which regulates the admission of oil (under pressure) to the relay —or servomotor cylinder, the piston of which acts either directly on the steam valves, main and bypass, or determines the motion of the mechanical valve-gear, as in the General Electric turbine.

This method of oil-relay regulation is most sensitive; quite sudden load changes have only a slight reaction on the speed of the machine, and, with full load thrown off, the increase in the number of revolutions is ordinarily guaranteed not to exceed 2 or 3 per cent.

An important feature is that the oil supply to the bearings and to the governor are interconnected so that the relay-piston will automatically shut off the steam if the oil supply fails and endangers the bearings.

In case of any accidental derangement of the main governor mechanism, a separate safety-stop or overspeed governor, mounted directly on the shaft, comes into action and instantly shuts off the steam supply, whenever the speed exceeds a predetermined number of revolutions (say, for instance, 8 to 12 per cent. over the normal).

Attendance

Obviously, all precautionary measures have been taken to make the turbine the most reliable prime mover, and the easiest to handle. For simplicity of attendance, low cost of maintenance and repairs the turbine is far ahead of the reciprocating engine. There are, in particular, no use for the oil can, no filling of oil cups, no oil dripping and constant wiping, as on the engine. There are no reboring of cylinders, no setting of valves, no readjustment of worn out parts, loose pins and journals and so on. There is practically no packing, as, apart from the main stuffing-boxes, no glands whatever are used, valve stems, etc., being ground in their bushings.

Accidents

Stunts of various kinds and breakdowns have happened to turbines in the past and will happen in the future, but accidents with heavy loss of property and life, as they occur time and time again with reciprocating engines (flywheel explosions) happen very seldom with turbines. One could easily sum up the few cases of turbine explosions due to runaways, where wheels or rotors have burst on account of excessive velocity, and water-hammer is an unknown danger in steam turbines.

Nobody can, evidently, give an absolute guarantee that occasional material defects will not cause a blade to break and cause some mischief in the machine, but such an event, if it is not unavoidable, is at least no extreme misfortune. A row or two of the damaged blades are quickly taken out, and if they cannot be replaced at once, they may be left out and the machine run without them until repairs can be made.

Starting a Turbine

A turbine to be started for the first time, or after a long stop, should be cleaned and inspected with special care. This applies chiefly to the bearings, the oil pipes and oil reservoirs, where the least particle of dirt or grit is harmful. The steamways and piping should be gone over, too, very carefully. Instances where bolts, nuts, waste and other articles have been found on such inspection in the casing, the steam channels or valves, are not at all uncommon. The steam pipe should be blown out from the boilers, the last connection to the turbine being removed. All kinds of dirt, chips of gaskets, red lead, etc., will appear at this occasion, which would have obstructed the steam passages (guide vanes) if not removed.

The oil should be strained through cloths when poured into the machine. A sufficient quantity must be provided. The air should be driven out of the oil piping by operating the auxiliary oil pump. The bearings will have to be generously oiled before starting.

Revolving the shaft by hand will show if there is any hindrance to its free rotation. This preliminary work being done, the warming up of the turbine can begin. Like a steam engine a turbine needs a thorough warming up before being loaded. This prevents unequal expansion of rotating and fixed parts, which would follow a sudden exposure to steam. To begin with, open the throttle valve just a little and increase gradually; open the drains and leave them open until the turbine is nearly full speed.

The time required for warming up depends upon the type and size of turbine. Large turbines need more time than small ones. Reaction turbines require a more careful treatment on account of the smaller clearances and the large mass of the rotor. One-half to two hours is not too much for these machines. For impulse turbines half this time may be sufficient. In an emergency, when a turbine must take up the load of a broken-down unit as quickly as possible, the time for warming up may be considerably shortened. A practice followed in Germany, is to start the turbine running slowly from the beginning, the warming up going on while the speed is gradually increased.

During the time the turbine is warming up, the auxiliary machinery can be got under way. Where the turbine has water glands, the vacuum can only be created and the air pump started after the turbine is at a certain speed, because the glands do not act before a certain centrifugal force is developed and until then a great quantity of air can pass through the stuffing-boxes into the condenser. If a central condenser is used, such a turbine has to be put up to speed with the valve between the turbine and the condenser closed, exhausting in the meantime to the atmosphere.

A turbine should always be speeded up slowly. If accelerated too quickly, vibrating may start quite unexpectedly and continue a good while after starting. The passage of the critical speed, if a flexible shaft is used, is an exception to the above rule; it has to be done promptly, as in that range of revolution vibrations have a particular tendency to develop and to increase dangerously.

After normal speed is reached, with a good vacuum established, the load should be put on by steps, to avoid a sudden inrush of steam into the turbine, which will very likely be accompanied by a water slug.

If the turbine is to run in parallel with other units the governor can be adjusted for variable speed while the turbine is in operation, thereby facilitating the synchronizing of alternators.

It is helpful to always follow the same routine of duties when a turbine is started or stopped; for instance, the operation of the auxiliary oil pump has to be combined with the observation of the pressure gage which indicates the moment when the main pump begins to work; the moment for opening the water-supply valve to the cooling coil must be delayed until the oil is warmed up a little and becomes more fluid, etc.

Operation

Although the turbine takes care of itself automatically, so to speak, and intelligently, it must receive due attention.

At regular intervals, the engineer in charge of a watch should go over his machine, observe the pressure and vacuum gages, the tachometer, measure the

temperatures of the bearings and of the oil, and inspect the water supply to the cooling coil, the packing glands or the water-cooled bearings, if there are any.

An equal share of supervision should be given to the condensing plant, and especially should any drop of the vacuum or irregular behavior of the vacuum gage incite to a thorough investigation.

The sources of trouble in a turbine can often be located in the oiling system. It should be seen to that the cooling coil remains effective, that deposits of mud through the water, or of gum or vaseline-like substances on the side of the oil, be removed. The water used for cooling and other purposes should be clean; if necessary it should be filtered or treated against scale-forming elements.

The oil should be of the best quality; the cheaper grades are always more costly in the end, they deteriorate more rapidly and besides cause trouble. Only pure mineral oil of not too high viscosity —a light-bodied oil—is to be used. Acids in the oil are strictly prohibitive. Admixture of vegetable oil gives a tendency to gum; the hydrocarbons present in mineral oils, when mixed with vegetable oil separate out as precipitate and cause trouble by clogging.

The temperature of the oil at the outlet of the bearings should be about 140 deg. F.; not much higher, for fear of heating, nor much lower, for the sake of fluidity.

If a bearing gets hot it can be easily smelled and seen by the dense white fumes of the burning oil. There is generally some radical cause for a hot bearing, which should immediately be removed—a tight cap, grit or foreign substances in the oil, insufficient oiling. The maximum oil supply should be tried, and if this gives no relief the turbine should be stopped at once rather than have the bearing burn out. Then cleaning and inspecting should follow. It is to be remembered that quick action is necessary; the high circumferential speed of the journal does not allow much chance for reflection or hesitation, and there is no time for nursing the bearing back into good condition. Troubles with the bearings are, however, rare if the oil is carefully looked after, filtered frequently and the strainer cleaned.

Vibration

Any other, and especially internal diseases of a turbine, are indicated by one sure symptom—vibrating of the machine. The vibrations tell the sensitive expert a whole story about the complaints of his engine; they constitute, in power-house language, the particular mode of speaking of the turbine, less expressive perhaps than the pounding and knocking of a reciprocating engine, but always a certain sign that something is wrong.

The diagnosis of the evil is not always easy and the causes of vibration may be various. First, it is never possible to balance the parts of a rotating mass so as to make the center of gravity coincide exactly with the geometric center about which it revolves; hence vibration. At certain definite speeds, called critical speeds, this vibration becomes excessive, while below and above, the motion may be perfectly smooth, as long, at least, as the unbalanced masses are small.

A considerable lack of balance causes vibration at any speed, and is the principal, if not the only, cause of all vibration. It has been known to result from the irregular expansion of the rotor due to quick warming up, but every sudden change (especially rise) in temperature may equally provoke distortion of the wheels or a warping of the spindle. When superheated steam is dealt with this point should always be observed. A change from condensing operation to atmospheric exhaust calls for attention for this same reason, and should not be rushed.

On the other hand, water drawn into the turbine with the steam will act as a very effective unbalancing mass and mostly with very bad consequences; and very obviously a blade flying off will create an overweight on the opposite side of the rotor without regard to the other effects.

Vibration due to any one of these causes always has the same effect; it brings the revolving and stationary parts of the labyrinths into contact, with friction, heat and deformation as results, and the least that will happen will be the destruction of the small clearances, and increased leakage and steam consumption.

It is therefore always advisable, when a turbine begins to vibrate for some unknown reason, to moderate the speed at once and to give the rotor an opportunity to get into good shape again. If the vibration persists after a new start, and goes on increasing, the trouble is probably of a serious nature and should be looked after.

Sometimes it is difficult to locate exactly the source of vibration, as it may be due also to the driven machine. Or each one of the two machines may individually be well balanced, but if both shafts are not truly aligned they will never work properly together and permanent vibration will result, especially if a rigid coupling is used. For this reason preference is given to elastic couplings, which will accommodate themselves to a slight lack of alignment without affecting the behavior of the whole machine.

A few rules as to desirable general operating conditions might here be touched upon.

Superheated Steam

If at all possible, turbines should be supplied with superheated steam. Conclusive tests have shown its advantageous influence on the water-rate efficiency, but there are, besides, a few practical reasons which must be considered. In the first place the probability of getting water into the turbine becomes much smaller, and further, superheated steam causes far less erosion of the blades than wet steam. The cost of adding superheaters to an existing plant should not be regarded if this improvement is possible; the money spent will prove a good investment. Saturated steam should be controlled in its degree of moisture; forcing boilers increases this amount and is not good practice.

Vacuum

Another important point in the line of economy is maintaining a high vacuum. The ability of the turbine to make use of the expansion up to the last limit is sufficiently known, and thermally spoken, it is the last inch which gives the largest gain. Evidently the question is to be considered from the standpoint of the condensing plant, too, for at a certain limit the surplus work for the water-circulating pump overweighs the gain of the turbine.

In turbine plants large dimensioned condensers are usually provided. With either surface, barometric or jet condensers, a vacuum of 27 to 28 in. can easily be maintained and should be. The water temperature is here the principal factor, and other facts might have to be considered, which would require a detailed treatment. Running the condensing apparatus properly is a task which takes generally more of the engineer's time and attention than the turbine itself. Before leaving this subject the importance of frequently inspecting relief valves may be mentioned.

A last rule for the benefit of economy: reduce the amount of idle running of a turbine working in parallel with other units. Care should always be taken to handle the load so that no turbine runs for any length of time at much less than half load.

Shutting Down

When an independent turbo-generator is to be shut down, the throttle valve can be closed before the load is taken off; the latter acts as a brake and brings the turbine quickly to rest.

A turbo-generator running in parallel must have the load thrown off first and then the throttle quickly closed. Then the condenser is stopped and the steam drain opened to break the vacuum. Finally the water supply is shut off, but the oiling should be continued until the turbine is completely stopped.

Inspection

At regular intervals, and at least once a year, the turbine should be thoroughly inspected and all details gone over. This

applies to the interior of the turbine especially. Having lifted the cover, the clearances between rotary and fixed parts should be examined and readjusted. The rotor had better be removed and the blades minutely examined for erosion, scale, cracks, etc. The guide blades should undergo a thorough inspection, too, and be cleaned of obstructing matter.

The whole oiling system, chiefly, wants complete cleaning; the oil coils should be scraped outside and blown out with steam. Paraffin oil removes the gummy oxidized deposit, mentioned before as a result of bad oil. The oil strainer should be cleaned and an entirely new supply of oil put in. This is usually only necessary once a year; in the meantime the oil can be filtered and some new oil added to compensate for losses from leakage and evaporation. Bearings and stuffing-boxes will probably want some overhauling, too, and the governor and valves, the gears and oil pump should be gone over thoroughly. Great care must be taken in handling the rotor that no blades be injured; much trouble may arise from small damage.

The inspection will have to extend to the condensing plant. Air leaks may be looked for, and water leaks in surface condensers. Pumps, pipe lines, especially foot valves and strainers, must be cleaned and put in good shape.

Summing up the preceding statements, it doubtlessly will be conceded that the turbine requires less care and attendance than a reciprocating engine, and, if reasonably operated, maintenance and repairs amount to little. The engineer is thus relieved of much worry; he gets time now to devote his full attention to the problem of economical operation of his plant.

Let him think of plans to increase the general efficiency, of labor-saving devices and safety appliances, of precautions against shutdowns and of means for keeping track of performances. Recording thermometers and curve-drawing instruments should become his friends and help him watch conditions at any of the 24 hours of a day.

His duties are still more, for the modern engineer needs to know methods of testing fully, and, in the turbine plant, steam-consumption tests are the only means to control the efficiency of the machine. The indicator diagram, which affords an excellent guide to the reciprocating engineer, is of no use with a turbine. Steam measurements, compared with electrical output, are here the thing required.

The average engineer cannot be expected to proceed with the care used in acceptance tests, and refined measuring apparatus will generally not be at his disposal, but a scale for weighing the condensed steam and a few thermometers will give sufficient data for fairly approximate results.

In conclusion, let the engineer be careful of the external appearance of his turbine plant. He should pride himself upon things looking neat not only in the engine room, but also in the adjoining spheres of his activity, boiler rooms, basements, etc. In this endeavor modern design of power plants facilitates his task, as space, light, ventilation are more and more being provided.

Due to the reduced floor-space dimensions of a turbine, as compared with steam engines, the engine room will generally appear quite spacious, though in the basement things are still often crowded into inaccessible places. Let the engineer help progress also in this direction. And let him keep pace with the engine room; let him improve, with the introduction of turbines, not only his ability and knowledge, but also his personal appearance, social standing and dignity.

Shorter Cutoff, Greater Load

By F. R. Low

The later the cutoff on the low-pressure cylinder of a compound engine the less work it does. The article explains this apparent paradox in such a way as to be easily understood.

One of our subscribers has been having an argument with some fellow-engineers about the effect of changing the cutoff on the low-pressure cylinder of a compound engine. He writes:

"For the benefit of several engineers in this city, and possibly other places, I would be pleased to have you explain the apparent "paradox" whereby shortening the cutoff on the low-pressure cylinder of a compound engine will cause it to take more of the load.

"Some declare that the load can not be shifted from one cylinder to the other by manipulating the cutoff on the low-pressure, except to a very small extent. They declare that if they commence to shorten the cutoff on the low, and keep doing so, the engine will stop when that cylinder has taken a very small amount of the load, beyond what it had to begin with, assuming the load was equally divided at the start.

"I tried to tell them that it is possible to put nearly all the load on the low-pressure cylinder by shortening the cutoff. Also if the steam valves could be adjusted on the high-pressure side so as to take steam full stroke, that boiler pressure could be secured in the receiver by shortening the cutoff on the low, thereby causing it to take all the load, and a diagram from the high-pressure side would become a straight line.

"There is no difference of opinion as to the economy; the question is, can we change the load between the cylinders by changing the cutoff on the low-pressure, and to what extent? Please make this plain for there are many to whom this question is a puzzle."

The lower portions of Figs. 1 and 2 are the conventional diagrams of the high- and low-pressure cylinders respectively of a compound engine with the following data:

Initial pressure, 120 lb. absolute.

Cutoff in high-pressure, ¼ stroke.

Cutoff in low-pressure, ⅓ stroke.

Displacement of low-pressure piston, three times that of high.

Scale of high-pressure diagram, 60.

Scale of low-pressure diagram, 20.

These diagrams, to be comparable, must be reduced to the same scale, both of pressures and volumes. The high-pressure diagram would have been 60 ÷ 20 = three times as high as it is, if it had been plotted with the same scale of pressure (20) as the low, instead of on the 60-lb. scale to which it was plotted. The first step is to replot it upon the larger pressure scale.

Using the atmospheric line as a base, make $AB' = 3$ times AB; $CD' = 3$ times CD; $EF' = 3$ times EF, etc. The diagram would then become $B'D'F'G'HJB'$.

The horizontal dimension of such a diagram represents volume. The low-pressure piston displaces three times as much volume per stroke as the high, so that, if the length of the low-pressure diagram is taken as representing the displacement of that piston, the length of the diagram from the high-pressure cylinder must be made one-third the length of that of the low, for its length to correctly represent, to the same scale, the volume of the steam when it has expanded to fill the high-pressure cylinder.

Set off $ab = ⅓$ the length of the low-pressure diagram. Now squeeze the diagram up horizontally to this length. This is conveniently done by putting ordinates as 1′ 2′, 3′, 4′, into the reduced length spaced relatively to ordinates, as 1, 2, 3, 4, on the large diagram and transferring the pressures from one set of ordinates to the other. The pressure on ordinate 1 of the long diagram is 120 lb., as shown at D'. Make it the same on ordinate 1′ of the short diagram as at d. The pressure on ordinate 2 of the long diagram is 60 lb., as shown at F'. Transfer the same pressure to ordinate 2′ of

the short diagram as at *f*. Similarly the pressure, 40 lb. at *G'* on ordinate 3 of the large diagram is transferred to *g* on ordinate 3' of the reconstructed diagram. All that is necessary is to carry the pressures across horizontally. As many ordinates may be used as one chooses, but they must have the same relative positions in the two diagrams.

The high-pressure diagram, replotted to the same scales of pressure and volume as that of the low-pressure cylinder becomes *B'dfghJB'*. Placing this upon the low-pressure diagram so that the atmospheric lines coincide, we have the combined diagram shown in Fig. 2. Steam entering the high-pressure cylinder at 120 lb. absolute is cutoff at ¼ stroke, as shown at *B*, expanded to four times its volume at cutoff and hence to 120 ÷ 4 = 30 lb. at the end of the stroke in the high-pressure cylinder, as shown at *D*. The low pressure has three times the volume of the high, so that if it cuts off at ⅓ stroke it will take out of the receiver at each stroke the same volume that the high pressure delivers to it. There will therefore be no change of pressure in the receiver. The low-pressure cylinder takes out at each stroke a volume *CD*, Fig. 2, at 30-lb. pressure, and expands it to three times that volume (its cutoff being at ⅓) or to a pressure of 10 lb., as shown at *F*.

Whatever happens to the steam on its way through the engine it starts with a volume *AB* at the point of cutoff in the high-pressure cylinder and winds up with the volume *EF*, that is, the whole volume of the low-pressure cylinder, and, as it expands, the relations of its pressures and volumes will be expressed by some such line as *BDF*.

The work done in each cylinder is proportional to the area of its diagram. The work done in the high-pressure cylinder is proportional to the area *ABDCA* and that in the low to *CDFHGC*.

Now suppose the cutoff in the low-pressure cylinder to take place at ½ instead of ⅓ stroke. The volume of the high pressure is only ⅓ that of the low. If the low-pressure cylinder takes ½ of its own volume out of the receiver per stroke when only ⅓ of that volume is delivered to it the pressure in the receiver must drop and the steam in it expands until the greater volume taken out by the low pressure per stroke contains only the same weight as the lesser volume delivered to it by the high.

This means that the receiver pressure will be lower. Since the high pressure exhausts into the receiver its back pressure will be lower. The back-pressure line *CD*, of the high-pressure diagram would drop to *C'D'* and the area *CDD'C'C* would be added to the high-pressure diagram and taken out of the low, which would become *C'JFHGC'*. Since the work is proportional to the areas, the high-pressure cylinder is doing more work and the low-pressure cylinder less on account of the later cutoff on the low.

The high pressure does more on account of its reduced back pressure. The low pressure does less on account of its reduced initial pressure. A little area, *DJD'*, is not included in either diagram. This represents the work lost by free expansion in the receiver. It may have to be taken care of by a little later cutoff in the high-pressure cylinder, but this does not materially affect the argument.

If the cutoff in the low-pressure cylinder were carried back to ¼ stroke, as at *K*, the expansion would not follow the dotted line from that point, making that cylinder do less work, but the receiver pressure would rise to *LM*. The high-pressure cylinder diagram would become *ABMDNMLA*, having the loop *MND*, because it would expand the steam to *D* which would be lower than the receiver pressure. The further back the cutoff is carried the greater the loop becomes,

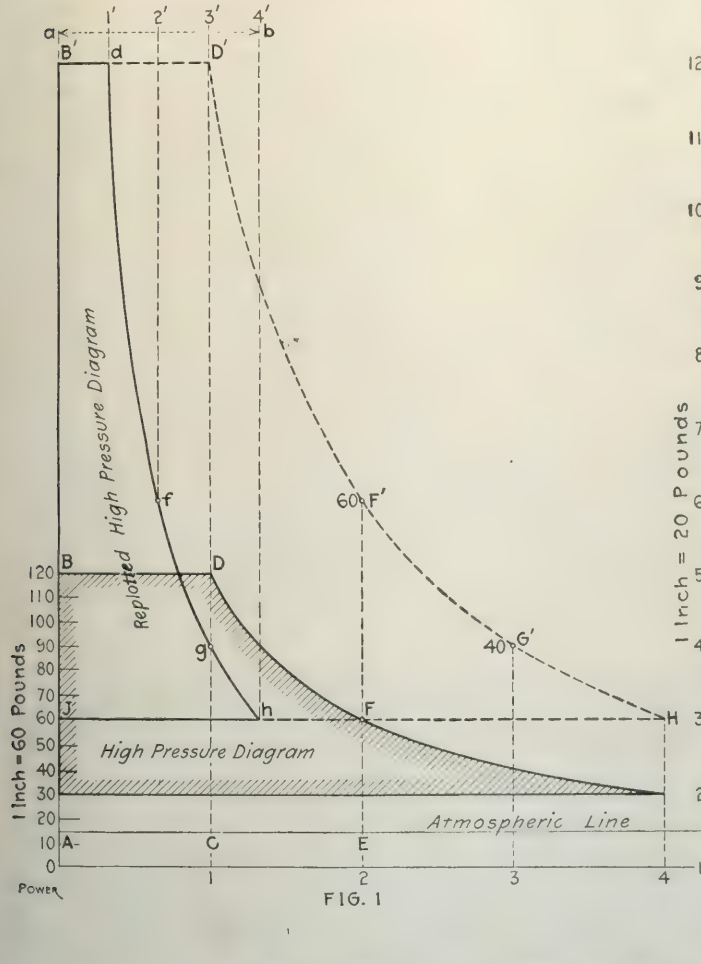


FIG. 1

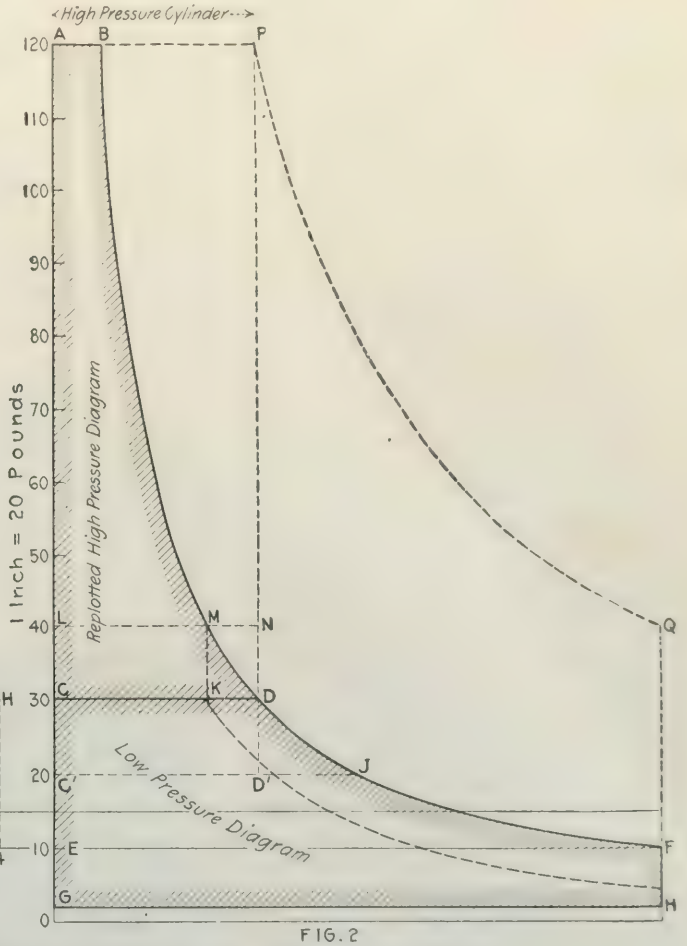


FIG. 2

THE SHORTER THE CUTOFF IN THE LOW-PRESSURE CYLINDER THE MORE LOAD IT CARRIES

until if it were carried back to $\frac{1}{12}$ stroke so that the volume taken out of the receiver were equal only to the volume *AB* coming to the high-pressure cylinder from the boiler, the receiver pressure would be equal to the boiler pressure and the high-pressure cylinder diagram would be a short straight line with an enormous negative loop represented conventionally by *ABDPBA*.

This loop represents work done by the high-pressure piston in pumping back into the receiver the steam which flowed from the receiver into the high pressure cylinder when its exhaust valve opened. It is therefore negative work, work imposed upon the engine in addition to its regular load. When the area of the loop becomes equal to the other, or positive part of the high-pressure diagram, the high-pressure cylinder is contributing nothing to the load, and the work available for running the engine will be only that represented by the low-pressure diagram. The useful work is the sum of the positive portions of both diagrams less the loop of the high-pressure diagram. For a constant load therefore the cutoff in the high-pressure cylinder would have to advance as soon as the diagram from that cylinder began to loop.

If, as our correspondent suggests, the steam were carried full stroke in the high-pressure cylinder with the low-pressure cutoff set so that the volume up to cutoff in the low just equalled the volume of the high, the conventional high-pressure diagram would become a straight line *AP*, Fig. 2, and the low-pressure diagram would represent the total work done by the engine.

The Legislature of Louisiana has passed, and the governor has signed an act making an eight-hour day for stationary engineers in that state.

Boiler Flue Deformed by Low Water

Inspector W. Ernst, in a communication to *Zeitschrift der Dampfkesseluntersuchungs und Versicherungs-Gesellschaft*, of Vienna, describes a serious case of low water where rupture of the boiler was prevented only by the prompt action of the operating engineer.

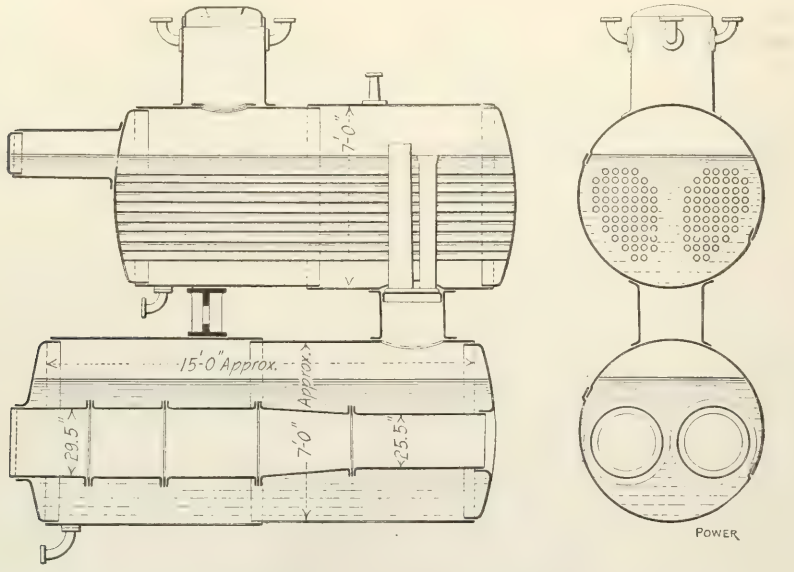


FIG. 1. SECTION THROUGH BOILER

The boiler was of the Tischbein design, as shown in Fig. 1, and was built in 1899 with about 1600 sq.ft. of heating surface and for a working pressure of 156 lb. gage. The upper and lower elements of the boiler had two water columns complete with all fixtures, including two gage-glasses for each column and a feed connection. Normally and at the time of the accident only the upper element was being fed as the water was supposed to flow to the lower element through the shorter of the two interior pipes. This extended vertically from the dome of the lower element to the designed water level of the upper one, and was about 16 in. in diameter.

Two boilers of this kind were in service at the time, both fed and fired by one man. The recording-gage chart showed a pressure between 142 and 156 lb., indicating uniform firing. Not so, however, with the feeding, as at 4:30 o'clock that morning the operating engineer noticed that the water was not visible in either glass of the lower element of one of the boilers in service. A quick glance into the flues showed him the danger of the situation and notwithstanding the great distortion of the flues he showed his coolness by drawing the fires with the assistance of the boiler attendant and thus probably at almost the last moment prevented the rupture of the flues, the condition of which is shown in Fig. 2.

The pressure then dropped to about 100 lb., through the demand of a turbine up to that time in operation. The stop valve was then closed and the pressure again rose to 156 lb. as a result of the further evaporation due to the hot brickwork. At six o'clock the blowoff was opened and the boiler blown down.

An internal inspection of the boiler showed on the shell the gradual fall of the water level, but particularly was this noticeable on the conical course of each flue so that undoubtedly low water was the cause of the injury. The first three courses of each flue were bagged in, the deepest depression being about 10 in.

In this plant the feed water is chemically purified and but little sediment is deposited. The good condition of the boiler inside and of the water column and its fixtures and the tight blowoff valve showed conclusively that only the fireman's carelessness was responsible for the sinking of the water level to a point where the crowns of the flues were uncovered.

The operating engineer was rewarded by his employer for his opportune action and bravery in preventing a disaster.

FIG. 2. SHOWING PARTIAL COLLAPSE OF FLUE

Each person in the United States used 11,000 lb. of coal in 1910, said a speaker before the Missouri School of Mines recently.

Boilers with and without Conveyors

By Henry J. Edsall

Some calculations showing the saving effected under certain conditions by the installation of an overhead bin and conveyor system. The figures cited are taken from the operating records of an actual plant.

While it is a well recognized fact among engineers that one of the greatest savings obtained by the use of mechanical stokers is the economy in the labor cost, it is seldom possible to get a direct comparison of two boiler houses with the same boiler and stoker equipment, and the coal handled mechanically in one case and by hand in the other. The writer recently saw two such boiler houses, however, in the same plant of a large corporation and, as the conditions effecting the cost of making steam are about the same in the two boiler houses, a comparison will be interesting.

Boiler House A

A cross-section of boiler house A is shown in Fig. 1. The boilers are set in two rows, with a low trestle between, on which the coal cars are run in and unloaded, so that the coal is placed directly within reach of the firemen; no wheeling or second handling is necessary before shoveling it into the stoker hoppers, unless the coal pile under the trestle gets low. This arrangement, therefore, is better than many which are still to be seen in good-sized boiler houses. In addition to shoveling the coal, there is the additional labor of breaking up the large lumps and, as this is not always properly done, there is more or less grate-bar trouble with the stokers, thus reducing the efficiency of the steam plant as well as the additional cost of maintenance on the stokers.

Ash Handling

For ash-handling there is a bucket carried by an electrically operated telpher running on an overhead track. The ashes are raked from under the stokers up to the boiler-room floor and then shoveled into the bucket, which takes them out and delivers them into a railroad car. This means considerable labor and, as there is but little room between the boilers and the coal pile, the men get into each other's way.

Boilers

There are sixteen 260-hp. Babcock & Wilcox boilers in this boiler house, and all are equipped with Roney stokers. This makes a total rated capacity of 4160 hp. At the time the figures were obtained only 10 boilers were in use, the other six being cleaned and repaired.

The weekly coal consumption with 10 boilers in service averages about 910 tons, or 91 tons per boiler. Assuming roughly 10 per cent. ash, there would be about 91 tons of ashes to be disposed of each week. Table 1 gives the total weekly cost for labor, power and supplies directly connected with coal and ash handling.

In Table 2 the figures for boiler house A are changed to what they would be with 12 boilers in service, thus affording a direct comparison with boiler house B, where there are, as a rule, 12 boilers in use. The only changes are providing a fireman for each additional boiler, both day and night, and a slight increase in the power cost.

Boiler House B

In boiler house B there are fourteen 260-hp. Babcock & Wilcox boilers, or a total of 3640 hp. Usually 12 of these boilers are in use at one time, the other two being held for cleaning and repairs. In this boiler room the coal is fed from an overhead bin to the Roney stokers and a conveyor system handles both the coal and the ashes. The run-of-mine soft coal is unloaded into a track hopper underneath the track, then delivered to the crusher by means of an apron feeder, and after passing through the crusher a bucket carrier elevates it and distributes it in the overhead bin. The ashes are raked out of the hoppers and into the lower run of the carrier, which elevates them to an ash pocket from which they are discharged into railroad cars. Fig. 2 shows a cross-section of the boiler room thus equipped.

Fig. 1. Boiler House without Conveyor System, Ashes Removed by Means of a Telpher

TABLE 1. BOILER HOUSE A

Coal Trestle and Ash Telpher Equipment

Sixteen 260-hp. boilers, ten in service; weekly coal consumption, 910 long tons

No. of Men	Duties of Men	Hours per Week, Each	Hourly Rate	Weekly Rate, Each	Total per Week
1	Telpher operator (day)	79	$0 18	$14 22	$14 22
1	Telpher operator (night)	89	0 18	16 02	16 02
3	Loading ashes to telpher (day)	79	0 18	14 22	42 66
3	Loading ashes to telpher (night)	89	0 18	16 02	48 06
3	Unloading coal from cars (day)	72	0 15	10 80	32 40
2	Unloading coal from cars (night)	78	0 14	10 92	21 84
1	Chief (day)	79	0 25	19 75	19 75
1	Chief (night)	89	0 25	22 25	22 25
1	Assistant chief (day)	79	0 22½	17 78	17 78
1	Assistant chief (night)	89	0 22½	20 03	20 03
10	Fireman (day)	79	0 18	11 22	142 20
10	Fireman (night)	89	0 18	16 02	160 20
					$557 41
	150 hp.-hr. for telpher motor at 3 cents	..	..	.....	4 50
	Supplies for telpher	..	..	.....	2 00
	Total per week	..	....	.....	$563 91

Coal Consumption

The average amount of coal consumed per week with 12 boilers in service is about 1260 tons, or 105 tons per boiler. This is an increase of about 15 per cent. over the amount of coal consumed by the boilers in boiler house A, which means that a greater output of steam is obtained from these boilers, because the coal is properly crushed and is fed continuously to the stokers, resulting in better and more even fires. Figuring 10 per cent. ash, as before, about 126 tons of ashes are disposed of.

Hours and Wages

The hours and wages of the men are given in Table 2, which shows a total

weekly labor cost of $395.09. The power used for handling coal is figured on the basis of 30 tons per hour, which is about the average amount handled. This gives a total of 42 hours per week, and taking the average horsepower consumed as 20, gives a total of 840 hp.-hours per week for handling coal. The actual amount of power used in handling ashes is hard to determine, but 660 hp.-hours per week would cover it, making a total of 1500 hp.-hours per week for handling both coal and ashes.

This gives a total weekly cost of $445 for boiler house B against a cost of $625 for boiler house A, or a saving of $180 per week and $9360 per year. This, of course, does not include the cost of maintenance and the depreciation of the coal- and ash-handling equipment or the interest on the investment in either case.

Overhead Bin and Conveyor System

Since the foregoing figures were obtained, four more boilers have been installed in boiler house A and, as the possible saving increases with a greater number of boilers, the advantages of an overhead bin and conveyor system for handling both coal and ashes are still more strikingly shown when the costs for this boiler house, with twenty boilers, are worked out.

Such a comparison is shown in Table 3, the first part showing the costs with the present type of equipment and the second part showing the costs with the overhead bin and conveyor system, the number of boilers in service being assumed as 16, and the wages, etc., being based on the actual figures obtained for the two boiler houses. No men are added for the present type of equipment except the firemen, though with this number of boilers in service there might be more men required for loading ashes to the telphers or for unloading coal from the cars.

With the overhead bin and conveyor system two day men and two night men are included for handling ashes. Having an efficient modern system this number should be ample; in fact, one man should be able to handle the 10 or 12 tons of ashes on each shift—about one ton per hour.

The excavating and concrete work for the ash hoppers and tunnels, etc., would make the installation of this system in an old boiler house considerably more expensive than in a boiler house designed especially to suit these conditions. The total cost of installation has, therefore, been assumed as $50,000, including an overhead coal bin with a capacity of 900 tons.

TABLE 2

Boiler House A

Coal Trestle and Ash Telpher Equipment

Sixteen 260-hp. boilers, twelve in service; weekly coal consumption, 1092 long tons

No. of Men	Duties of Men	Hours per Week, Each	Hourly Rate	Weekly Rate, Each	Total per Week
1	Telpher operator (day)	79	0.18	$14.22	14.22
1	Telpher operator (night)	89	0.18	16.02	16.02
3	Loading ashes to telpher (day)	79	0.18	14.22	42.66
3	Loading ashes to telpher (night)	89	0.18	16.02	48.06
3	Unloading coal from cars (day)	72	0.15	10.80	32.40
2	Unloading coal from cars (night)	78	0.14	10.92	21.84
1	Chief (day)	79	0.25	19.75	19.75
1	Chief (night)	89	0.25	22.25	22.25
1	Assistant chief (day)	79	0.22½	17.78	17.78
1	Assistant chief (night)	89	0.22½	20.03	20.03
12	Firemen (day)	79	0.18	14.22	170.64
12	Fireman (night)	89	0.18	16.02	192.24
					$617.89
	Weekly cost of power for telpher	..	.....	.....	5.11
	Weekly cost of supplies for telpher	..	.....	.....	2.00
	Total per week	..	.....	.....	$625.00

Boiler House B

Overhead Bin and Conveyor Equipment

Sixteen 260-hp. boilers, twelve in service; weekly coal consumption, 1260 long tons

No. of Men	Duties of Men	Hours per Week, Each	Hourly Rate	Weekly Rate, Each	Total per Week
1	Conveyor operator (day)	76	$0.30	$22.80	$22.80
1	Conveyor operator (night)	85	0.30	25.50	25.50
2	Feeding ashes to conveyor (day)	76	0.16½	12.54	25.08
2	Feeding ashes to conveyor (night)	85	0.16½	14.03	28.06
3	Unloading coal from cars (day)	72	0.15	10.80	32.40
1	Chief (day)	79	0.25	19.75	19.75
1	Chief (night)	89	0.25	22.25	22.25
1	Assistant chief (day)	79	0.22½	17.78	17.78
1	Assistant chief (night)	89	0.22½	20.03	20.03
6	Firemen (day)	79	0.18	14.22	85.32
6	Firemen (night)	89	0.18	16.02	96.12
					$395.09
	Weekly cost of power for conveyor system (1500 hp.-hours @ 3 cents per hour)	..	.	.	45.00
	Weekly cost of conveyor-system supplies	.	.	.	4.91
	Total per week	..			$445.00

TABLE 3

Boiler House A

With Present Coal Trestle and Ash Telpher Equipment

Twenty 260-hp. boilers, sixteen in service; weekly coal consumption, 1466 long tons

No. of Men	Duties of Men	Hours per Week, Each	Hourly Rate	Weekly Rate, Each	Total per Week
1	Telpher operator (day)	79	$0.18	$14.22	$14.22
1	Telpher operator (night)	89	0.18	16.02	16.02
3	Loading ashes to telpher (day)	79	0.18	14.22	42.66
3	Loading ashes to telpher (night)	89	0.18	16.02	48.06
3	Unloading coal from cars (day)	72	0.15	10.80	32.40
2	Unloading coal from cars (night)	78	0.14	10.92	21.84
1	Chief (day)	79	0.25	19.75	19.75
1	Chief (night)	89	0.25	22.25	22.25
1	Assistant chief (day)	79	0.22½	17.78	17.78
1	Assistant chief (night)	89	0.22½	20.03	20.03
16	Firemen (day)	79	0.18	14.22	227.52
16	Firemen (night)	89	0.18	16.02	256.32
					$735.85
	Weekly cost of power for telpher	..	.....	.....	8.15
	Weekly cost of supplies for telpher	...			3.00
	Total per week	..	.		$750.00

Boiler House A

Equipped with Overhead Bin and Conveyor System for Handling Coal and Ashes

Twenty 260-hp. boilers, sixteen in service; weekly coal consumption, 1680 long tons

No. of Men	Duties of Men	Hours per Week, Each	Hourly Rate	Weekly Rate, Each	Total per Week
1	Conveyor operator (day)	76	$0.30	$22.80	$22.80
1	Conveyor operator (night)	85	0.30	25.50	25.50
2	Disposing of ashes (day)	76	0.16½	12.54	25.08
2	Disposing of ashes (night)	85	0.16½	14.03	28.06
3	Unloading coal from cars (day)	72	0.15	10.80	32.40
1	Chief (day)	79	0.25	19.75	19.75
1	Chief (night)	89	0.25	22.25	22.25
1	Assistant chief (day)	79	0.22½	17.78	17.78
1	Assistant chief (night)	89	0.22½	20.03	20.03
8	Firemen (day)	79	0.18	14.22	113.76
8	Firemen (night)	89	0.18	16.02	128.16
					$455.57
	Weekly cost of power for modern conveyor system	..	.....	.....	40.00
	Weekly cost of supplies for modern conveyor system	..	.....	.....	4.43
	Total per week	..	.....	.....	$500.00

Cost of Maintenance

Assuming then that a loan of $50,000 is obtained at 5 per cent. interest, and that 3 per cent. per year will cover the cost of maintenance, the following saving is effected:

Weekly saving, $750 — $500 = $250; yearly saving, $250 × 52 = $13,000.

TABLE 4

	Maintenance	Interest	Net Saving	Unpaid Balance of Loan
First year.	$1500	$2,500.00	$9,000.00	$41,000 00
Second year...	1500	2050.00	9,450.00	31,550.00
Third year...	1500	1577.50	9,922.50	21,627.50
Fourth year...	1500	1081.38	10,418.62	11,208.88
Fifth year	1500	560.44	10,939.56	269.32

That is, the installation would pay for itself in a little over five years, and each year after this shows a clear saving of several thousand dollars. The 3 per cent. for maintenance should be ample, as much of the installation cost is for foundation work, concrete and steel bins, etc., which work would be either permanent or require but little for maintenance. Besides, coal and ash conveyors have reached a point of perfection in design and construction where they will stand up wonderfully well under very severe service, and the handling of both coal and ashes in one carrier is now good engineering practice.

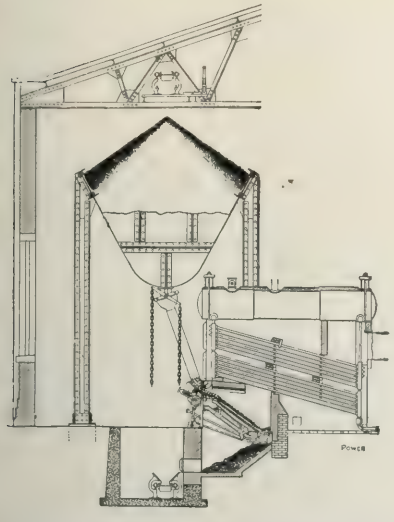


Fig. 2. Section through Boiler House Showing Conveyor System

In fact, a well designed carrier embodying first-class construction throughout should, when handling both coal and ashes, require little or no repairs in the first four or five years, and after this 3 per cent. per year would probably cover them, unless the amount handled is unusually large, in which case the cost per ton of material would show to even better advantage.

Notes on Friction and Lubrication

By Richard Howarth

The friction between two surfaces is greater when at rest than when in relative motion, but when in motion it is practically independent of the velocity so long as the surfaces are kept cool. The friction between two surfaces, dry or only slightly lubricated, is in direct proportion to the force with which they are pressed together within the limits of abrasion, and it is independent of the area of the surfaces in contact. With ample lubrication, the friction is naturally reduced, but the heavier the pressure per unit of surface, the greater must be the consistency of the oil to prevent it from being squeezed out of the bearing.

The friction under motion when the surfaces are dry is on the average 20 to 40 per cent. less than the friction of rest, and with ample lubrication the reduction averages 80 per cent. for low pressures to 98 per cent. for high pressures. If one could take a pair of absolutely true surface plates and place them together they could be moved over each other with a minimum force, as if the thin film of air formed a roller bearing; but if pressure is applied and the air is squeezed out, the surfaces are liable to seize and can be moved over one another only with difficulty.

The ordinary theory of solid friction is that it varies in direct proportion to the load; that it is independent of the extent of the surface in contact, and that it tends to diminish with an increase of velocity beyond a certain limit. The theory of fluid friction, on the other hand, is that it is independent of the pressure per unit of surface, and is directly dependent on the extent of surface and increases as the square of the velocity. The results of various experiments tend to show that the friction of a perfectly lubricated journal follows the laws of fluid friction more closely than those of solid friction; they show that under these circumstances the friction is nearly independent of the pressure per square inch, and that it increases with the velocity, though at a rate not nearly so rapid as the square of the velocity.

The coefficient of friction μ is the ratio of the pressure P required to overcome the friction of a body on any given horizontal surface to the whole load W or

$$\mu = \frac{P}{W}$$

Trigonometrically, it is equal to the tangent of the limiting angle of resistance,

$$\mu = tan.\ \phi$$

Friction of any kind, however produced, results in the conversion of mechanical work into heat. One horsepower, or 33,000 ft.-lb. of work per minute, expended in friction produces

$$\frac{33,000}{778} = 44\ B.t.u.\ per\ minute$$

Bearings

Increasing the diameter of a shaft increases the friction because the rubbing surfaces have farther to travel in one revolution; whereas, increasing the length reduces the friction per square inch, but does not affect the total friction, because for a given space passed through with a constant load the friction is independent of the surfaces in contact.

The bearing area is taken as the length multiplied by the diameter. When an overhanging bearing is increased in length the diameter must also be increased slightly to give the same strength as before.

Pressure on the journals in pounds per square inch longitudinal section may be

$$P = \frac{70,000}{50 + V}$$

in which V = velocity in feet per minute. Of course, P must never exceed 1000 as a maximum, or preferably about 800 lb. in slow-running engines down to 400 lb. in very high speed engines.

Lubricants

The lubricants commonly used for various classes of machinery, with very great pressure and slow speed, are graphite, tallow, vegetable grease and soapstone; with heavy pressure and high speeds, sperm oil, refined petroleum, olive and rape oil are employed. For steam-engine cylinders heavy mineral oil is best. If animal and vegetable fats were used in engine cylinders they would be decomposed by the high temperatures into acids having a corroding effect on the cast iron.

Cylinder oils, in common with other lubricants, should be absolutely neutral in their natural condition and incapable of developing acid when in use. They should also be free from mucilaginous or gumming properties and from dirt or grit. A good cylinder oil should have a specific gravity of about 0.9 at a temperature of 60 deg. F., and a flash point not less than 500 deg. F.

Cylinder oils may be rapidly tested for gumming properties by exposing them for 6 hr. in thin films to the action of steady currents of hot air at a temperature of about 300 deg. F. Oils have a tendency to oxidation and different oils display to a widely divergent extent their affinity for oxygen. In the best grades of mineral oils this tendency is entirely absent. Some of the lower grades of mineral oils have a tendency to discolor bright metallic surfaces and to produce thin, varnish-like films, which are doubtless due to the presence of the products of oxidation.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Inspecting and Testing Electrical Apparatus—I

By A. L. Cook

In large systems it is often necessary to make extensive tests before the apparatus is put in service and in such cases an expert is usually called in. Where no very complicated switching is involved or the system is small and the expense of a specialist not justified, the electrician or operator would be expected to make tests.

In this series of articles will be taken up the methods of inspecting and testing ordinary alternating- and direct-current generating and transforming apparatus.

Direct-current Systems

The direct-current systems most common are those supplying power and light to relatively small areas, or for a single building using 125 or 250 volts on a simple two- or three-wire system; and those for railway or power service, using 500 or 650 volts on a two-wire system.

It is essential, before making tests on new apparatus, to make sure that the switchboard equipment is in proper condition. All main connections should be carefully checked to see that the terminals of each generator connect through the switches and fuses and circuit breakers to the proper busbar. Two-wire compound-wound generators usually have the series field connected to the positive terminal. However, in grounded systems, such as for railway service, the series winding is often on the negative or grounded side. Anywhere that generators are to be run in parallel, all the series windings should be on the same side so that they can be placed in parallel through the equalizer busbar.

If circuit breakers of the ordinary laminated brush, carbon-break type are used the contact surfaces should first be wiped off as a little dirt may cause heating sufficient to oxidize the copper contacts, which will further increase the heating, and finally take the temper out of the brush. An imperfect contact can be detected by closing the breaker with a piece of tissue paper between the brush and the blocks. When the breaker is open, the mark left on the paper will indicate whether it is properly adjusted. The carbon blocks should make firm contact until the main brush has left the upper block, so that the final break in the circuit will take place on the carbon blocks, and save the main copper contacts from pitting. The breaker should be tripped several times by hand, and finally set to trip at the desired overload. For a new machine it is well to set this low, say 25 per cent. overload, until the generator has been well tried out, after which it can be set at 50 per cent. overload or more, if the generator will stand it.

Knife switches should be examined for clean surfaces and good contact between the clips and the switch blade. Vaseline or oil can be rubbed on the contact surfaces to keep them from cutting; a very little is sufficient. The back of the switchboard should be inspected to make certain that all contacts are tightly bolted and that there are no short-circuits. Fuses should be examined and the wiring tested for grounds and to make sure that the insulation resistance is not below a safe minimum.

Testing Insulation

Several methods for testing insulation are suitable for direct-current systems. A ground or short-circuit can be detected by a battery and buzzer or by an incandescent lamp, placing the apparatus in series with the testing circuit. These methods will indicate, however, only a complete ground or a leak in the insulation of a very low resistance, and if possible, some other method should be used.

A very common method, which is also employed extensively during the erection of electrical apparatus, is the use of a magneto-generator and bell, similar to that used for telephone lines. Both the generator and bell are generally mounted in one box having two terminals for the testing leads. One lead is connected to ground and the other to the apparatus to be tested. When the magneto-generator is turned rapidly, the bell rings if a ground exists on the apparatus. These magneto sets are rated according to the resistance through which they will ring; that is, a 100,000-ohm magneto will ring when the resistance between the ground and the apparatus is 100,000 ohms or less. The loudness of the ringing is an indication of the insulation resistance tested; if the bell does not ring, the resistance is more than 100,000 ohms. The same applies to magneto-generator sets rated lower. A magneto-generator cannot be used satisfactorily to test an underground cable 1000 ft. or more long, as the electro-static capacity of the cable causes the bell to ring even if no ground exists.

Sometimes it is necessary to test insulation resistances higher than can be indicated by a magneto. Then, if a supply of direct current is available, a voltmeter may be used to advantage with the connections as shown in Fig. 1. The supply voltage is first read by connecting the leads A and B, as indicated by the dotted line. Then this connection is broken and lead B is connected to the ground and lead A to the conductor to be tested and the voltmeter is again read. The resistance between the conductor and the ground may be expressed by the formula:

$$R = r\left(\frac{V - v}{v}\right)$$

where

R = Resistance to be measured;
r = Resistance of the voltmeter (generally marked on the inside of the instrument case);

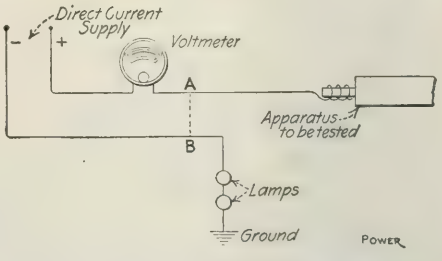


Fig. 1. Showing Method of Testing for Ground

V = Supply voltage as first measured;
v = Reading of the voltmeter when connected to the apparatus.

For example assume the voltmeter resistance is 15,000 ohms, the voltage V is 125, and v is 2 volts; then

$$R = 15{,}000\left(\frac{125 - 2}{2}\right) = 922{,}500 \text{ ohms}$$

If the supply is taken from the busbars and it is not desirable to ground one of them directly, the ground can be made through a low resistance, such as one or more incandescent lamps in series, the number depending upon the voltage of the system. If the test is made with a railway circuit which is grounded, the voltmeter should be connected to the permanently grounded busbar. In any case the voltmeter should be placed in series with the lead connected to the apparatus and not that connected to ground. One must be careful not to touch the bare terminal of lead A, or the conductor of

the apparatus being tested, as the leakage through the body would be sufficient to greatly change the reading v of the voltmeter. Sometimes when lead A is first touched to the apparatus to be tested, there is a momentary kick of the pointer. This is not an indication of the insulation resistance and the reading v should be taken only after the voltage has been applied for a few seconds and the voltmeter pointer has come to rest.

The resistance which can be measured by this method depends upon the supply voltage and the resistance of the voltmeter. Special voltmeters are made having a resistance of about 1,500,000 ohms with a 600-volt scale which, on a 500-volt testing circuit with a reading v of 5 volts, would indicate a resistance of 148,500,000 ohms or 148.5 megohms. These instruments are somewhat delicate to handle, however, and are more sensitive than is needed for ordinary tests. The ordinary switchboard and portable voltmeters of the permanent magnet type are good for this testing. Most makes of these have a resistance of about 100 ohms per volt of the scale or 15,000 ohms for a 150-volt meter, and if the exact value of r cannot be obtained, this value can generally be used in making tests.

Adjusting Brushes

Unless the generator is shipped completely assembled it is necessary to check the spacing of the brushes. To do this, place a strip of paper around the commutator under the brushes, and mark the position of each set. The strip of paper can then be removed and the distance between marks measured. If the distance varies the brush holders should be moved. The degree of accuracy with which the brushes should be set will depend upon the type of machine; with interpole generators the spacing should be more accurate than with other types of machines. The brushes on a given stud should be staggered with respect to those on the adjacent studs so that the entire commutator will be covered by the brushes, except a slight space at each end. This prevents grooves wearing in the commutator. After the brushholders are set, the brushes need fitting to the commutator. The carbon brushes which are usually used, except upon very low-voltage machines, can easily be fitted by using sandpaper while the machine is still. A strip of sandpaper is placed between the brush and the commutator, and pulled in the direction of rotation while the brush is pressed against the rough surface. The brush should be lifted while the sandpaper is returned, and then dropped down again for another stroke. It is also well to curl the sandpaper slightly so that it will lie close to the surface of the commutator, to prevent rounding over the back edge of the brush. The spring pressure on all the brushes should be the same and just sufficient to insure firm contact with the commutator.

After fitting the brushes, all dust should be blown out of the machine by compressed air from a hose. Too high a pressure should not be used at the nozzle, as it may tear the insulation; a pressure of 50 to 80 lb. per sq.in. is frequently used with a ⅛-in. nozzle. However, this is really more than necessary for cleaning the windings, and when so used the pressure at the nozzle should be reduced by throttling. It is convenient, however, to have a rather high pressure for cleaning out the air ducts in the laminated frame of a machine, where dust saturated with oil often lodges.

The bearings should be cleaned either by taking them apart, or by washing out with gasoline, and then they should be filled with oil to the proper height. If the field of the generator is mounted on a foundation separate from that of the armature, the air gap should be checked with a wedge-shaped gage. It is best to select one point on the armature and gage between this and each pole face, by turning the armature over as required.

The brush-holder rigging should be rotated until the brushes are set in the proper running position. In all except interpole generators the brushes are given a forward lead, that is, are moved in the direction of rotation a slight distance from the neutral point. Generally the neutral point is marked by the manufacturers, and is where the brushes will be approximately opposite the centers of the main poles. If not marked, it can be determined, if the armature conductors can be traced out to their connection with the commutator. If the two sides of a coil span the whole distance between adjacent poles, that is, if the throw of the coil equals the pole pitch, the brushes would be on the neutral point when they short-circuit the coil which is exactly midway between the poles. If the throw of the coil is less, the neu tral position of the brushes must be approximated as closely as possible. With interpole generators, the brushes should generally be set on the neutral point or in some cases may be given a very slight forward lead. A forward lead makes the generator more stable when operating in parallel with other machines, but reduces the compounding somewhat.

Grounds and Drying Out Generator

Before starting the generator it is best to test for grounds, particularly in 500-volt machines. A magneto may be used if nothing better is available, but would only indicate a bad ground and is not satisfactory. If direct current is available, it is desirable to test for grounds with a voltmeter, using the method previously described. Ordinarily, if the machine is in proper condition, the voltmeter will not deflect appreciably; anyway an insulation resistance of 1,000,000 ohms would be sufficient. A reading of the full supply voltage would indicate a "dead ground," or an actual metallic circuit from the windings to the frame. In a new machine, this would probably be caused by a mistake in erection, or by some damage to the insulation. The cause of this ground should be discovered and removed. A voltmeter reading less than full-line voltage indicates low insulation resistance, but no complete ground. This may be due to dirt on the insulation of the brush-holder parts or terminal blocks. If no trouble is apparent here, probably the insulation is damp and the machine should be dried out.

To do this, the generator should be run with the armature short-circuited beyond the ammeter and circuit-breaker, or fuses, and the current in the shunt-field circuit adjusted to give approximately full-load current. With a compound-wound machine, the speed may have to be reduced for this run to limit the current. The temperature of the parts of the windings which are accessible should not be allowed to exceed 160 deg. F., and the current should be adjusted to maintain this temperature. Tests of the insulation resistance should be made at intervals during the drying-out run.

As the machine heats, the insulation will test lower than when cold, but after a time the resistance will increase, and when thoroughly dry will remain practically constant as long as the temperature remains the same.

When the generator is ready to operate, the main switch should be opened and all the field resistance cut in; then the generator can be started slowly. The oil rings should be inspected to see that they revolve freely, and the machine can be brought up to speed. The voltage may now be gradually raised to normal, by cutting out resistance in the field rheostat, noting, as this is done, any tendency to sparking at the commutator. If sparking occurs the brushes should be shifted slightly until it ceases.

Reversed Polarity

The polarity of the generator should next be checked. If the new generator is to be put in service with machines already in use, the polarity of the busbars is fixed. If a voltmeter is available, two temporary leads may be connected to it and the instrument used for checking the polarity before closing the main generator switch for the first time. The voltmeter leads should first be attached to the busbars, noting which, when connected to the positive lead of the voltmeter, deflects it to the right; this is the positive bar. The generator leads at the switch terminals should be tested similarly, and the positive terminals determined.

If the connections cannot be checked this way, a simple test may be made as

shown in Fig. 2. The switch *S* should be open, the circuit-breaker *C* closed and incandescent lamps connected between the switch terminals. For a 125- or 250-volt system, a single lamp of the same voltage should be used on each side. For higher voltages enough lamps should be placed in series so that the sum of the voltages of the lamps on each side will approximately equal the voltage of the system. If the polarity of the generator is correct, the voltage of the generator will be opposed to that of the busbar and the lamps will not burn. If the polarity is reversed, the two voltages will be added and the lamps will burn brightly. It is assumed that the connections between the generator and main switch have either been traced out, or tested with a bell and buzzer, or by some other method, so that the equalizer terminal and the other two terminals of the generator can be identified. As previously pointed out, the series windings must be in parallel, hence the proper connection of the generator leads at the main switch is fixed. If the polarity of the generator is reversed, it cannot be corrected by reversing the leads, but the polarity of the generator must be changed, usually by reversing the residual magnetism of the field. To do this, all the brushes should be lifted clear of the commutator, leaving no circuit through the armature. The generator circuit-breaker should be opened and its terminals shunted by a small fuse. Upon closing the main switch the field will become magnetized in the proper direction by current from the busbars. Before opening the main switch, the field rheostat should first be all cut in to reduce the current as much as possible, and then the circuit should be opened slowly by the main switch. These precautions should be observed particularly on large or high-voltage machines, otherwise the field winding will be subjected momentarily to a very high voltage. If the field magnetism cannot be reversed by current from the busbars, the necessary changes may be made sometimes at the generator. The brushes should be moved a distance corresponding to the space between adjacent poles and the terminals of the shunt and series fields reversed.

Sometimes, in starting a new machine, the voltage will not build up when the field rheostat is cut out. This is usually due to a poor contact in the field circuit caused by a loose connection, or dirt on the commutator. Sometimes the brush pressure can be increased on two brushes of opposite polarity by pressing them down until the voltage starts to build up. If this is not effective the whole field current should be examined for a loose connection or imperfect contact, not forgetting the connections between the field coils. If the machine still fails to generate, the terminals of the shunt field should be reversed, as it may be connected so that it tends to build up in opposition to the residual field. If this is ineffective, the connections should be made as originally. If other machines are available, the generator field can be magnetized as described for correcting reversal of polarity.

A voltmeter, if available, can be connected across the armature terminals to measure the voltage due to the residual field. With the shunt-field circuit open, the voltmeter should indicate several volts. Loss of magnetism is rare, however, and can be restored by exciting the field from the busbars. It is always advisable to see that the shunt and series coils are properly connected. To do this, the voltage should be raised to normal and then some load placed on the machine. The voltage should be noted, and then the terminals of the series coils short-circuited. If the voltage drops, the connections of the series coils are correct; if it rises, the connections are wrong and should be reversed.

Putting Generator on the Line

When preparing to put a shunt-wound machine in parallel with other machines

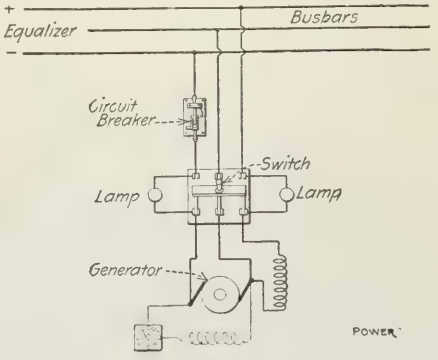


Fig. 2. Testing for Polarity of Generator

which are in service, the circuit-breaker of the incoming machine should be closed and the voltage made slightly higher than that of the busbar. Then the main switch can be closed and the field rheostat adjusted until the generator takes its share of the load. With compound generators, where single-pole switches are used for the positive, negative and equalizer leads, the two switches connected to the series field should be closed first. The shunt-field rheostat is then adjusted until the voltage of the incoming machine approximately equals that of the busbars; then the remaining switch can be closed. The shunt-field rheostat can then be adjusted until the incoming machine takes its share of the load. If a three-pole main switch is provided, the voltage of the incoming machine should be made a few volts higher than the busbar voltage, after which the switch can be closed and the field rheostat adjusted to give the proper load. If a new compound or interpole generator will not take its share of the load, the machines should be tested separately to see that they give the same voltage for corresponding loads, and, if necessary, the shunts on the series coils should be adjusted so they will.

After the machine has run satisfactorily for a short time, the load may be shifted from the other machines to the new one until 25 per cent. overloaded.

Switches and Feeders

The instruments on the switchboard should be checked as far as possible. All voltmeters should be compared by placing on the same voltage. Watt-hour meters should be checked to see that the disk is rotating in the proper direction. If an ammeter appears to be in error, the connections from the shunt to the meter should be examined for loose connections and to make sure that the serial number of the shunt and the meter are the same.

If new feeders are to be put in service they should first be tested for grounds and short-circuits. Grounds or partial grounds on short feeders can be detected with a magneto, but more satisfactory is to use a voltmeter as previously described. When using a voltmeter, lead (*B*), Fig. 1, should be grounded and lead (*A*) connected to the feeder. The allowable insulation resistance depends upon the character of the feeder. The National Board of Fire Underwriters requires completed wiring, exclusive of sockets and receptacles, to have a resistance of 200,000 ohms for a system of 100 amp. capacity, 50,000 ohms for 400 amp. and other capacities in proportion. This includes all branch wiring, so that a single feeder should have a much higher insulation resistance. Underground feeders should, in general, have a high insulation resistance. An overhead feeder or an underground feeder connected to a third-rail system would have a rather low insulation resistance. A bank of lamps may be used to test this class of feeders, since high insulation resistance is not necessary.

Storage Battery

After a storage battery has been erected, it requires a special forming charge before being put in regular service. This charge usually requires from 40 to 50 hr., but need not be continuous after the first 12 hr. Care should be taken to have the positive terminal of the charging circuit connected to the positive terminal of the battery. The acid should not be placed in the cells until ready for the forming charge. A large amount of gas is given off near the end of this charge and for this reason the battery room should be thoroughly ventilated and no flame brought near the cells as the gas sometimes forms an explosive mixture with the air. The rate of charge is specified by the manufacturers and generally starts at a value somewhat greater than the 8-hr. discharge rate, and is decreased as the charge proceeds.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

American Practice in Rating Internal Combustion Engines—III*

By T. C. Ulbricht and C. E. Torrance, Jr.

American Fuel Oils and Gases

To complete the tables of data as given by the manufacturers, the authors found it necessary to collect all available information on the analyses of American fuel oils and gases, the average values of which are shown in Tables 1 and 2. To make this as complete as possible, a brief description is here given of each type of gas used for power puposes, taken in the same order as found in the tables.

All results in the tables, which are inclosed in brackets, represent inserts by the authors as a result of their best judgment for the case considered.

The latest determinations of heating values that could be found were used in checking the data given by the various analyses, for higher and lower calorific values. After careful checking with several American and German authorities, the values given by Levin, were selected as the standard with which to compare given calorific values, both high and low.

Power Gases

All power gases contain more or less hydrogen, in addition to hydrogen gases, the latter being combinations of hydrogen and carbon in various ratios. Over 200 hydrocarbon compounds exist in gaseous, vaporous liquid and solid states. The gaseous forms are known as fixed, or permanent gases, since these do not condense to form tarry liquids and sooty deposits when subjected to a reduction in temperature, as do the various compounds.

The two principal hydrocarbon gases are methane or marsh gas (CH_4) and ethylene (C_2H_4). Both give off considerable heat during combustion and require more air than does hydrogen to insure complete combustion with oxygen. The products of combustion are water vapor and carbonic-acid gas. The compounds of carbon and oxygen are carbon dioxide (CO_2)) and carbon monoxide (CO). The former is inert and therefore valueless for power purposes, and merely dilutes power gases, so that designers always attempt to obtain as little as possible. Carbon monoxide, however, is an active gas, and has the power of combining with oxygen.

Therefore, since the inflammable gases are composed of several gases, both combustible and noncombustible in varying ratios, they are not of equal value as far as power production is concerned. Yet when mixed with air in such proportions as will permit proper control of explosions and greatest efficiency when burned within gas-engine cylinders, the amount of heat produced per charge does not differ greatly.

For instance, natural gas may evolve from 800 to 1000 heat units for every cubic foot of air, while producer gas will produce from 120 to 150 heat units under similar conditions. Yet in practical gas-engine work, it is necessary to dilute the natural gas so that only about one-tenth of the charge is combustible, while the producer gas needs only a little more than its own volume for dilution. Thus, whatever power gas is used, the amount of power produced per explosion is practically the same.

To produce an ideal power gas, the following points, as noted by J. R. Bibbins, should be observed:

1. Cheapness of generation.
2. Simplicity of operation, for which reason the recovery of the byproduct should not be undertaken by plants, unless they are engaged in the manufacture of crude or refined chemicals.
3. A continuous process.
4. A low percentage of hydrogen and inert gases should be obtained.
5. Because of higher compression in engines, etc., a clean gas, free from dust and other foreign substances is essential.

Producer Gas

This is the name applied to the group of gases which are similar in composition, but vary in the proportions of constituents, that are generated by passing air or a mixture of steam and air, through an incandescent bed of carbon.

When a producer is working normally, there is generally a thin bed of ashes on the grate; on top of this is a deep bed of incandescent fuel, and above this a layer of fuel that has not attained such a high degree of incandescence. The gas is produced by drawing or forcing air through this incandescent bed. With air alone, however, the temperatures become excessive, and the tendency is to generate a superfluity of carbon dioxide (CO_2). To counteract this steam is introduced into the air blast, and serves to absorb the surplus heat while decomposing, and to enrich the gas with hydrogen; also additional oxygen is furnished for the gasification of the carbon. Dowson, Riche, Mond, Siemens, etc., are familiar names for gases in this class that are similar in character.

Illuminating Gas

This is made by three distinct methods:

a. Water gas—resulting from passing steam through a bed of incandescent carbon (coke), which has first been raised to a high temperature by a forced blast of air. The process is intermittent, and the gas burns with only a pale-blue light.

b. Carbureted water gas—produced much the same as plain water gas, except that oil is added, sprayed into the water gas to give it illuminating properties. This process is also intermittent.

c. Coal gas (bench gas)—which is the result of heating bituminous coal in air-tight retorts. The volatile constituents of the coal are driven out by being subjected to great heat for about six hours, and are passed through various washing and purifying processes.

Natural Gas

Natural gas is obtained from the interior of the earth, in various parts of the world, but especially in the neighborhood of oil fields. Its chief constituent is marsh gas (CH_4), with small amounts of hydrogen, carbon monoxide and unsaturated hydrocarbons. This gas is well adapted to gas-engine use, and is devoted to this purpose extensively in the natural-gas regions around Pittsburgh, Buffalo, Indianapolis and throughout Ohio.

Blast-furnace Gas

This gas is given off from cupolas, used in the production of iron and steel. To make pig iron, the ore is treated in a blast furnace, with coke or anthracite coal and some kind of flux. Some of the fuel is burned by an air blast, and as a result of this heat, the balance unites with the oxygen of the iron ore, leaving more or less pure iron. The gas given off contains from 20 to 34 per cent. of carbon monoxide, some hydrogen and hydrocarbons, from dissociation of water and volatile matter in the fuel, some carbon dioxide, and much nitrogen. Blast-furnace gas is always weak, because of the small amount of CO, and requires a much larger cylinder capacity for a given power than other gases. However, under cer-

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TABLE 1. AVERAGE ANALYSES OF AMERICAN GASES

Gas	Average Constituents of Gas—in Per Cent. Volume									Theoretical Calorific Value per Cu.Ft. of Gas at 62° and Atmospheric Pressure B.t.u.		Calorific Value per Cu.Ft. by test B.t.u.		Theoretical Air per Cu.Ft. Gas	B.t.u. per Cu.Ft. of Mixture		Calculated B.t.u. per Cu.Ft. of Mixture—Theoretical Air		Calculated B.t.u. per Cu.Ft. Mixture of Mixture—Excess Coef.-as Below-				
	CO_2	CO	H	CH_4	C_2H_6	C_2H_4	C_6H_6	N	O	High	Low	High	Low		High	Low	High	Low	1.1	1.2	1.3	1.4	1.5
Producer																							
Anthracite	6.03	22.38	13.38	1.96	....	0.36	....	55.65	0.803	138.4	125.7	142	131.3	1.053	67.9	64.1	67.5	61.3	58.2	55.4	53	50.8	48.7
Bituminous	9.12	17.54	11.73	4.28	....	0.36	....	57.24	0.265	137	125	154	136	1.120	(72.7)	47.8	64.6	59.0	56	53.4	50.8	48.7	46.6
Coke	4.90	27.30	10.07	1.18	....	....	....	56.60	0.550	128.7	126.3	133.7	(125)	0.985	(67.4)	(63)	65.0	63.7	60.7	57.8	55.4	53.1	51.0
Lignite	9.43	18.90	15.13	3.65	....	0.424	....	52.50	0.582	144	134	156.4	....	1.145	(72.8)	....	67.2	62.5	59.3	56.4	53.8	51.5	49.3
Oil	4.10	11.40	5.57	5.87	....	3.10	....	66.90	3.060	176	151	....	....	0.990	....	....	88.4	75.9	72.2	69.0	66.0	63.2	60.7
Peat	12.40	21.00	18.50	2.20	....	0.40	....	45.50	0.000	153	141	175	....	1.170	(80.6)	....	70.6	65.0	61.6	58.8	56.3	53.5	51.2
Wood	13.90	20.03	21.00	2.79	....	0.60	....	41.80	0.185	137	128.7	138.7	....	1.070	(67.0)	....	66.2	62.2	59.2	56.3	53.8	51.5	49.4
Illuminating																							
Water	4.72	34.8	48.81	4.06	....	1.97	....	7.16	0.503	322.1	304	329	278	2.63	(90.6)	(76.6)	88.8	83.8	78.0	73.2	68.8	64.8	61.5
Carbureted Water	2.07	24.1	32.40	23.40	....	12.52	....	3.75	0.510	646.4	589	677	632	6.84	99.1	90.94	82.4	75.1	69.0	63.9	59.5	55.7	52.3
Coal or Bench	1.21	6.18	43.94	37.78	....	5.87	4.16	3.50	0.502	677	608	655	592	6.00	97.8	86.40	96.7	86.8	80.0	74.2	69.2	64.7	60.8
Natural																							
Average	0.684	0.647	10.89	79.67	....	1.26	6.40	5.89	0.79	948	853	965	855	9.85	96.2	87.2	95.2	85.7	78.5	72.7	67.5	63.0	59.1
High Hydrogen	0.580	0.730	20.56	50.30	....	2.07	5.88	8.80	1.13	923	834	895	810	8.66	(92.6)	(83.8)	95.5	86.3	79.2	73.1	68.0	63.5	59.5
Low Hydrogen	0.800	0.525	1.92	91.40	....	0.515	....	3.25	0.50	967	870	971	862	9.20	96.2	87.2	94.7	85.2	78.3	72.4	67.1	62.7	58.8
Blast Furnace	11.80	26.75	3.40	0.30	....	....	....	58.8	0.182	98.7	95.2	98.7	....	0.735	55.3	54.8	56.8	54.8	52.7	50.5	48.6	46.8	45.2
Coke Oven	2.1	6.51	51.56	32.9	(1.4)	2.5	....	4.0	0.36	577.0	487.0	598.0	506	4.76	96.7	84.0	100.0	84.5	78.2	72.4	67.8	63.5	59.8
Oil Gas	2.79	4.96	17.70	23.60	18.78	5.53	....	37.55	0.702	595.0	542.0	662.0	....	4.94	(111.3)	....	100.0	91.2	84.2	78.2	73	68.4	64.4

TABLE 2. AVERAGE ANALYSES OF AMERICAN FUEL OILS

	Baumé Gravity Scale	Specific Gravity	Per Cent. C	Per Cent. H	Calorific Value per Lb. B.t.u. Calculated B.t.u.		Calorific Value per Lb. Actual by Test B.t.u.		Theoretical Air Cu.Ft. per Lb.	Calorific Value per Cu. Ft. of Mixture	Theoretical Air B.t.u.	O Plus N	Per Cent. N	Per Cent. O	Per Cent. S	Calorific Value per Cu.Ft. Mixture (Low) Calculated with Given Excess Coefficient, B.t.u.					Boiling Point	Flash Point
					High	Low	High	Low								1.1	1.2	1.3	1.4	1.5		
Gasoline	70.2°	0.704	82.54	14.91	21,271	19,818	19,740	18,500	193.5	109.2	102	4.3	1.93	0.31	0.09	93	85.2	78.8	73.2	68.2	176°	55
Kerosene	47°	0.863	84	14.14	20,892	19,600	20,849	18,755	189	110.3	103.6	2.3	1.838	2.0	0.23	94.3	86.3	79.7	74	69	437	130
Crude oil	18.3°	0.877	84.24	13.44	20,162	19,160	19,560	18,636	187	111.7	102.3	3.1	0.2	1.93	0.823	93.2	85.3	78.8	73.1	68.2	..	..
Fuel oil	..	0.939	83.04	11.58	19,183	18,138	19,121	..	178	110.2	102	..	0.60	2.82	0.90	92.7	85.0	78.5	73.0	68.2	.	.

tain conditions, its use has resulted in a marked economy.

Coke-oven Gas

This is produced in large iron and steel works, from the furnaces or ovens used in producing coke for smelting processes. It is similar to the illuminating gas known as coal gas, and in both cases the tar and liquor produced during the purification process, results in the recovery of valuable byproducts.

Oil Gas

Oil gas is the result of heating crude petroleum, refined petroleum or any other oil composed of hydrocabons, in a closed retort, so that "cracking" results, and hydrocarbons which are gaseous at atmospheric pressure and temperature are obtained. By properly regulating the time and temperature, the process of decomposition can be stopped at any desired point. Thus, liquid hydrocarbons may be decomposed into liquid hydrocarbons of less density, and further heating decomposes these into gaseous hydrocarbons, which process is termed "cracking." Oil gas is not the same as vaporized oil, since in the latter the hydrocarbons are unchanged, and if the vapor is cooled, a liquid will result.

Calorific Values of Fuel Gases

All computations for the calorific values of fuel gases are taken for the lower heat values. The standard formula for the calorific value is:

$$H_v = (\text{——})\ CO + (\text{——})\ H + (\text{——})\ CH_4 + (\text{——})\ C_2H_6 + (\text{——})\ C_2H_4$$

the symbols representing the percentages of the constituents in the gas. After careful checking with German and American authorities, the following values in B.t.u. per cubic foot (lower value) at 62 deg. F for the constants in the foregoing formula were selected as giving the heat value:

$$H_v = 324\ CO + 275\ H + 910\ CH_4 + 1620\ C_2H_6 + 1512\ C_2H_4.$$

As will be noted from table 1, no matter what gas is used, the heat value per cubic foot of mixture is very nearly the same. This is because of the variation in the quantities of air required to burn the different gases. For instance, a gas with a low heat value, would normally give a low horsepower for a given size of cylinder. The air required to burn this, however, is less per cubic foot, so that more gas is taken into the cylinder and more heat units are liberated.

Liquid Fuels

The liquid fuels comprise crude petroleum and various tarry residues, which consist principally of combinations of carbon and hydrogen, together with comparatively small amounts of nitrogen, oxygen and sulphur. Taking hydrogen and carbon as the main constituents, it is found that those oils which are rich in the former element have a light specific gravity, as compared with those rich in carbon. The range of specific gravity of California oils may be taken from unity to 0.84 or from 10 to 36 on the Beaumé scale. The majority of fuel oils will range from unity to 0.9 or from 10 to 23 Beaumé.

The Beaumé scale bears the following relationship to specific gravity.

$$Specific\ gravity = \frac{140}{130 + B}$$

or

$$B = \frac{140}{specific\ gravity} - 130$$

The calorific value of liquid and solid fuels can be computed by Du Long's formula, in which the most approved heat values (low) have been used. This is:

$$\text{Calorific value} = 14{,}600\ C + 52{,}275\ (H - O\ 8) + 4050\ S$$

where H, C, O and S represent respectively the percentages of hydrogen, carbon, oxygen and sulphur.

CORRESPONDENCE

Tar Troubles with Producer Gas

One of our troubles in the operation of bituminous producers is the filling up of the horizontal runs of pipe between the vaporizer and the scrubber with a compound of tar and soot, which is too thick to flow. This apparently gathers on the upper half of the pipe until it becomes heavy enough to break loose and fall to the bottom, there building up until the passage is so restricted that the generators must carry 2 or 3 in. pressure.

Removing this is hard and uncomfortable, and precautions must be taken to prevent the men from being overcome by

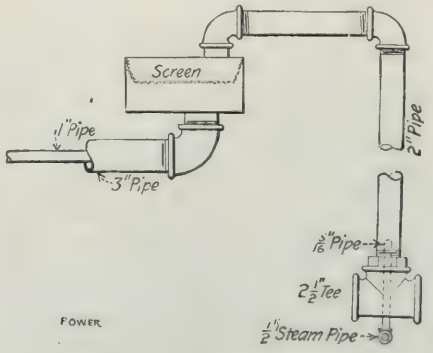


FIG. 1. TAR ELEVATING JET

the gas. Even with the passage to the generator closed, enough gas will remain in the pipes and apparatus to overcome a man, and this dead gas seems to be more poisonous than when the plant is in operation. The tar which was not exposed to the air also seems to give off a poisonous gas.

To avoid accidents when doing this or similar work, it is best to induce a circulation of air through the piping by a fan or blower. The least it has cost us to clean the two pipes, each 25 or 30 ft. long, has been $14.50, and this is only one of the various items of expense that go to make up the cost of producing gas from bituminous coal.

We put a 1-in. steam pipe in the gas lines, hoping to be able to warm them up and blow the tar over into the scrubber where it would be easier to remove. We

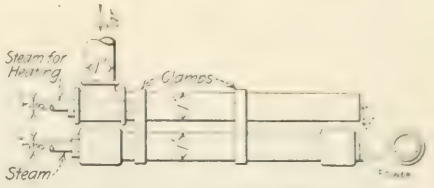


FIG. 2. TAR BURNER

are doubtful, however, as to its effectiveness. We find it necessary to run steam pipes to all the main gas valves to warm the tar in order to be able to open them when starting. Also all separating and tar wells, as well as pipes carrying tar, have to be provided with steam pipes and heating coils. Exhaust steam is used wherever possible, but at the same time considerable live steam is used and the available heat from the tar that we are able to save and burn for steam making amounts to not more than 30 per cent. of the heat necessary for this purpose.

We tried handling the tar with pumps but found it too thick and dirty, and were beginning to despair of a way out of the difficulty when we thought of blowing it through with steam. Fig. 1 shows the device used. The tar flows in by gravity, and, after being heated by the steam, is forced up through the pipe. The tar requires large free openings up to where the steam is applied. Where the jets discharge from the separating wells to the receiving well, we have placed a strainer box to catch the lumps of soot which, if not strained out, would cause trouble.

From the receiving well the tar is elevated about 12 ft. above the burner at the boiler, which is 100 ft. away, and to which it is conveyed through a 3-in. pipe, having a 1-in. steam pipe inside. The tar makes an intense flame, hence must not be too close to the shell. We have overcome this by dropping the grates 4 ft. below the shell, at which point the tar burner enters from the side. Also a small coal fire is kept on the grates to ignite the tar if the burner becomes temporarily clogged and the flame goes out. This burner, as shown in Fig. 2, is very simple, being made of pipe and fittings.

A burner having small openings is useless with producer-gas tar. If our burner becomes plugged, opening the regulating cock wide for a few seconds will pass the lumps through.

J. O. BENEFIEL.

Anderson, Ind.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

The Heating and Ventilating Engineer

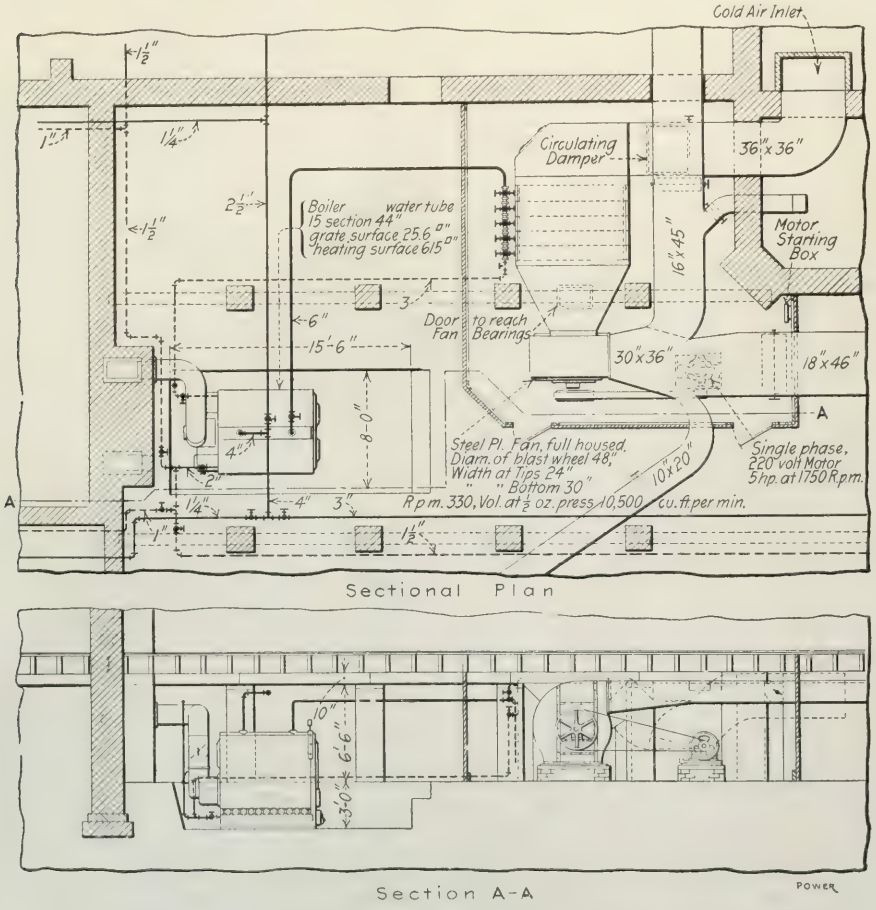
By J. P. Lisk

Under present conditions the work of a heating and ventilating engineer is difficult and unsatisfactory. Generally he is not consulted concerning the construction of the building, but must take it as it is given him, and fit his apparatus to it, avoiding all interference with architectural or constructional features. The apparatus must be ample in size, highly economical in operation and, of course, inexpensive. Any place in the basement or cellar that cannot be used for some other purpose, is generally assigned as the place to put the heating and ventilating apparatus.

Although not generally so understood the equipment is essentially a power plant requiring in its design and installation specialized engineering ability and experience. The foregoing statement is made to correct the prevailing impression that heating and ventilating is a simple side line of either mechanical or electrical engineering and may be picked up and successfully applied by any bright assistant in the office.

Doubtless as the price of fuel advances more attention will be given to the design of heating and ventilating installations, based on the economy of operation rather than the economy of installation, now so painfully apparent to the occupants of underheated apartments, and to the heating contractor who would like to do good work but cannot on account of the prices owners are willing to pay for this very essential part of their building equipment.

The reason that ventilating systems fail to work is that they are not worked. An occasional private owner and generally municipal authorities will employ a competent engineer to design a heating and ventilating system, have it installed, pay for it, and after they have learned by one season's experience that it costs money to ventilate a building, they have the ventilating part of their system put out of commission and blame the engineer for not providing a more economical plant.



A Typical Heating and Ventilating Plant Design

Without question there are many engineers in this country and in Europe who are perfectly competent to design systems for heating and ventilation to meet the requirements of any public or private building in temperature, humidity, volume and purity of air supply as prescribed by physicians and sanitary experts. Unfortunately, there are no legally fixed standards, either municipal or state, for volume or purity conditions to guide the engineer. This statement is made with full knowledge that there are certain sections in New York State laws and also in the New York City charter which mention ventilation but give absolutely nothing to guide the heating and ventilating engineer. He is still obliged to assume the prerogatives of the state and municipal authorities and physician when he is called upon for service and is expected to render a quality of service that will bear inspection by the most critical.

There should be fixed standards of ventilation for every class of public building. Sanitarians should determine what would be reasonably good hygienic conditions to maintain. The various societies interested could influence the passage of laws embodying the necessary standards. The engineer could then lay out a system to fulfill the required conditions with some degree of satisfaction.

The engineer is not a physician, nor a sanitarian, but he is perfectly capable of fulfilling conditions prescribed by either. If the public had some definite standard by which to measure his work, they would know what they were getting, and general satisfaction would undoubtedly prevail.

The following shows that it is easy to obtain desired results even in a small plant with fixed conditions to meet. A building for which the author designed and installed a hot-blast heating and ventilation system, contained several auditoriums, the seating capacity of which totaled 675 and the longest period of occupancy was two hours.

The net volume of the combined auditoriums was 121,700 cu.ft. The temperature to be maintained within the building, with an outside temperature of zero, was 65 deg. F. A sanitary expert specified 900 cu.ft. of air per hour per seat, which was to be delivered hot enough to in-

sure a temperature of 65 deg. in the rooms.

The total volume of air would be

$675 \times 900 = 607{,}500$ *cu.ft. per minute*

The heat lost by ventilation, the air being discharged through the vent flues at a temperature of 65 deg. F., was estimated to be 13,163 B.t.u. and the heat lost through walls and glass 11,645 B.t.u., making a total of 24,808 B.t.u. per minute, to be supplied by the apparatus.

With a fixed volume of 10,125 cu.ft. it was necessary to deliver the air at 120 deg. F. at the inlet register to maintain the desired temperature of 65 deg. in the auditoriums. The room construction was such that the air could be delivered most advantageously 10 ft. above the floor, which permitted a delivery velocity of 6000 ft. per minute without inconvenience to the occupants.

From the above data the size of the necessary apparatus can be determined. This includes a boiler to generate the steam, a radiator to transfer the heat to the air, a fan to move the air and an electric motor to turn the fan.

Formulas based on 12 years' experimenting and testing apparatus installed by the author and checking by installations of other engineers under working, not theoretical, conditions follow: The symbols used are given first.

A = Volume of air in cubic feet per minute passing through heater under ordinary barometric and humidity conditions at sea level;

H = Heat units per cubic foot of air to obtain the desired temperature;

U = Latent heat in B.t.u. in one pound of steam at atmospheric pressure = 970.4;

N = Number of four-row sections in one group;

C = Relative condensation per section or group of sections;

V = Velocity of air in feet per minute passing through heater;

R = Constant (0.00125) based on the velocity of the air passing through heater;

F = Square feet of heating surface in the heater;

W = Weight of the water of condensation in pounds;

a = Free area through heater in square feet.

To find the square feet of surface in the heater, use the following formula:

$$F = \frac{\frac{AH}{U}}{VCR} \times 60$$

To determine the pounds of condensation use

$$W = \frac{AH}{U}$$

For the free air through the heater the following formula may be used:

$$a = \frac{A}{V}$$

Applying the formulas to the case in hand, $A = 10{,}125$. The temperature of delivery must be 120 deg. F., which requires a five-section heater. The relative condensation of a five-section heater is 1.08 lb. per square foot and the number of heat units required is 2.2 per cubic foot. The total surface in the heater is, therefore,

$$F = \frac{\frac{AH}{U}}{VCR} \times 60 = \frac{\frac{10{,}125 \times 2.2}{970.4}}{1200 \times 1.08 \times 0.00125} \times 60 = 850 \text{ sq.ft.}$$

The water of condensation from this heater will weigh

$$W = \frac{AH}{U} \times 60 = \frac{10{,}125 \times 2.2 \times 60}{970.4} = 137 \text{ lb.}$$

The free area through the heater is

$$a = \frac{A}{V} = \frac{10{,}125}{1200} = 8.44 \text{ sq.ft.}$$

The size of the heater being determined, it remains to find the boiler power required, size of fan, and motor to drive it. The accompanying plan gives the size of the apparatus installed and pipe sizes. Besides providing for the hot-blast system the boiler was proportioned to care for 1300 sq.ft. of direct radiating surface.

With an outside temperature from 1½ deg. below zero to 1 deg. above, the delivery at the register was 11,200 cu.ft. of air per minute at 124 deg. F. with 5 lb. gage pressure at the boiler. These results go to show that, with given conditions to meet, a system can be designed which will give satisfaction, and besides there is a basis by which to judge the work.

Sectional Steam Heating in Chicago*

By S. Morgan Bushnell

The Illinois Maintenance Co. has operated on a scheme somewhat diverse from that followed by most steam-heating companies. It has installed no plants of its own and, with one or two exceptions, has installed no boilers. It simply contracts to operate boiler plants already installed in buildings, so as to furnish steam for heating and other purposes. It might be thought that the central-station idea was being lost sight of and that a policy of decentralization, rather than centralization, was being adopted. To a certain extent this is true, but it must be borne in mind that the conditions to be met in the central parts of larger cities are radically different from those in smaller towns.

The heating company in the smaller town which conducts a house-to-house business has much in its favor. In the first place, there is a marked difference in economy between the small heating boilers and the large heating boilers of from 300 to 500 hp. found in central stations. Under tests conditions, house-heating boilers of 25 hp. show an average efficiency of from 40 to 50 per cent. Probably under ordinary working conditions the percentage might fairly be assumed at 35. Assuming an average working efficiency on the larger sizes of boiler of 65 per cent., gives a difference of nearly 100 per cent. in favor of the larger boiler.

Furthermore, small residences and stores find it somewhat impracticable to burn soft coal and usually adopt anthracite on account of its comparative cleanliness, reduced danger from explosive gases, and the greater ease of controlling the fire. While there is no great difference in the heat value of anthracite and the better grades of bituminous coal, the anthracite costs fully 100 per cent. more.

Multiplying these two factors together, an ordinary householder will pay four times as much for a given number of heat units produced in his boiler as the central-station manager. This saving would offset a considerable expense through fixed charges on investment, distributing losses, etc., and still leave a margin of profit for the station.

In the business center of a large city like Chicago, however, the conditions are different. The tall office buildings in the downtown section have boilers ranging from 100 to 400 hp. capacity. Their efficiency will probably average at least 55 per cent. They generally have automatic furnaces adapted to use the cheaper grades of soft coal. The only advantage, therefore, to be gained, from the coal standpoint, would lie in the comparatively small difference in efficiency between the medium-size boiler and the very large one. But this is not all. While gaining only a comparatively small amount in efficiency by substituting central-station boilers, the difficulties of steam transmission in a large city, where the streets are already occupied with various other pipes and conduits, make the cost of transmitting the steam very high.

From time to time estimates have been prepared on the cost of installing a large steam-generating plant in Chicago, but thus far no one has seemed to be able to figure out enough profit from operating the larger plant to offset the heavy investment required. It has, therefore, been the policy of the steam company in Chicago to operate steam plants for only a very limited territory, using one set of boilers for a single building or for several adjoining buildings.

In a number of cases several buildings have been connected to one set of boilers, while each building, as a rule, retains its own boiler, or boilers, for use in

*Abstract of paper read before the National District Heating Association, at Detroit.

emergency or in extremely cold weather. Although the company operates some boiler plants for single buildings, these contracts were taken with a view to combining the building with others in the vicinity as soon as contracts could be negotiated.

All the boiler plants are supervised by a first-class operating engineer, who employs the men. This man visits all of the plants daily, watches the general upkeep and the handling of the boilers by the men, and also keeps close account of the coal consumption. He has an assistant, who visits the plants and also acts as an emergency man to assist in any temporarily short-handed plant. This assistant also makes night inspections to see that the night engineers are as strict in performing their duties as the day men.

The company originally operated entirely on flat-rate contracts, furnishing steam to each building at a fixed price per year. This method of selling steam has been unsatisfactory, and the contracts are being changed as fast as practicable to a meter basis. In the smaller buildings ordinary condensation meters are used, while in the larger ones steam is measured at high pressure by St. John meters.

The rates charged are considerably lower than the meter rates in New York and other large cities, due mainly to the company having little or no investment costs on which to figure a return.

Thus far very little steam piping has been done in the streets, most of the work done up to the present time being to connect buildings on opposite sides of a street or alley. As the field of the company's operations gradually extends, a somewhat more elaborate system of street connections will probably be installed, the plan being to shut down some of the plants now running all summer, to cut down the labor cost. More than fifty boilers are now in charge of the company, but probably less than half of them are used at any one time, on account of the method of operation.

The company's experience in operating boiler plants has shown that the average steam plant in a store or office building is operated with considerable waste and that careful superintendence will show economies which, taken together, will amount to considerable. These economies, however, are largely offset by waste on the part of a customer when steam is sold on a flat-rate basis.

The operation of these plants thus far has not been financially successful. The line of work being a comparatively new one, a great many things were to be learned by experience, both in handling the plants and in negotiating with customers. The net loss in operation, however, has been gradually reduced, and the outlook appears more promising now than at any time since it was organized.

Someone has called this system of heating "Heating from Decentralized Plants." This is true only to a very limited extent. The work of the company would more properly come as a division under the head of "District Heating," only the districts are much smaller than those usually served from heating stations. It is expected that as the plans of the company develop, some of these districts will be considerably enlarged and that from year to year additional buildings will contract for their heating from these centers.

For example, in the case of the Insurance Exchange Building on Jackson Blvd. and Fifth Ave., four 350-hp. boilers are now installed. Space has been provided for considerable extension, if desired. The stack and breeching have been designed for a capacity for nearly 3000 hp. Negotiations already have been started to supply some of the surrounding buildings, and as the company has a long-term contract with the building, it will probably be able to connect others to this center. Another typical plant is that of a large department store which, in addition, supplies steam for an adjoining office building and two restaurants.

Running pipes underground has been quite a serious question in the operation of these plants. Apparently, from the nature of Chicago soil, some methods of installation which have given fairly satisfactory service in other cities deteriorate here very rapidly, and it has been found that in a very few years after installing the pipes nothing has been left, but the iron pipe and the concrete. The iron pipe also seems to be attacked more or less, either by acids or electrolysis, and in several cases has had to be replaced or a smaller pipe installed within the original one.

A year ago the company laid an 8-in. pipe line, 300 ft. long, connecting the boilers in the old Edison Building on Market St. to the boilers in the new office building at Madison and Market Sts. for the *Chicago American* and *Chicago Examiner*. Where the line crosses the street it was laid at an average depth of 9 ft. A tile sewer drain was first installed, in gravel, and over this was laid a bed of small rock and sand to promote drainage. This was covered with reinforced concrete, in which roller bearings for the piping were embedded. The pipe was covered with two thicknesses of $\frac{1}{16}$-in. sheet asbestos paper, painted with special tar paint. A dead air space of 1 in. intervenes between the asbestos covering and a paper felt conduit 1½ in. thick lined with sheet tin. About this were two layers of tar felt paper completely covering the line. This also was painted with special tar paint, and outside was a casing of reinforced concrete 3 in. thick.

The operation of the line thus far has been very successful. The steam service has been continuous. No appreciable leaks have developed, and the loss through condensation has only amounted to 300 lb. per hour. As the steam transmitted in cold weather is about 10,000 lb. per hour, a fairly satisfactory efficiency in transmission is obtained.

The connected radiation amounts to less than 500,000 sq.ft., but more than 30 elevators, ice machinery, pumps, hot-water heaters, and other apparatus give a summer load requiring a coal consumption of upward of 2000 tons per month. Thus nearly half the coal consumption of the company is for other purposes than serving radiators, but expressing the steam demands in terms of radiation, the equivalent is over 800,000 sq.ft.

As the large cities grow older and more compactly built up, it is a question as to what extent the theory of combination and coöperation will be applied to the production of heat, light and power in large buildings. Some middle ground may be found between the large central sources of supply and entirely decentralized operation, along the lines of which the greatest economy can be secured. The above scheme of operation is being gradually extended in an effort to discover this middle ground.

CORRESPONDENCE

Radiators Bypassed

Some years ago I had charge of an exhaust steam-heating system that did not give satisfaction. The radiators were all supplied with steam from a common riser running from the basement to the third floor near the center of the building. All of the returns from the radiators were led into a common 3-in. return discharging into a sewer. Although a back pressure of from 10 to 12 lb. was often carried, some of the radiators always remained cold.

The trouble was that the steam passed from the riser through the nearest radiators to the return and never reached the radiators near the ends of the building. Air valves on the radiators did but little good. I made a dead-weight safety valve out of an old globe-valve, put it in the return pipe and set it to open at about 5 lb. This did very well as a substitute for a trap, and from 2 to 3 lb. back pressure was all that was required.

J. H. Campbell.

Bloomington, Ill.

In too many cases the engineer and manager are intellectual and moral strangers to each other, even after years of close association. The manager can do more to promote intellectual acquaintance than the engineer and it will in most cases pay to make the effort. Some have said that "friendship and business cannot mix"; maybe not, but pleasant relations most certainly lead to mutual good will and confidence, the basis of a successful business of any kind.—*Ice.*

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POWER

Turbine Classification

Ask "What is a spiral stairway?" and it is ten to one that the person addressed will indicate, with a helical movement of his finger in the air, his conception of the structure, before he can get it into words.

Ask an engineer "What is a Curtis turbine?" and ten to one his fingers will interlock in imitation of the active and reversing blades of this type. He will explain that the Curtis turbine is a "velocity-stage" machine in which the steam is expanded in nozzles until it attains a high velocity, and that this velocity is abstracted in two or more stages or steps by allowing the steam as it emerges from the nozzle to impinge upon a blade moving at a velocity considerably less than one-half that of the jet, so that it leaves the blade with considerable residual velocity. The latter is utilized by turning the jet through a stationary or reversing blade so that it is flowing again in the direction of the rotating vanes upon which it is allowed to impinge with a further abstraction of its velocity.

When the velocity induced by its expansion in the first set of nozzles is sufficiently abstracted the steam is allowed to expand to a still lower pressure, restoring its velocity which is again abstracted in two or more stages. This is continued until the condenser pressure is reached, the designer seeking to divide the work equally among the several pressure stages, which will be greater in number as the difference between the initial and condenser pressures is greater, and as the peripheral speed of the vane is less.

The Curtis turbine is therefore a combined pressure-stage and velocity-stage turbine.

Not necessarily. Almost everybody thinks so, and that is the way the type is ordinarily built; but Curtis' patent calls for *one or more* velocity stages for each pressure stage, and the Algemeine Elektricitäts Gesellschaft builds them with several velocity stages in the first-pressure stages, but with simple rotors with one row of blades in the subsequent stages, which thus become similar to those of the Rateau or Zoelly machine. But the Curtis patents antedate those of Rateau and the courts are now busy deciding whether any impulse turbine divided into pressure stages does not infringe them.

Ask an engineer "What is a Parsons turbine?" and one hundred to one he will tell you it is a combined impulse and reaction turbine. The steam is expanded in the stationary blade passages and discharged into the moving blades driving them by its impact. It is further expanded in the moving blades themselves, so that it is shot backward with an increased velocity. The velocity of the jet relatively to the casing or ground is thereby decreased, and the energy equivalent to the difference in that relative velocity when it enters and leaves the blade is left in the rotating member, in addition to that imparted by its impact.

This is the popular conception of the Parsons turbine, and is in fact the way it is ordinarily built. But reduced to its lowest terms and greatest efficiency the Parsons turbine is a pure reaction machine. A nozzle is very efficient; 90-odd per cent. of the energy represented by the heat fall involved in the expansion will be present as kinetic energy in the jet issuing from a properly designed nozzle. The impulse bucket is less efficient. There is loss in the jet between the nozzle and the blade, there is loss by impact and friction and eddies, so that the energy expended in rotative effect is a less proportion of that inherent in the steam than is that from the moving reaction nozzle.

The ideal Parsons turbine, then would not be a combined impulse and reaction, but a pure reaction machine. The steam would be expanded in the stationary blades only until the component of its velocity in the direction of the blade movement was equal to the blade velocity, and it could get aboard without shock or impact, as one runs until he acquires the same speed as a moving train

and then swings aboard as though he were a part of the moving mass.

This would require either a very high blade velocity or a small pressure drop at each stage, which, in turn, means a great number of stages. For ordinary commercial conditions the designer is obliged to allow the steam to enter the buckets faster than their motion, and thus, to some extent, to act by impulse.

The Power Situation

It is reported that Francis S. Peabody, acting for the Commonwealth Edison Co., of Chicago, is about to effect a great combination of coal-mining interests of central Illinois, preparatory to one of the greatest, if not the greatest, centralized system of power production ever planned.

After the mining interests merger, it is planned to erect a huge power plant at Kincaid, Christian County, from which current will be supplied to all the Commonwealth properties, including those in Chicago. Kincaid was selected on account of the water power available there. A part of the plan is to merge all, or at least nearly all public-service plants in the state.

This announcement, quickly following that of the proposed Lehigh Navigation Electric Co.'s 100,000-kw. plant, eventually to be built at Hauto, Penn., and the huge Southern power developments involving in one case a capitalization of $120,000,000, is indicative of a veritable revolution in power generation, and it is all most commendable in that it will undoubtedly mean cheaply produced power and a great reduction of our annual coal consumption. But will it mean a relatively cheaper power to the consumer? That is the vital question.

Recent testimony given before the Maryland Public Service Commission revealed that a hydro-electric power company realized a revenue of $125,000 annually from current sold to a copper works at ½ cent per kw.-hr. The city of Winnipeg, Manitoba, now sells current for lighting at a base rate of 3⅓ cents, with discounts ranging from 10 to 60 per cent.; for heating and cooking (separate meter service), 1 cent, and for power from 3⅓ cents to 0.08 cent per kw.-hr. It is said that highly suitable returns are realized on these rates. These cases are exceptional, but they show what is actually possible with a fair return on the investment. We hope the consumer will enjoy an equitable share of the benefits to be gained by such centralization of power as is now going on throughout the country.

Electrification of Railroads

The steam-locomotive engineer's attitude toward the electric locomotive, as evidenced at the Manhattan Beach conference, recalls the antagonism of the old stationary engineers to steam turbines and gas engines. Of a truth they were "stationary" engineers, but something better might have been expected of "locomotive" engineers.

Progressive engineers, of either calling welcome each advance in their respective fields and further it. Attempts to hinder it are evidences either of fear or prejudice, and certain to have their reward in the obstructors being left at the side of the road of progress blinded by the dust of the passing show. Such is the fate of the "standpatter."

The locomotive engineer is fighting for the sole right to operate electric trains of either the traction or multi-unit type, although in the same breath he protests against their existence—a philosophical acquiescence to the inevitable. We admire his common sense.

The electrification of roads is growing fast. By 1913 the New York, New Haven & Hartford R.R. will have about 552 miles electrified. The Southern Pacific Co. will shortly spend $8,000,000 on such work in and around Portland, Ore. The Pennsylvania extensions are well known. Elaborate experimental work is now being carried on near Toulouse, France, on the Midi Ry. On the Lauban-Konigszelt line of the Prussian State Ry., many miles will be electrified following experiments to determine operating and maintenance costs. A saving of 15 per cent. is expected. The mileage cost there for electric locomotives is said to be $3.73 per 100 miles against $11.20 for steam. The shop maintenance cost is $23.20 per 1000 miles against $60.48 for steam locomotives. England and Germany are also rapidly developing electric railways.

All this will require large power plants equipped with modern units for generating high-tension current, and highly competent engineers will be in demand to operate them. It is a development, therefore, that is likely to be watched with interest by readers of POWER.

Shorter Cutoff Greater Load

The engineer who is accustomed to see the cutoff on the high-pressure cylinder grow later as the load increases, and who knows in a general way that the greater load requires more steam and that it takes a later cutoff to supply it, is often puzzled to understand how a later cutoff on the low-pressure cylinder of a compound engine makes it take *less* of the load.

The difference is that the high-pressure cylinder has the boiler to draw upon. The longer the admission valve of the high-pressure cylinder stays open the more steam will come into the engine. The low-pressure cylinder, however, has to take what steam the high-pressure cylinder gives it. All of the steam admitted to the high-pressure cylinder must pass through the low whether the cutoff on the latter be late or early. The cutoff on the low-pressure cylinder cannot determine, therefore, how much steam it shall get. It has got to take the same weight of steam that the high-pressure cylinder took. If it takes it at a greater volume it will be at a lower pressure. Therefore if its cutoff is late and it takes a large volume out of the receiver it will take it at a lower initial pressure for itself, making its capacity less, and leave a lower receiver pressure or back pressure for the high-pressure cylinder, making its capacity greater. An attempt to explain this diagrammatically will be found on page 180.

Although Herbert Knox Smith, Commissioner of Corporations, has resigned, we trust that the campaign to retain, lease and develop water-power sites will be as vigorously carried on as before. Mr. Smith has done much in this direction, and we hope his successor will do as well or better.

The Bureau of Standards is able to determine the millionth part of an inch, says *Technical World Magazine*. Now we can approximately estimate the "progress" made by a few people we all know, where heretofore we were doubtful if some of their little movements had any forward significance.

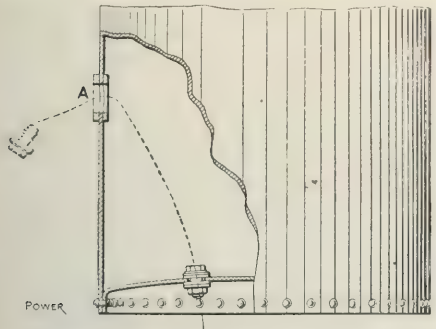
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Novel Leaky Tank Repair

A jet of water about the size of an ordinary sewing needle, shot forth from the bottom of a small tank which an engineer was testing. The usual way of repairing such leaks is to drill and tap a hole for a machine screw. This would not do in this case for the drilling revealed that corrosion had eaten the bottom of the tank at this point to not over ⅛ in. thick.

It was decided that a patch should go on the inside, something after the style of a manhole plate in a boiler. A ¼-in. stove bolt with a smooth, flat surface on



SHOWING HOW A LEAKY TANK WAS PLUGGED

the under side of its head was fitted with a small washer of thin good quality packing; a smooth fitting iron washer followed next and finally another washer of rubber packing ⅛ in. thick.

A piece of flexible lamp cord was passed up through the leak hole and hooked out through the 1¼-in. hole *A* in the illustration. The wire in the end of the strand was untwisted and bound to the threaded end of the bolt, over which had been slipped the several washers. The bolt and washers were guided through the 1¼-in. hole and the end of the bolt passed through the leak hole as shown. A washer of ⅛-in. rubber packing, a metal washer and a nut were then put on the outside and the leak was effectually stopped by tightening the nut.

T. H. REARDON.

North Adams, Mass.

Shaft Governor Lubrication

The speed-control device on reciprocating engines driving generators, where the machines must carry a fluctuating load, must be so designed as to afford all simplicity, accuracy and good workmanship possible. This is a vital matter to the different engine manufacturers.

Some time ago I bought two second-hand 50-kw. 250-volt direct-current generators directly connected to simple engines equipped with a shaft governor. These engines, as could be plainly seen, had been severely used; the governor pin, holding the eccentric to the arm and counterweights of the governor, and the bushing in the flywheel were so worn that before reinstalling these engines, it was necessary to rebush the bearings to get the eccentric face parallel to the faced part of the flywheel; otherwise the governor would not work properly.

When I removed these old brass bearings I found that the grease-holes did not penetrate through the bushing, making it impossible to get lubrication at the bearing point. I found also that in the governor there is a vital part where lubrication is needed badly, yet the only way to apply it is with the ordinary oil can. When these engines were started they would race past the rated speed, making it necessary to throttle them by hand, unless the load was put on as the normal speed limit was passed. It occurred to me one day to thoroughly oil the bearing end of the spring, where the loop connects to the shift pin in the governor arm. This proved a remedy for the racing.

Obviously the spring tension and the pulling force caused by the rotation of the counterweights in the governor arm at a point off center, act through an arc determined by the point of suspension and considerable friction is caused at this point. This friction is such that it does not let the weights act to throttle the engine at the proper moment.

A close regulation in speed has been obtained ever since, by keeping this bearing point well greased.

ALBERT R. MESA.

New York City.

Power Plant Dangers

Several years ago a water arch on a boiler was cut out of service by removing the steam connection while the water nipple was capped and left in the boiler. Lately it became necessary to rebuild the brickwork of this furnace and this gave an opportunity to remove the water arch. The old nipple was noticed and a wrench was applied to remove it, but the cap dropped off with the slightest touch.

Undoubtedly something serious would have happened if this burnt nipple and cap had blown off while the boiler was under high pressure. This case shows that there are many little dangers lurking in our boiler rooms, and that an eternal vigilance should be exercised to keep all things right. It would be interesting to learn of the many things in need of immediate attention which can only be discovered by a systematic search for them.

F. WEBSTER.

Scranton, Penn.

Cement Enclosed Leaky Pipe Joint

Often, in an emergency, it becomes necessary to make up joints in an unusual manner, due to lack of facilities, or peculiarities of the situation not allowing ordinary treatment.

An unusual experience came to me last year while isolated in the wilds of southern Mexico, where supplies of all kinds were out of immediate reach. We had an "air-return" water-supply system for a sugar factory and distillery, using two large compressors in the factory and four submerged tanks at the river in a pit. The discharge pipe from the river tanks was of 16-in. spiral construction.

A section near the river had to be renewed, and where the pipe reached the river bank was a rise which made an angle in the line of about 40 deg. The pipe was fastened at the joints with flanged stuffing boxes, and on putting in the new section and realigning it, we found that the angle was too great to allow using the stuffing-box joint at the angle. The condition was aggravated by the ragged condition of the pipe end on the old section.

Cement had served so well in so many emergencies, that I concluded to build a cement block around the joint, so I had my "albanils" (as we call the Mexican brick masons) build a brick and cement wall around the joint, thinking it would at least serve until I could devise something better, or get a special construction of joint.

We built a block of ordinary red brick and cement around the joint for a distance of 4 ft. and with 9-in. minimum thickness at the four curves of the pipe. This joint was allowed to dry for about two weeks, much longer, perhaps, than was necessary, before we started the pumping. At the starting, and never since, has this joint leaked a drop of water, and it bids fair to equal, if not outlast, the tubing.

A considerable pulsation in the operation of the submerged tanks at time of filling caused vibration in the piping, and, in addition, the joint was exposed to expansion from a tropical sun.

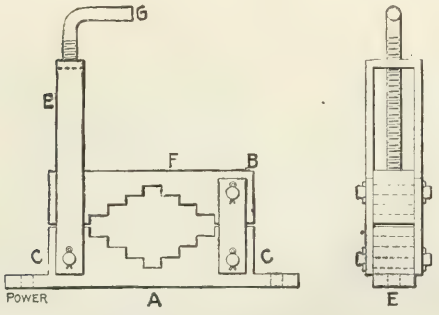
The strength of the joint was not questioned, but it was doubtful whether the brickwork would adhere to the pipe close enough for a water joint, with the vibration, exposure and a distance of 800 ft. and a head of 110 ft. The whole thing was a success and I am more friendly than ever toward cement.

J. O. Frazier.

New Orleans, La.

Simple Home-Made Pipe Vise

The illustration shows a pipe vise which will hold pipe from ¾ in. up to 1¼ in., and which can be carried in an inside pocket of a coat. It weighs no more than a 14-in. pipe wrench. This is the design of a vise actually in use.



Home-made Pipe Vise

At *A* is shown the vise assembled and ready for use. The cotter pins *BC* should be used to secure the cross pins on which hinge the yoke *E* and the upper jaw *F*. By making another hole nearer the right side of *F*, 1½-in. pipe may be held in the vise. The width of the vise should be sufficient to withstand the strain put upon it.

John Strong.

London Junction, Ont., Canada.

Improvised Planimeters

Although the use of a jack-knife as a planimeter is old, some may not know that quite accurate results can be so obtained.

To find the area of a diagram, the approximate center of gravity is first marked and two lines are drawn at right angles through this point, as at *A*, Fig. 1. The knife should be opened as shown and the point of the small blade placed at *A*, and the large blade on the line which determines the point *A*. Then, with the knife held vertical, the outline is traced with the point of the small blade by moving it around from *A* to *B* in the direction *B, C, D, E, F, B*, and back to *A*, as shown by the full line arrow, Fig. 2. The large blade will move as shown by the curves *a, c, e, x*, Fig. 2, and will stop at *x* when the point of the small blade stops at *A*. The small blade is again placed at *A* and the large blade at *a*. Now, tracing the outline as shown by the dotted arrows, Fig. 2, in the direction *A, D, C, B, F, E, D*, and back to *A*, the large blade will move as shown by the dotted curve *a, c' f*, Fig. 2, and stop at *y* when the small-blade point stops at *A*. The distance between *x* and *y* is called the altitude, and one-half of it, multipled by the length of the instrument, as shown, gives the area of the figure.

A hatchet planimeter can be made by bending a piece of wire into the shape shown in Fig. 3. One end of the wire is flattened and filed to a sharp edge and the other end to a sharp point. To give accurate results the edge should be in line with the point, which can be tested by drawing a straight line and placing both point and blade on the line. When the point is moved along the line, if the blade continues to follow the line, the instrument is correct.

In using the hatchet planimeter, the motion of the blade should not be controlled by the operator, but should depend only on the movement of the point. This is easily done by holding the instrument with the thumb and second finger at the bend above the point and pressing down with the first finger about an inch from the bend.

By tracing the diagram in both directions, any error in choosing its center of gravity, or in the alignment of the instrument, is compensated.

Fig. 3. Hatchet Planimeter

When the greatest dimension of the diagram does not exceed two-thirds of the length of the instrument, the result will generally be accurate to within 2 per cent., if the work has been carefully done. The planimeter should be held as nearly vertical as possible and the blade should be kept sharp.

The accuracy of the instrument can be checked by using it to measure a circle or other figure of known area.

C. A. Gilson.

East Lansing, Mich.

CO_2 Recorder Salesman's Mistake

Recently a salesman 'phoned me requesting permission to take readings with his CO_2 recording apparatus as he had not take any from a down-draft furnace such as ours and would be pleased to note results. We permitted him as we were anxious to know the percentage of CO_2 we were getting. He talked of CO_2 and conditions of furnaces in other plants which interested me in his recorder.

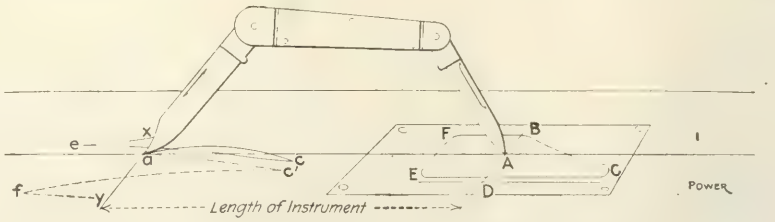


Fig. 1. Showing How Knife Planimeter is Used

We have nine boilers of about 2000 hp. and tried a 300-hp. boiler, but the percentage of CO_2 was very low, owing to some baffle bricks being broken over the tubes. It was suggested that repairs be made, after which the taking of readings could be continued. This was done but the new brick did not fit well, so our readings were still very low. Then we thought to cover the top of the brick with fireclay and put some asbestos under them, as this would stop all the air leaks at this point.

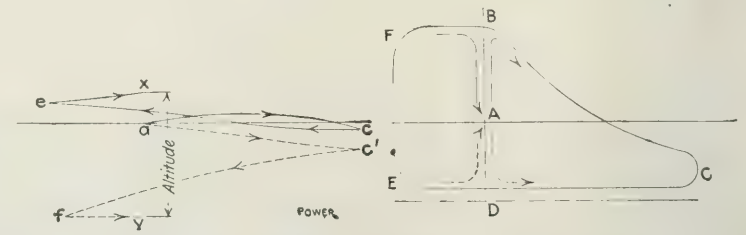


Fig. 2. Curves Made with Knife Planimeter

We were ready for the third readings, but they have not yet been taken. Instead I was called in the office by the manager who informed me that a letter from the firm that sold us the boiler we tested for CO_2 stated that from information received they believed our firemen were "green" men and were breaking the firebrick over the tubes, causing low furnace efficiency. This was untrue, as these bricks could not be broken with the fire bars and the men were not "green" but of good character and ability.

How did the boiler firm know these bricks were broken? Their representatives were not there to see them, so we concluded and later proved that our

friend the CO_2 man peddled the misinformation.

We had faith in this man and did him a favor; furthermore, we became so interested in his recorder that we intended purchasing one. But on account of his indiscretion he could never sell recorders or anything else to us now. A salesman should remember that it is the engineer after all that has the "say so" about buying material or apparatus and he should aim to treat him right.

JACOB SWARTZ.

Philadelphia, Penn.

Overflow Oil from Turbine Step Bearings

Engineers in charge of vertical Curtis turbines know from experience the disagreeable mess resulting from a "slush" of oil out through the bottom seal pipe. These so-called slushes are usually the

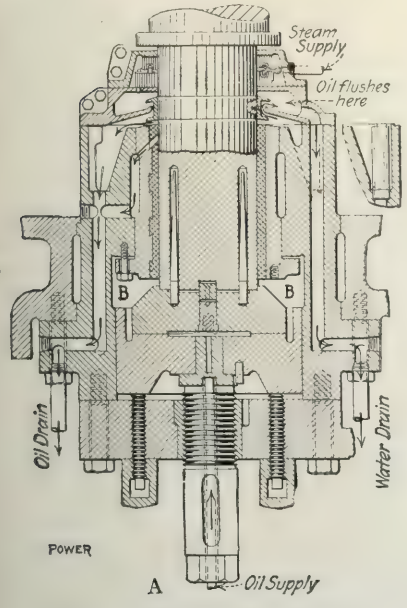


FIG. 1. TURBINE STEP BEARING

result of an oversupply of lubricant to the step bearing. Referring to Fig. 1, the oil enters at *A*, finds its way through the step bearing to the chamber *B*, and works up along the shaft and out through the oil drain, as indicated by the arrows at the left. An oversupply of oil is likely to find its way into the water drain, as indicated by the arrows at the right.

Were the return pipe large enough to take care of such an oversupply, no slush would result; but the pump speed cannot always be adjusted finely enough under all conditions to avoid these occurrences.

A vertical Curtis turbine is to be started. It is cold and the oil is cold also. After the large oil cup at the top has been set to feeding, the oiling system turned on, the governor well oiled, the swivel also, and so on, with the governor links down to the valve-gear cams, the hydraulic valve control in action, etc., we are ready to start the oil pump. Crack the steam throttle, start the pump moving slowly and observe the oil pressure rising on the gage. If 200 lb. is usually carried, some 125 lb. may now register, with the throttle just cracked.

Give it more steam, until you see 200 lb. oil pressure. Then slow the pump down gradually slightly and 200 lb. may still be indicated on the dial. Slow it down again, but be careful not to overdo this, for you must be certain that the gage registers no less than the proper pressure, or the step-bearing surfaces may be injured by rubbing.

The main throttle of the turbine may now be cracked open. You will see that the turbine is turning by glancing at the governor at the top. By being allowed to run slowly it gradually heats up equally in all its steam chambers and parts. It is advisable to let it run thus for not less than 10 min., after which it is ready to be set to work.

To return to our oil slush, let us assume that the machine is running at full speed with only half the usual load. The oil, being comparatively cold, does not return as readily to the filter as later when heated. It may be necessary to speed up, or even to slow down, the pump slightly, for right here is where keen judgment must be exercised in adjusting the pump speed to maintain the 200 lb. of step-bearing pressure, and simultaneously avoid an oversupply.

The overflowing of oil takes place at the steam seal of the gland, as shown in Fig. 1, and continues for several minutes, during which the attendant is kept dancing between the Scylla of an insufficient oil pressure and the Charybdis of a slush. It gradually diminishes and finally ceases as the fluid warms up and takes its regular course through the return pipe to the cooler and filter.

On some turbines the seal is turned upward, instead of downward, and the overflowing oil runs into the vacuum chamber and continues on into the condenser, instead of out the seal pipe; where adequate means are not provided to trap the oil, if the condensation water is used for boiler feed, the oil eventually finds its way into the boilers.

I take care of the escaping oil by means of the arrangement shown in Fig. 2. The leakage drips into the can, and the oil and water separate by gravity. The oil is drawn off by the cock *C*, and returned to the system; the water by the cock *D*, into a drain which leads to the brook. If it should be neglected until the level rises to *A*, the contents will siphon over to the drain, the water going over first.

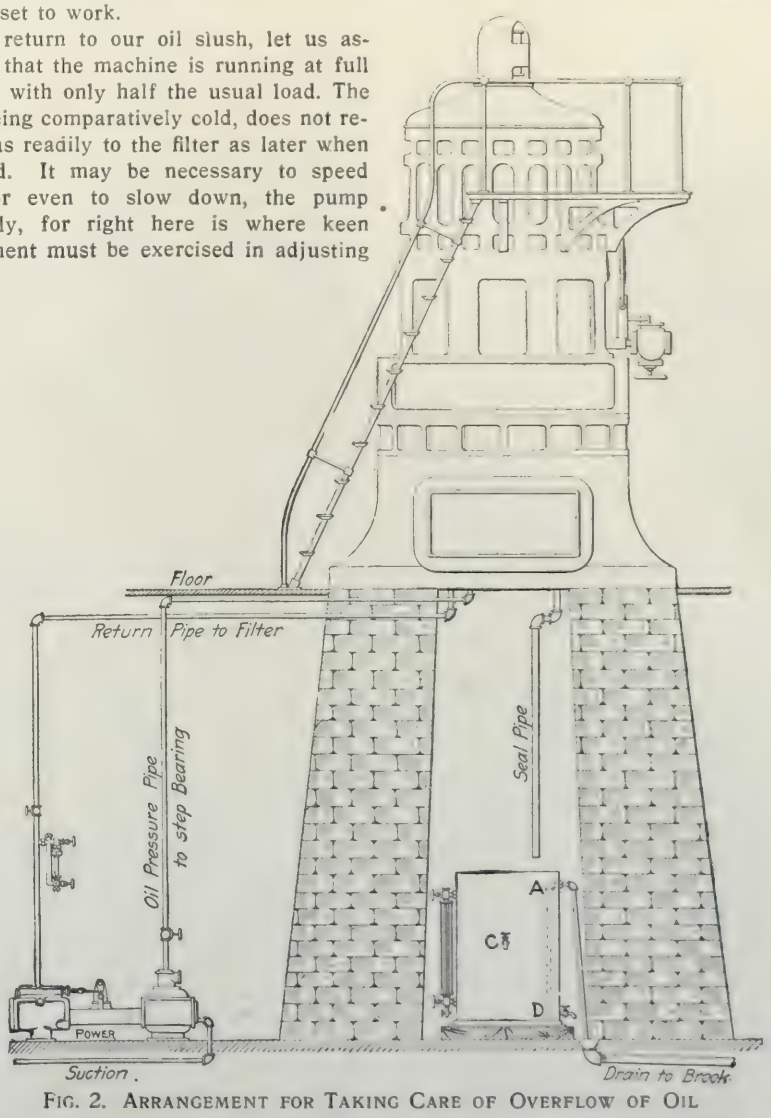


FIG. 2. ARRANGEMENT FOR TAKING CARE OF OVERFLOW OF OIL

LUKE MARIER.

Fall River, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Efficiency of Joints in Boiler Shells

H. L. Strong's article under this title in the May 14 issue, is an excellent treatment of the subject which is the kind POWER readers appreciate. I cannot understand, however, why Mr. Strong did not give the analysis of well designed joints instead of poor ones.

The joints mentioned by him have efficiencies of only 42.9 per cent. for the single-riveted lap joint, 64 per cent. for the double-riveted lap joint, 75.9 per cent. for the triple-riveted butt-strap joint and 84.5 per cent. for the quadruple-riveted butt-strap joint. With well designed joints he could have obtained 60, 70, 86 and 94 per cent., respectively, for the above joints.

When he figured the bursting pressure of shells having these joints he could have given his readers accurate and up-to-date information, and still show the method of calculating the efficiencies of the different joints.

F. P. SOMERS.

Cleveland, Ohio.

F. P. Somers' commendation of my article is gratifying, coming as it does from one who is familiar with the subject. There is a little error in Mr. Somers' letter. The efficiencies of the joints as calculated were 42.9 per cent. for single-riveted lap seam, 64.3 per cent. for double-riveted lap seam, 75.9 per cent. for double-riveted butt-strap joint, 84.5 per cent. for triple-riveted butt-strap joint and 93.75 per cent. for quadruple-riveted butt-strap joint. It will be seen that the last joint is within ¼ of 1 per cent. of the value which he suggests as good practice for that type of joint. I also stated that this joint was of standard design.

I gave the rules, demonstrated how and why they were used, and gave the American Boiler Manufacturers rules for designing joints. I then used haphazard designs, first to impress upon the minds of interested readers the whys, wherefores and results of improper designs, and, second, so that those familiar with the subject could not say that it was "copied from a book." Not long ago I heard an engineer say: "Why go to so much trouble in figuring joints; I've got a rule that beats that because it gives the same result and is shorter."

The rule he referred to, and advocated using on all occasions, was the short rule which I mentioned. I have supplied the necessary fundamentals for calculating the efficiency of joints, and those interested enough to read the article carefully should find it of value.

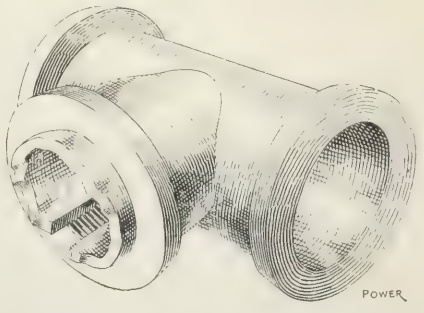
H. L. STRONG.

Portland, Maine.

Removing Broken Pipe Ends

A. T. Kasley gives a good suggestion on this subject in the Apr. 30 issue, and reminds me of the following way of doing this work.

I have found a cape chisel, machinist's hammer and a common hacksaw blade all



BROKEN PIPE-END SPLIT FOR REMOVAL FROM FITTING

that is necessary. Make two cuts ¾ in. apart in the pipe end with the hacksaw, sawing nearly through the pipe. With the hammer and chisel remove the ¾-in. strip, as shown in the illustration. The remaining ends can be curled into the center of the fitting and removed very easily.

CARL A. EVERETT.

Mescalero, N. M.

Careless Pipe Work

W. H. Wakeman's letter in the July 9 issue, under this heading, reminds me of a 150-hp. boiler installed in our plant in 1900. The feed-water piping was of extra-heavy pipe, and the only good joints were the flanged ones. The pipe was screwed up even with the face of the flange, but all the other joints were screwed in the fittings with only two and three threads holding. The joints in the water column were put together in the same careless manner. This work was done by a supposedly good firm considered to employ reliable men.

We discovered these defects about two years ago when the boiler was reset and all the piping renewed. The old piping was used steadily for 10 years, only being out of service while the boiler was shut down for cleaning. If any of these joints had failed the fireman would undoubtedly have been fatally scalded. We heat our feed water to 212 deg. F.; the pressure carried is 100 lb.

HARRY E. KOFFEL.

Doylestown, Penn.

A Contemptible Fraud

The editorial on the above subject in the June 11 issue is worthy of comment. In nearly every large power plant just such tricks occasion great annoyance, but to catch the guilty shirk is difficult.

As long as there is sufficient steam to keep the plant running, no attention is paid by the man in charge of the boiler room to who makes the steam, and only too frequently some good fireman will quit work without any apparent reason, knowing only too well the futility of preferring a charge of neglect against his co-worker. Generally a complainant is soon marked as a kicker and incurs the enmity of his fellows as well as the chief; for this reason most men prefer to leave rather than get into trouble with others.

I have seen this trick played with the blower valves and the feed valves and with all kinds of bad fires, and it is only detected when the good fireman lets the steam pressure drop to a point noticeable enough to attract the chief's attention. Even then the cause may not be found until this shortage of steam has become very annoying. Then the chief will scout around the boiler room when his presence is not known. But the shirk is usually clever enough to avoid immediate detection. By observing the amount of water evaporating in each boiler, the perfidious workman is soon discovered.

In large plants it is common for a fireman to leave an exceedingly poor fire when making the last shake, prior to being relieved. This compels his relief to start work at a great disadvantage, especially if the fuel is poor and the load heavy. It is the engineer's duty to eliminate this practice in his plant, even though every guilty fireman has to be discharged.

R. A. CULTRA

Cambridge, Mass.

Clearance by the Logarithmic Diagram

In a recent issue of POWER you had an article on the use of the logarithmic diagram for determining the conditions in a steam engine as shown by the indicator card, and in this article you asked for reports on results obtained by this method. I have used this method on two separate engines, one a single-cylinder simple engine and the other a tandem-compound engine. I am sending you with this letter blueprints of the curves obtained from each engine.

The curves from the simple engine, Fig. 1, seem to show that unless the engine has a tight piston the clearance cannot be estimated from these diagrams. The clearance of this engine is not known and the engine is known to have some leakage by the piston. This diagram was plotted assuming clearances from 6 to 26 per cent. and an attempt made to find out the true clearance from the shape of the compression and expansion lines. The compression lines seem to show the clearance as about 18 to 26 per cent. and the expansion lines show it from 6 to 8 per cent., that is, the lines are straight in these regions. As the compression and expansion lines are not straight for the same per cent. of clearance it seems to me that this shows a leakage and also, if there is such leakage, that the logarithmic diagram cannot be used to estimate the clearance. I am also sending the indicator card from which this is plotted.

The diagram, Fig. 2, plotted from the high-pressure crank-end card from a compound engine which we know is tight, seems to show a clearance of 1½ per cent. This seems to strengthen the statement in the previous paragraph that the cylinder must be tight, as there was no trouble in estimating the clearance in this engine. Due to the small clearance the diagram could not all be put on the paper at hand. I have also inclosed the indicator card from which this plot was made.

C. W. WILSON.

New Britain, Conn.

The new analysis of the indicator diagram given by J. Paul Clayton is bound to mark an epoch in power-plant work and engineers will do well to preserve for ready reference the very able analysis of his paper which appeared in POWER, June 18.

For the application of the new method it is essential that the clearance of the engine be accurately known and the proper determination of this constant of the engine requires considerable time and skill at best, while under engine-room conditions it is sometimes well nigh impossible.

Manifestly the best time to measure clearance is when the engine is on the testing floor of the factory. Here laboratory conditions prevail to a certain extent and men are available who are skilled in such work. Hereafter no engine, steam or gas, should be allowed to leave the factory without an accurate clearance determination, the result to be recorded at the factory and certified to the buyer, or better yet stamped on the name plate. This will only come about through the insistance of each buyer that he be furnished with this information, information which has now become essential in the uptodate management of a power plant.

It is also necessary for the man buying a gas engine to insist that his engine be properly equipped for indicator attachment, since most gas engines of small and medium size are made with no provision for attaching either indicator or reducing motion. Fitting a steam engine for an indicator is quite simple, but due to the water jacket and the absence of a crosshead, properly equipping a trunk-piston gas engine is frequently a serious problem.

The engineer *may* be able to measure the clearance and he *may* be able to equip for the indicator, but the engine builder is better fitted to do these things than any engineer and will do them if the buyer insists.

G. W. MUNRO.

LaFayette, Ind.

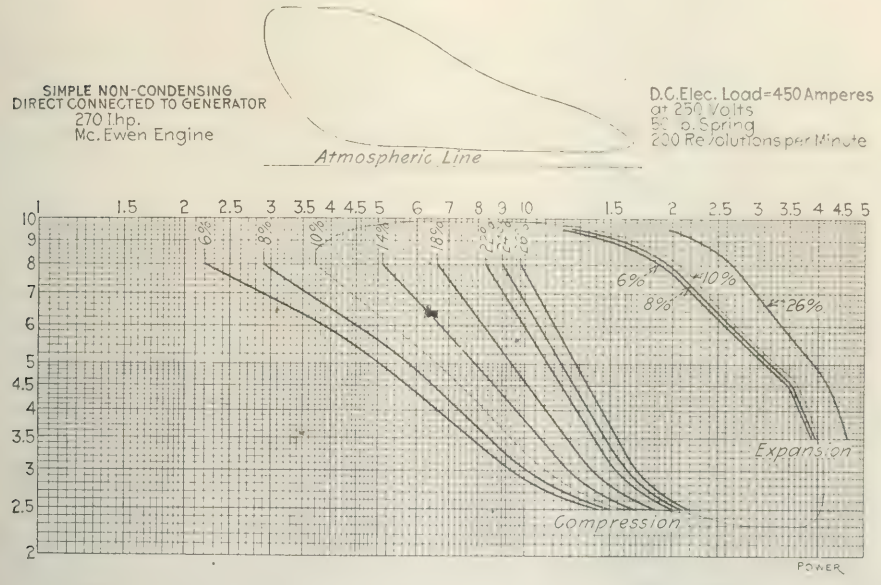


FIG. 1. SIMPLE ENGINE DIAGRAM CURVES PLOTTED LOGARITHMICALLY

During the past year, Stevens Institute of Technology has received $56,000 in gifts, as announced by President Humphreys at the recent alumni association meeting.

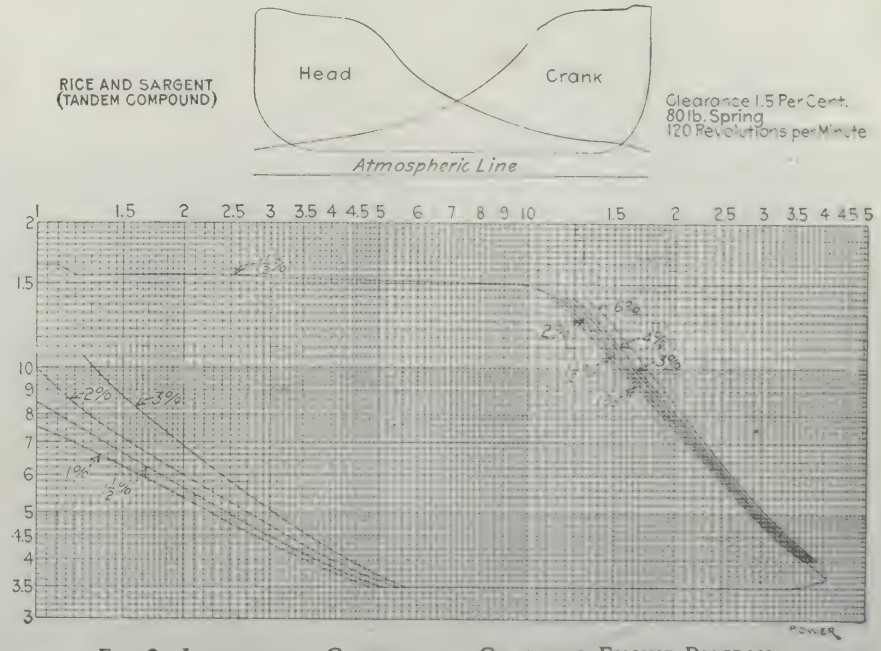


FIG. 2. LOGARITHMIC CURVES FROM COMPOUND ENGINE DIAGRAM

New Smoke Inspection Methods at Chicago

Heretofore a chimney was held to be a smoke nuisance by the Chicago smoke-inspection authorities when it emitted smoke of No. 3 density, according to the Ringelmann chart, for 7 min. during one hour, as based on the original ordinance. With this standard the owners of a chimney which emitted but a very small total quantity of smoke might be liable to punishment, whereas with a chimney which continuously emitted smoke of a density less than No. 3, the owners would be safe from legal prosecution, although the total quantity emitted might be many times as great.

In future, the total smoke emitted will be taken into consideration. Observations will be made on a given stack every 15 sec. throughout the entire day and the total "smoke units" will be recorded, from which the average smoke density for the entire period will be calculated.

A "smoke unit" is the equivalent of No. 1 smoke (Ringelmann scale) emitted for one minute. No. 1 smoke has a density of 20 per cent.; No. 2, 40; No. 3, 60; No. 4, 80, and No. 5 100 per cent. Thus, if a stack emits No. 3 smoke for 6 min., 18 smoke units are charged against it. If this smoke was emitted during one hour's observation, then

$$\frac{3 \times 6 \times 20}{60} = 6 \textit{ per cent.}$$

is the average density of smoke emitted during the period of observation.

If observations on a given stack show that the density averages more than 2 per cent., although the owner may not be legally liable, an appeal is made to his personal and civic pride by a representative of the smoke-inspection department. For example, if a certain hotel stack emits smoke of more than 2 per cent. average density, the smoke department finds a plant record of similar design and equipment, preferably a hotel plant, which shows a record well below the 2 per cent. mark. This plant is then pointed out to the owner or manager having the objectionable chimney and he is asked if he cannot do equally well when he has practically the same equipment, etc.

It has been found that this method of procedure often produces quicker and better results than a threat to go to law.

Otto Kretschmer, of the Charlottenburg Technical High School, in his plans for an unsinkable ship, embodies the principle of a hull within a hull, says the *Scientific American*. The inner body, wholly independent of the outer, contains the engines and boilers, and is walled in with steel plating through which are no doors communicating with the space between the inner and outer structures.

Indicating Hot Well

This attachment is secured beneath the condenser, as shown in Fig. 2. The opening in the bottom of the condenser is built so that the condensate drains into the left chamber of the hot well, and communication from this chamber to the hotwell pump suction is secured through an orifice in the dividing wall.

The velocity of discharge through an

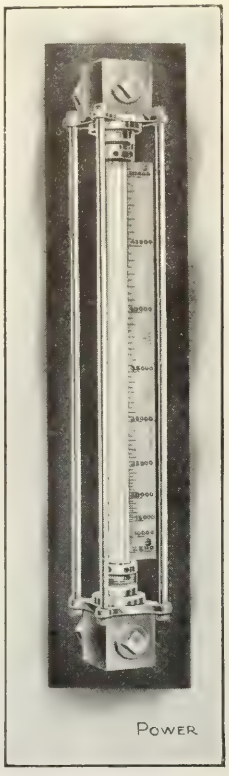


FIG. 1. THE GAGE

orifice of given diameter, varies directly as the square root of the head, and the quantity of water discharged is equal to the product of this velocity, the area of the orifice and the coefficient of contraction. With a properly designed orifice this coefficient remains almost exactly constant for widely different values of the head upon the orifice, and therefore the quantity of discharge is obtained with a high degree of accuracy by a carefully calibrated indicating gage-glass, Fig. 1, reading the head.

The orifice is formed in a brass plate inserted in the partition wall. It is polished and finished to insure accuracy of flow. Special fittings are used to attach the indicating gage to the shell of the hotwell. Ball check valves are provided in each fitting so that should the gage-glass break the inflow of air will be prevented, and the gage-glass can be replaced at leisure. A gage-glass is also provided to show the height of the water in the hotwell suction compartment, where the water must not be allowed to submerge the orifice. The scale attached to the indicating gage reads directly in pounds of steam per hour. Each orifice is made independently, carefully calibrated, and a special scale, etched upon steel, furnished for it. Over the whole range of readings the accuracy is said to be within 2 per cent. and for readings from 75 per cent. to 125 per cent. load, the accuracy is within 1 per cent.

The device is manufactured by the Wheeler Condenser & Engineering Co., Carteret, N. J.

Stumpf Engine of 2400 H.P.

There has been in use since February, 1912, at the Rombacher Iron Works, Germany, a 2400-hp. Stumpf mid-cylinder exhaust engine which is the most powerful unit of this type ever built. The cylinder diameter is 1080 mm. (43 in.), and the stroke 1300 mm. (52 in.). The engine runs at 120 r.p.m., a notably high speed for this type, giving a piston speed of 17.2 ft. per second. The regulation is exceptionally exact. No appreciable drop in speed can be detected even with the highest loads, nor can any overspeeding be discovered when the load is suddenly removed. The engine is used to drive a 600-mm. (24-in.) rolling-mill. It is coupled directly to the rolling-mill shaft and has a flywheel weighing 71 tons.

A new aërial ropeway of Potosi Department, Bolivia, reaches an altitude of 17,045 ft. It is somewhat more than two miles long and carries ore from tin mines to mills 2700 ft. lower down. The ore buckets, working by gravity, transport 6 to 8 tons an hour.

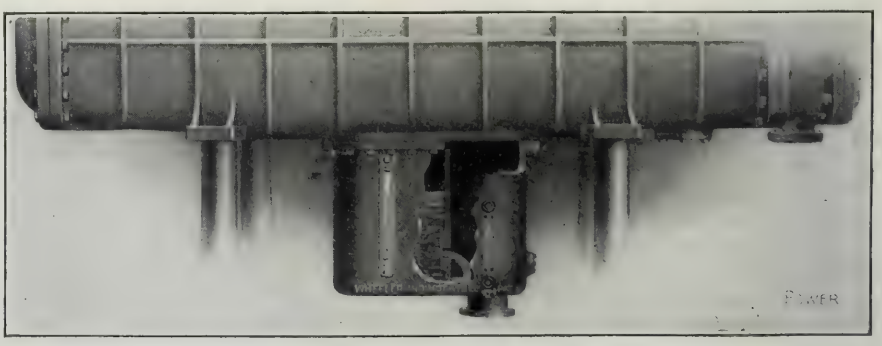


FIG. 2. WHEELER INDICATING HOT WELL

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Pumping Connected Wells

Can as much water be raised from three 2-in. pipe wells arranged in a straight line and spaced 25 ft. apart when all are connected into one suction pipe with a pump at one end, as when each well has a separate suction pump?

A. J. V.

No. In consequence of differences in frictional resistance to the flow in the well pipes and their connections to a single-suction pipe, the flow will not be the same from each well, and the most active discharge will retard the others by choking off their flow to the pump, without a proportionate increase of its own discharge.

Paint on Interior of Boilers

Would any benefit be derived from painting the interior surface of boilers?

L. P.

Yes. The interior surfaces of boilers which have been painted before any corrosion has taken place, have been known to remain free from rusting for several years, while adjacent surfaces on the same plates, left uncoated, have been found to rust and pit rapidly.

Very Small Water Power

How much power could be realized from a stream of water 18 in. wide, 6 in. deep and flowing with a velocity of 21 ft. per minute and a fall of 4 ft.

J. H.

The discharge would be $18/12 \times 6/12 \times 21 = 15.75$ cu.ft. per minute, which would be $15.75 \times 62\frac{1}{2} = 984.375$ lb., and with 4-ft. fall the discharge would develop $984.375 \times 4 = 3937.5$ ft.-lb. per minute $\frac{3937.5}{33{,}000} = 0.119$ hp. in the water. The amount of power realized would depend on the efficiency of the water wheel. An overshot wheel could be constructed with curved buckets giving an efficiency of 75 per cent. so that there could be realized about 0.09 horsepower.

Strength of Staybolts

The wrought-iron stay-bolts of a locomotive type of boiler are 1 in. in diameter and spaced 5 in. on centers each way. How much boiler pressure per square inch should they resist?

P. C.

Assuming that the stay-bolts are screw cut, with 12 V-threads per inch, they would have a diameter at the bottom of the thread of 0.8557 in. and cross-sectional area of 0.575 sq.in.

Placing the allowable load at 6500 lb. per sq.in. would make $6500 \times 0.575 = 3737.5$ lb. allowable load. Spaced 5x5 in., each stay-bolt would sustain 25 sq.in. and the allowable boiler pressure would be $\frac{3737.5}{25} = 149.5$ lb. per sq.in.

Noisy Exhaust Valves

What is the cause of the slamming noise of the exhaust valves of a Corliss engine when driving a light load and how can it be prevented?

J. B.

When the initial pressure is so high and cutoff so short that expansion takes place below the pressure into which the engine exhausts, the sudden reversal of pressure on the exhaust valve forces it from its seat. Although there may be very little play for movement of the valve, it is arrested and returned to its seat like the pounding of a check valve. On account of the large area of the exhaust port, only a very small change of pressure is sufficient to produce this annoyance. It can be prevented by carrying lower initial pressure and later cutoff, or for the high-pressure cylinder of a condensing engine, by carrying a lower receiver pressure. Where the load of a noncondensing engine is variable and it is impracticable to throttle or carry lower pressure when driving light loads, the noise can usually be stopped by joining together the indicator connections of the opposite ends of the cylinder with a stop valve in the connecting pipe nearly closed, or if not objectionable, by leaving separate indicator connections slightly open to the atmosphere.

Coal Required for Increase of Back Pressure

If the diagram from a noncondensing engine with 5 per cent. clearance shows an initial pressure of 115 lb. abs. and the cutoff is at one-half stroke, how much must the initial pressure be increased to develop the same power with the same point of cutoff, if the average total back pressure is increased 5 lb. per sq.in. for exhaust-steam heating, and the exhaust valves are set for the same increase of average back pressure from compression? How much more steam and how much more coal will be required, the feed water in each case being at 200 deg. F.?

M. C. G.

For any percentage of clearance and any point of cutoff the mean effective pressure per pound absolute may be found by the formula

$$P_m = (1 + \log_e R)(f + c) - c$$

in which

P_m = Mean forward pressure per pound initial;

R = Ratio of expansion $= \frac{1 + c}{f + c}$;

f = Fraction of stroke completed at cutoff;

c = Percentage of clearance;

where

$f = 0.5$;

$c = 0.05$;

$$R = \frac{1.05}{0.5 + 0.05} = 1.90909$$

The hyperbolic logarithm ($\log_e$) of 1.90909 = 0.6466.

Substituting these values in the formula:

$(1 + 0.6466) \times (0.5 + 0.05) - 0.05 = (1.6466 \times 0.55) - 0.05 = 0.8556$

Therefore to reëstablish the same total mean effective pressure after increasing the average back pressure 5 lb., the initial pressure of 115 lb. abs. would have to be raised $5 \div 0.8556$ lb. = 5.84 lb., i.e., it would have to be increased to practically 121 lb. The density of steam at 115 lb. pressure abs. being 0.2577 lb. per cu.ft., while the density at 121 lb. pressure abs. is 0.2705 lb. per cu.ft., then as the point of cutoff would be the same in both cases, the same number of cubic feet of steam would be used, but the weight, 0.2705 — 0.2577 = 0.0128 lb. more per cubic foot would be required; i.e., the percentage $(0.0128 \times 100) \div 0.2577 = 4.967$ per cent. more steam. As one pound of steam at 115 lb. abs. raised from feed water at 200 deg. F. requires (1188.8 — (200 — 32) = 1020.8 B.t.u., and for 121 lb. abs. would require (1189.7 — (200 — 32) = 1021.7 B.t.u., then each pound raised to the higher pressure would require $(1021.7 - 1020.8) \times 100 \div 1020.8 = 0.088$ of 1 per cent. more heat. The 4.967 per cent. more steam would therefore require $104.967 \times 100.088 - 100 = 5.059$ or about 5 per cent. more heat and consequently that much more coal.

Study Questions

This Week's Questions
Last Week's Answers

(61) What will be the temperature rise, neglecting radiation, of 1 cu.ft. of water containing a coil of 5 ohms resistance through which a current of 22 amp. flows for 15 min.?

(62) The heat required to evaporate 1 lb. of water from and at 212 deg. F. is 970.4 B.t.u. How much water will be evaporated by 1 ton (2000 lb.) of coal of a heat value of 14,000 B.t.u. per lb.?

(63) What horsepower is transmitted by a shaft carrying a 14-in. pulley running at 180 r.p.m. when the tension in the tight side of the belt is 600 lb. and in the loose side 300 lb.?

(64) If a flywheel weighs 7000 lb. and has a mean radius of 5 ft., how much energy is stored in it when revolving at 70 r.p.m.?

(65) Suppose, after measuring a number of indicator diagrams, it is discovered that the planimeter was set for a diagram length of 3 in., when the actual length of the diagrams was 3½ in. The average horsepower as found was 435 i.hp. What should it have been?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(56) Let x = height of stack.

Then $x - 62$ = distance of man from top and $100 - (x - 62)$ = distance from top to ring $= 162 - x$.

By the relations of the sides of a right triangle

$$(162 - x)^2 = 54^2 + x^2$$

$$26{,}244 - 324\,x + x^2 = x^2 + 2916$$

$$324\,x = 23{,}328$$

$$x = 72 \text{ ft.}$$

(57) The formula for the Wheatstone bridge measurement of resistance is

$$x = \frac{r_1}{r_2} R$$

From the problem then

$$x = \frac{10}{100} \times 343 = 34.3 \text{ ohms}$$

(58) The heat consumed would be merely the equivalent of the work done, or

$$\frac{2000 \text{ lb.} \times 500 \text{ ft}}{778 \text{ ft.-lb.}} = 1285.35 \text{ B.t.u.}$$

(59) The uniform load may be considered as acting at the center of gravity, therefore, having an arm of 7½ ft. about one support. The concentrated loads about the same support have arms of 5 and 10 ft. respectively. From the summation of moments then

5000 lb.	×	7½ ft.	= 37,500
3000 lb.	×	5 ft.	= 15,000
4000 lb.	×	10 ft.	= 40,000
			92,500

is opposed to the reaction of the other support acting about an arm of 15 ft. Therefore

$$\frac{92{,}500}{15} = 6166.66 \text{ lb.}$$

is the weight on one support. The weight on the other support might be found similarly by taking moments about the support just figured, but it is quicker to subtract the latter from the total load as the sum of the loads must equal the sum of the support reactions. Therefore

$$5000 + 3000 + 4000 - 6166.66 = 5833.33$$

is the load on the other support.

(60) The conditions are the reverse of a free fall, the cannon ball being retarded by gravity. Therefore the formula applies

$$s = \frac{v^2}{2g}$$

where

s = Height;
v = Velocity;
g = Acceleration of gravity = 32.2.

Substituting

$$s = \frac{1000^2}{64.4} = 15{,}527 \text{ ft.}$$

Big Power Plant at Providence

One of the features of the electrification of the New York, New Haven & Hartford R.R. between Boston and Providence will be the erection of a big power plant at that city. The railroad announced recently that work on the electrification of that section of the system would be begun next fall and that it will mean an outlay of between $6,500,000 and $7,000,000.

It is proposed to operate this line by a single generating plant to be established at Providence, to supply the current for both southbound and northbound trains. The entire electrification will be overhead trolley and single-phase alternating current, corresponding to the type employed on the New York-to-Stamford end.

Toxaway Water Power Sold

W. E. Moore, according to the Asheville, N. C., *Citizen,* has purchased the Toxaway water power for $400,000. Much attention has been attracted to the report on water power and forests recently issued by a committee of the Western North Carolina Association. The fact that there is 20,000 hp. ready for use simply by laying conduits or pipes, as expressed by the engineer, means that there will be rapid development of the water power from Toxaway.

Education for Self Supporting Boys

Chattanooga Institute of Technology has arranged with manufacturers of that city whereby engineering students are enabled to earn all expenses while attending school. The time required for completing the course is 60 months.

The student works in some industrial establishment for one week and attends the institute the next week, thus devoting half his time to theory and the other to practice while he is obtaining wages enough to pay all of his expenses. The president of the institute is Prof. H. E. Bierly, Chattanooga, Tenn.

Three Cents per Kilowatt Hour

In July last, a schedule of rates to be charged by the Winnipeg municipal plant was prepared, the same being 25 per cent. under the rates charged by the private company, which rates had been set at 10c. per kw.-hr. with a 10 per cent. discount at the time of initiation of service of its own hydro-electric plant in 1906.

Since that date, the municipal authorities have taken a step which is perhaps hard for an engineer to consider warranted—that of reducing the rate for residential lighting to the very low figure of 3c. per kw.-hr. This rate was instituted in December, 1911, and was immediately met by the private company.

The chief argument which prevailed in the making of this decision was that of endeavoring to compel the citizens to become customers and supporters of the municipal institution, the City Council assuming that earnings on similar low rates would make the plant self-supporting when the present equipment should be loaded to full capacity.—*Engineering News.*

Flood Shuts Down Power Service

Flood damage, according to a press report, caused a $200,000 loss at Wausau, Wis., on July 24, by the breaking of two dams on the Wisconsin River. Three bridges have been washed out and a fourth partly destroyed. All the electric power plants and the street-car service were put out of commission.

Alligator in Wrong Tank

At the power plant of the Struthers-Wells Machine Co., Warren, Penn., on July 18, one of the big gas engines stopped and 400 men were afforded a two-hour rest. The experts investigated the cause of the shutdown, and in a short time discovered that an alligator had been pumped into the water tank and was stuck fast in the engine.

Minnesota N. A. S. E. Convention

On June 17, the third annual Minnesota state convention of the National Association of Stationary Engineers, in session July 17 to 20, convened in Philharmonic Hall, Winona, Minn., amid a representative gathering of delegates, visitors and ladies.

James F. McCoy, of Winona, chairman of the convention committee, made the introductory address, followed by the invocation of the Rev. Frank Doran, pastor of the First Methodist Church of Winona, Mayor J. R. Schroth's address of welcome was responded to by Jesse M. Williams, Minneapolis No. 2.

Hon. James A. Tawney, former chairman of the appropriations committee of the House of Representatives, spoke on the principles of universal peace. Mr. Tawney was an engineer before he studied law and entered public life, and his address was roundly applauded. National-Secretary Raven reviewed the work of the National Association of Stationary Engineers and the steady progress it has made.

An informal reception was tendered the visitors in the evening and an inspection was made of the 40 booths of the Central States Exhibitors' Association, arranged on both sides and in the rear of the convention hall.

On Thursday, with State-President Frank J. Strieff presiding, Charles W. Naylor, of Chicago, reviewed the history of the association and talked interestingly on the work under way in Illinois. He praised Minnesota for having acquired a state license law and spoke of the efforts being made for a license law in Illinois. He predicted that public confidence in the association would continue. In his annual address, President Strieff advocated several changes throughout the state. The treasurer's report showed that the total yearly receipts were $515.35; the expenditures $326.60, leaving a balance of $188.75.

The afternoon session developed several interesting discussions and a brief talk by National-Secretary Raven. Messages of regard were received from President E. H. Kearney and Mayor Herbert Keller, of St. Paul. The session adjourned at four o'clock.

During the session the Central States Exhibitors' Association had prepared a "kangaroo court" in the convention hall, with a judge, policemen and a prosecuting attorney. "Judge" Raven fined the prisoners from 50 cents to $5 each. The delegates entered into the fun like schoolboys and when the day was over nearly $100 had been secured for the ladies' auxiliary. The delegates' ladies took an automobile ride during the day and the evening was spent with the exhibitors in the hall.

On Friday morning the routine business was transacted. At the afternoon session, Gen. James H. Harris, of Chicago, lectured on "The Relationship of the Engineer to his Employer." W. A. Converse, of Chicago, followed with an address on "Some Boiler Troubles Attributable to Feed Water." An address was also made by W. L. Osborne, secretary of the Central States Exhibitors' Association, for which he was tendered a vote of thanks.

Election of Officers

Joseph B. Craine, Duluth, was elected president; H. F. Mueller, Minneapolis, vice-president; Secretary James McGeary, St. Paul, was reëlected, as was Treasurer J. Albert Johnson; Ivan Frator, Markato, conductor; James F. McCoy, doorkeeper. Fred Kellerhals, John McIver, and William Eldred, trustees, to serve three, two and one years, respectively. The convention recommended T. S. F. Hayes as state deputy.

The entertainment Friday was a sail down the Mississippi River to Homer, Minn., to visit the government fish hatcheries. In the evening the delegates, ladies and visitors were the guests of the Central States Exhibitors' Association at a banquet and "booster" meeting, held in the parlors of the Hotel Winona. The national and state officers were present and made short addresses.

On Saturday, all unfinished business was concluded, the report of the credentials committee was read and the mayor and merchants of Winona were thanked for their hospitality. Duluth was chosen as the meeting place of the 1913 convention.

The big social feature of the convention was the industrial parade. Winona responded by turning out part of the fire department, the street-sprinkling wagons, and all the members of the city council were in the parade. The Pioneer Traction Co. paraded several automobiles and traction engines which created considerable interest. Thousands of people lined the streets as the long procession swept by.

Members and Visitors Attending the Minnesota Convention

The Exhibitors

An excellent showing was made by the Central States Exhibitors' Association, and no detail for the entertainment of the delegates was overlooked. The following members exhibited:

American Steam Gauge & Valve Mfg. Co., V. D. Anderson Co., Crandall Packing Co., Dearborn Drug & Chemical Works, Garlock Packing Co., Greene, Tweed & Co., Hawk-Eye Compound Co., Hills, McCanna Co., Home Rubber Co., Jenkins Brothers, H. W. Johns-Manville Co., Keystone Lubricating Co., Lunkenheimer Co., Osborne Valve & Joint Co., William Powell & Co., POWER, Schaeffer & Budenberg Manufacturing Co., C. E. Squires Co., United States Graphite Co., Quaker City Rubber Co., Western Supply Co., Crane & Ordway Co., Robinson, Cary & Sands Co., R. B. Whitacre & Co., Viscosity Oil Co., Kunz Oil Co., W. S. Nott & Co., Enterprise Machine Co., Bryan-Marsh Co., William Bros Boiler Works, Hanks Price, Yale & Towne Manufacturing Co., Lyons Boiler Works, Norgaard Soap Co., J. Sherman & Sons, H. C. Witt, T. J. Green, Winona Boiler Works, Standard Oil Co., R. D. Cone Co., A. M. Ramer Candy Co., Union Fibre Co.

Power Company Wins

In the appeal of the Toronto & Niagara Power Co., to the Privy Council, at London, England, the Judiciary Committee has reversed the judgment of the Court of Appeals of Toronto. The Council finds that the company is entitled to erect poles and string wires along the streets of North Toronto for power distribution without the consent of the city corporation.

The company recently decided to devote $3,000,000 to extensions of its system at Niagara Falls, making its total capacity 125,000 hp.

The proposition over which the interested parties came into conflict was the erection of an 85,000-volt transmission line from Niagara to Toronto.

Canadian Stationary Engineers' Convention

On July 23, 24 and 25, at Belleville, Ont., was held the twenty-third annual convention of the Canadian Association of Stationary Engineers. The headquarters were at the Hotel Quinte, and the business sessions were held in the council chamber of the City Hall. There were many visiting engineers and guests and their ladies. In the main auditorium was the exhibits display, and the decorations were tastefully and effectively arranged.

Executive President William Norris presided over the opening session on Tuesday morning, and Mayor Vermilyea welcomed the convention to Belleville, A. M. Wickens and W A. Crockett responding. The association adjourned until 2 o'clock, after the usual announcements and the appointment of committees.

The several business sessions disposed of the work of the convention rapidly and harmoniously. Official reports showed the organization to be in a healthy financial condition, and that the prospects for further increasing the membership and strengthening the influence with manufacturers and superintendents were better than ever before in its history.

A visit was made on Tuesday morning to Belleville's steel plant. On Tuesday evening, accompanied by several distinguished government officials and other guests, the delegates enjoyed a moonlight sail on Quinte Bay. An excellent dinner was served on board, followed by a most enjoyable entertainment.

Wednesday morning the ladies were given an automobile ride through Belleville, and in the afternoon the members and ladies went to Victoria Park to witness the outdoor sports provided. A smoker and general good time were given in the exhibition hall in the evening.

Election of Officers

At the closing session on Thursday afternoon, the following officers were elected for the ensuing year: William Norris, past president; J. A. Robertson, president; C. Cosford, vice-president; A. Kastello, secretary; W. E. Archer, treasurer; F. Chesher, conductor; H. L. Bishop, doorkeeper.

The Canadian Exhibitors' Association elected the following officers: J. E. Fiddes, president; E. A. Wilkinson, first vice-president; B. M. Walthall, second vice-president; H. V. Tyrrell, treasurer; J. N. Charles, assistant treasurer; G. C. Keith, secretary; C. F. Heatherington, chairman of exhibits; C. P. Archibald, chairman of entertainment committee; C. F. Heitzmann, chairman of reception committee.

On Wednesday morning, the Ladies Auxiliary elected these officers: Mrs. A. M. Wickens, past president; Mrs. W. A. Crockett, president; Mrs. F. H. Chesher, vice-president; Mrs. A. Kastello, secretary; Mrs. C. J. Peppin, treasurer; Mrs. S. Cosford, conductor; Mrs. William Cook, doorkeeper.

During the convention, W. A. Crockett and A. M. Wickens were presented jewels, and William Cook was given a clock. The 1913 convention will be held at Owen Sound.

DELEGATES AND GUESTS AT THE CANADIAN ENGINEERS' CONVENTION

Exhibitors

Jenkins Bros., H. W. Johns-Manville Co., Goldie & McCulloch, Dart Union Co., Garlock Packing Co., the Lunkenheimer Co., POWER, Canadian Steam Boiler Equipment Co., *The Power House*,

H. L. Peiler & Co., Dunlop Tire & Rubber Goods Co., *The Canadian Manufacturer,* Chicago Surface Co., Greene, Tweed & Co., Canadian Morehead Trap Co., Peerless Rubber Manufacturing Co., Canadian Fairbanks Co., James Morrison Brass Manufacturing Co., Quaker City Rubber Co., Trill Indicator Co., F. Reddaway & Co., Boiler Repair & Grate Bar Co., S. F. Bowser & Co., Joseph Dixon Crucible Co.

Wisconsin N. A. S. E. Convention

The twelfth annual convention of the Wisconsin State Association of the National Association of Stationary Engineers was held in Racine, July 25 to 28. On Thursday evening, at 7 o'clock, the exhibition hall was officially opened by Robert E. Hills, chairman of the executive committee of the Central States Exhibitors Association. John F. Chapman, chairman of the local committee, introduced Prof. Pressinger, of Marquette University, Milwaukee, who lectured on power-plant testing. The most important item in the cost of operating the average plant, said the speaker, is the fuel bill; hence, it is essential to high economy that the proper grade of coal be selected to suit the conditions of each plant. He described how a proximate coal analysis is made and heat value is estimated by a fuel calorimeter. Stereoscopic views of several types of calorimeter were shown and the construction and operation of each type described. Flue-gas analysis, pyrometry and steam calorimetry were also discussed and boiler tests considered.

H. J. Mistele, chief engineer of the Falk Co., Milwaukee, in a highly interesting paper on power-plant costs, showed that all that is absolutely necessary in conducting a boiler test to approximate the cost of evaporation is the ordinary scales found in the average plant, and a thermometer reading to 400 or 500 deg. If a feed-water meter or an extra pair of scales is not available, or if the arrangement of the plant is such that the feed water cannot be weighed, then the feed pump can be "calibrated" by weighing its discharge for a given number of strokes when running at about the speed required under normal conditions, to feed the boiler to be tested. While one thermometer is essential to find the temperature of the feed water, a second is desirable to ascertain the quality of the steam being delivered. The speaker also showed how simple it is to measure the work being done by the engine by means of the indicator and from this and the boiler test to figure the cost per unit of output.

At the Friday morning session, Mr. Chapman introduced Mayor W. S. Goodland, who delivered the address of welcome. National Vice-President John F. McGrath in his response spoke of the birth and growth of the National Association of Stationary Engineers, and what it has accomplished.

President G. H. Wallace, of Racine No. 3, welcomed the delegates to Racine, where the state organization was formed 12 years ago. State President William Classmann told of the increasing interest taken in the state body and of its growing importance in the affairs of the association.

National Secretary Fred Raven spoke of the beneficial effects of the mechanical exhibit, now a feature of the state and national annual meetings, serving as it does, to arouse interest and attract a greater attendance.

John W. Lane, editor of the *National Engineer,* spoke of the engineer's lack of a knowledge of the value of publicity and its effect on his welfare and success, and Secretary W. L. Osborne, of the Central States Exhibitors Association, spoke briefly on the exhibit and its purpose.

State President Classmann in his opening address advocated a systematic educational program to be participated in by all the associations in the state, each association to compile and forward to the state secretary a limited number of questions relating to power-plant operation, the secretary to have them printed and distributed to all the subordinate associations for discussion and answers. He also recommended making the duration of future state conventions one day longer because of the increasing business coming up each year. After the appointment of the committee, the meeting adjourned until 1 p.m.

At the afternoon session the following committee reports were received: credentials, auditing and ways and means, the latter reporting favorably on both of the president's recommendations.

The recommendation to extend the time of the convention was voted down on the ground that it might work a hardship to many who could not spare the time and because of the additional expense to the exhibitors. The educational program was favorably considered and a committee appointed to work out the details.

After the session the J. I. Case Co. and the Mitchell-Lewis Motor Co. plants were visited. At 7 o'clock, E. W. Pfleger, of the General Welding & Manufacturing Co., Milwaukee, lectured on "The Application of Welding in Manufacturing and Repair Work." In the evening, the delegates and visitors attended the New Orpheum Theater and saw "The Show Girl."

At the Saturday morning session, Walter Krueger lectured on "Refrigeration" and Reinhart Kunz spoke on "Engineer's License Legislation."

H. S. Bowers presented a paper at the afternoon session on "Factory Heating and Ventilation" and Fred Ruck lectured on "Power Plant Records." The following officers were elected: H. J. Mistele, president; A. Gunderson, vice-president; Robert Fenn, secretary, reëlected; A. Pfaller, treasurer; J. Jensen, doorkeeper; G. H. Wallace, conductor; John Wickert, state deputy.

In the evening, the Central States Exhibitors' Association gave a complimentary smoker in the Hotel Racine dining room, at which many appropriate and witty speeches were made. Among the speakers were Fred W. Raven, John W. Lane, John Wickert, William Classmann, G. W. Wallace, C. H. Fiske, Thomas McNeil, H. J. Mistele, C. W. Naylor and Robert Fenn. W. L. Osborne was presented a gold watch and fob by the association in recognition of his work during the past year, the presentation speech being made by Fred Raven. Fred Hickey, of the Dearborn Drug & Chemical Works, sang several solos, and the entire company united in many popular songs.

On Sunday morning baseball teams composed of the engineers and the supplymen competed for a silver punch bowl donated by H. S. Bowers. After a close contest the supplymen won by a score of 15 to 14. In the afternoon, a reunion was held in Central Park at which many old acquaintances were renewed and new ones formed.

The Central States Exhibitors' Association at its business meeting on Saturday afternoon elected W. L. Osborne president and Charles Cullen secretary for the coming year.

Exhibitors

The mechanical exhibit, held in the Hotel Racine, was unusually large and interesting. There were displays by the following members:

Lunkenheimer Co., Greene, Tweed & Co., Dearborn Drug & Chemical Works, William Powell Co., Hills, McCanna Co., Hawk-Eye Compound Co., American Steam Gauge & Valve Manufacturing Co., Keystone Lubricating Co., Garlock Packing Co., Schaeffer & Budenberg Manufacturing Co., V. D. Anderson Co., Jenkins Brothers, Power, Osborne Valve & Joint Co., C. E. Squires Co., Home Rubber Co., Crandall Packing Co., J. H. Johns-Manville Co., United States Graphite Co., Lyons Boiler Works, White Star Refining Co., Viscosity Oil Co., Foster Lockwood Oil Co., Vilter Manufacturing Co., Julius Andrae Co., Perfection Heater & Purifier Co., Richardson-Phoenix Lubricator Co., O. L. Packard Machinery Co., George B. Carpenter & Co., F. Sprinkmann & Sons, Racine Electric Co., Chicago Rubber Co., S. Freeman Sons Manufacturing Co., Falls Machine Co.

The Casualty Company of America has given up boiler and flywheel insurance, its outstanding risks being reinsured by the Hartford Steam Boiler Inspection & Insurance Co.

Consults Employees Before Buying

For years it has been customary for the buyer of an engineering concern before purchasing supplies to send a note asking the opinion of all department heads, foremen and even the higher skilled employees, who usually know what is most needed.

Often, says *System,* he has been able thus to secure very valuable advice in getting the right goods at the right prices. If this procedure were more often followed before purchasing power-plant supplies, and the engineer's requisition more rigidly adhered to, conditions in both the office and the plant would be much improved.

Old Plant Again in Service

On July 22 the work of overhauling the old auxiliary steam plant of the Toronto Ry. Co., on Frederick St., Toronto, was completed. With the steam reserve plant of the Toronto Electric Light Co., it will be kept running continuously until the Electrical Development Co.'s new transmission line from Niagara is ready for service.

The old plant has not been in use for four or five years, but now that it is put in order, it is capable of generating the full rated capacity, 7000 hp., the same as the Toronto Electric Light Co.'s plant.

Receiver for Atlas Engine Works

Fred C. Gardner, treasurer of E. C. Atkins & Co., has been appointed receiver of the Atlas Engine Works, of Indianapolis, Ind. The application was made to the superior court by F. H. Wheeler and G. M. Schebler, creditors in the sum of $2387.50.

A Great Power Project

According to press reports, a mammoth combination of central Illinois mining interests is about to be effected. Francis S. Peabody, of Chicago, is behind the proposed merger. The purposes involved are a merger of all coal interests and companies in central Illinois, the erection of a big power plant at Kincaid, from which current will be taken for operating all the Commonwealth electric properties, including those in Chicago, and a merger of all or nearly all the public-service plants in the state.

It is said that Mr. Peabody is acting for the Commonwealth Edison Co., of Chicago, which has a capital of $40,000,000. The disclosure of the plan came with Mr. Peabody's purchase of the public utilities at Taylorville, Nokomis, Edinburg, Pawnee and other towns in Christian County. He has options on all but three of the 70 coal mines in Sangamon, Christian and Montgomery counties.

Engineers for the company have selected the Kincaid site for the power plant because of the availability of water. The plan includes building a power transmission line from Kincaid to Chicago. Editorial reference has been made to this project on page 196.

New Plant on Savannah River

The Georgia-Carolina Power Co. will build a hydro-electric power plant at Stevens Creek, on the Savannah River, nine miles from Augusta, Ga. At this point the river is about 2700 ft. wide and forms the boundary between Georgia and South Carolina.

The power house, about 360 ft. long, will be at the Georgia end of the 2300-ft. dam. The spillway section will be about 2000 ft. long and contain five waste gates about 5 ft. square. The average height of the dam will be 34 ft. Flash boards 3 to 4 ft. high are to be provided.

Ultimately the installation will be 18,000 kw. in ten main units, with two 200-kw. waterwheel-driven exciter units and one 200-kw. motor-driven exciter. The average head will be 27.3 ft. with extremes of 16 ft. and 32 ft. At present the installation will include five main and two exciter units.

Transmission lines will be constructed to Augusta, Ga., 10 miles, and from Augusta to Graniteville, S. C., 17 miles, making a total of 27 miles, to be operated at 33,000 volts. The generation voltage will be 2300. This work will be completed early in 1914 and the cost will be about $2,500,000.

Motors for Collier "Jupiter"

The two motors for the collier "Jupiter," now being built at the Mare Island Navy Yard, at San Francisco, were tested at the General Electric Works recently in the presence of navy and shipbuilding officials. The scheme of propelling vessels of large size with big electric motors as developed by W. L. R. Emmet, of the General Electric Co., makes possible simple control, direct from the bridge or wheelhouse, to regulate the speed forward or reverse, or stop. The turbine, generator and motors will be shipped within two months.

SOCIETY NOTES

The American Society of Heating and Ventilating Engineers announces the election of 18 full members, five associate members and three junior members.

The Verein Deutscher Ingenieure have invited the American Society of Mechanical Engineers to attend their next annual meeting in Leipzig on June 23 to 25, 1913, and the invitation has been accepted by the Council. A committee of arrangements will shortly be appointed and the society can expect an extraordinary opportunity to hold a meeting with the National Society of Germany, to make an official tour of the industrial centers, concluding with a meeting in Munich and a visit to the German Museum, of which Dr. von Miller, the president of the German Society, is also president.

PERSONAL

Waller Edwards has been appointed publicity manager of the Busch-Sulzer Bros.-Diesel Engine Co., St. Louis, Mo., succeeding A. O. Krieger, resigned.

Charles F. Uebelacker, Charles N. Black and William von Phul have been admitted to partnership in the engineering firm of Ford, Bacon & Davis, New York City.

P. W. Sothman, chief engineer of the Toronto Hydro-Electric Power Commission, will resign on Sept. 1. He will probably be retained as consulting engineer by the commission.

Charles M. Rogers is superintendent of generation for the Empire District Electric Co., Joplin, Mo. The reference to his recent appointment as combustion engineer for that company in the issue of July 16, was an error.

Nelson S. Thompson, chief mechanical and electrical engineer in the Treasury Department, will represent the American Society of Heating and Ventilating Engineers at the fifteenth International Congress on Hygiene and Demography at Washington, D. C., Sept. 23 to 28.

R. M. Harding, superintendent of lighting of the Pensacola (Fla.) Electric Co., has been appointed general superintendent of the Columbus (Ga.) Electric Co., which controls and operates the Columbus R.R., the Columbus Power Co. and the Gas Light Co. of Columbus.

Robert L. Brunet, of the Public Service Corporation of New Jersey and formerly with the General Electric Co., has been appointed city electrical and gas engineer of Providence, R. I. Mr. Brunet is a graduate of Union College and 27 years old. He will take up his new duties very shortly.

Charles F. Gray, formerly superintendent of construction for the Canadian Westinghouse Co., and chief erection engineer in charge of the installation of the company's apparatus for Winnipeg's municipal hydro-electric system, has opened a consulting electrical engineering office in Empress Block, Winnipeg. Mr. Gray had had 15 years' experience in engineering and construction work in many of Canada's hydro-electric developments, as well as in several of the large central stations in New York.

POWER

Vol. 36 NEW YORK, AUGUST 13, 1912 No. 7

Cash in Your Ideas

This is a call for help.

We want a greater variety of topics and treatment in the matter on this page, than is possible from the limited number on our own staff.

To enlist outside aid we make the accompanying offer for all accepted material.

Cartoons are particularly invited. They call for little or no text, telling their story by lines rather than words.

Any material inappropriate for foreword use but suitable elsewhere in the paper, will be so used and paid for at our regular rates.

Matter will be paid for upon acceptance, or if unavailable, promptly returned.

We will not suggest as to selection of subjects, for fear of defeating our purpose; namely, to get new views into these pages, not limited by our own range of vision.

Take past forewords for a guide, as to *what* and *what not* to send. The general character or style of treatment may be used as a pattern, but the same topics should be avoided, unless you can treat them from a new angle or bring out some new idea.

$5 for an idea for a foreword or a cartoon.

$10 for a rough sketch for a cartoon.

$15 for an unillustrated foreword.

$20 for a foreword with a sketch or suggestion for one.

$25 for a cartoon sufficiently well drawn to be reproduced directly.

What interests you, as a reader of POWER; may be expected to interest others. Keep to lines related to our field, but not technical things only. The human side should come in for its share, but don't always preach a sermon.

Just think of a foreword as an informal editorial; the rest may be left to your judgement. Above all don't be afraid to make a try. Perhaps you underestimate your own ability. We are ready to assist by touching up here or there if necessary.

Now, let's hear from you.

Station of San Diego Electric Ry. Co.

By Leroy W. Allison

A 4200-kw. plant employing a high-pressure reciprocating engine and low-pressure turbines. The latter drive their generators through Melville-MacAlpine reduction gears.

To meet the increased and growing demands of its city and interurban transportation system, the San Diego Electric Ry. Co., of San Diego, Calif., has erected a new electric-generating station at Arctic and E Sts. The plant has a capacity of 4200 kw. and is employed exclusively for railway service. In general construction and equipment it offers the highest type of the present-day generating plant and no expense has been spared to effect a commendatory result to the owners, the builders and the city. The electric traction system now comprises about 56 miles of local street railway and 18.7 miles of interurban lines to Otay, Point Loma and Ocean Beach, with other long distance lines contemplated.

The generating station, about 10 years ago, located on the same site as the new structure, had a capacity of approximately 400 kw. This consisted of G. E. generators, belt-driven by a 250-hp. tandem Corliss engine and a 250-hp. automatic cutoff American Ball unit. Five years later one 1000-kw. and two 500-kw. turbo-generators, with three new 250-hp. Keeler boilers, were installed.

Building

The former power house was demolished early in 1910 to make way for an entirely new plant, as shown in Fig. 1. The building, comprising boiler and engine rooms, is 160x170 ft. over all, with a basement under the latter room, 18 ft. high, for the auxiliary equipment. It is constructed of reinforced concrete throughout with steel trusses and roof framing. Corresponding with the imposing exterior appearance, the interior walls, columns and gallery (Fig. 2) form a noticeable contrast to customary construction. The balustrades surmounting the gallery are of solid concrete, cast from a prepared design.

The side wall of the building, adjacent to the low structure at the right of the illustration, Fig. 1, and which is now used as a paint shop by the company, is constructed in a temporary manner to allow for erecting an exact duplicate of the present plant on this adjoining property. The station will then be symmetrical about the large doorway at this point and will cover an entire city block. A standard-gage railway track enters the plant through this doorway and runs to the boiler room. The track, after crossing Arctic St., will connect with the new San Diego & Arizona Ry., a steam line now in course of construction and a subsidiary property of the Spreckels companies, which operate the electric-railway company.

Boiler Room

The boiler room is 97x160 ft. in size and is approximately one-half the height of the engine room, as will be noticed from Fig. 1. The installation comprises nine water-tube boilers aggregating 2200 hp., consisting of four Babcock & Wilcox units of 200 hp. each, two Keeler boilers of 400 hp. each and three Keeler boilers of 200 hp. each. They are arranged in three batteries of two units each and three single, as shown in the plan, Fig. 3. The boilers take up about one-half of the room, the remaining area allowing for future additions. The furnaces have Leahy oil burners and operate at a working steam pressure of 180 lb. Two self-supporting steel stacks, 8 ft. in diameter, carry off the products of combustion.

The boiler-feed pumps and similar equipment are arranged along the wall separating the boiler and engine rooms. One 14x9x24-in. Blake vertical pump supplies the boiler feed, supplemented by two 10x5½x10-in. Worthington horizontal pumps. These are connected to the boilers by a 4-in. feed line. A 12x8½x10-in. Worthington pump, working against a water pressure of 200 lb., is used for boiler washing and two 6x4x6-in. duplex pumps deliver the fuel oil at 40 lb. pressure.

An 8-in. suction line connects the boiler-feed pumps to the hotwell, and continues to a Day marine filter. The pumps are also connected by a 6-in. line to a 4000-hp. closed feed-water heater, which has a 12-in. exhaust to the atmosphere. A detail of this piping is given in Fig. 4.

Engine Room

A view of the engine room is shown in Fig. 2 and a plan in Fig. 3. As in the boiler room, only about half of the available space is now utilized, the remainder allows for duplicating the present installation.

One 28x60x48-in. cross-compound Allis-Chalmers Corliss engine is centrally located, as shown, directly connected to a 1200-kw. direct-current Westinghouse generator. Steam is supplied through a 10-in. main, which branches from a 14-in. main header in the boiler room with connection to a 10-in. steam separator. The exhaust line is 22 in. in diameter, reducing to a 10-in. line as the supply to a low-pressure turbine. This is a 1000-kw. Westinghouse machine and drives its generator through a Melville-MacAlpine reduction gear. The unit may be seen opposite the Corliss engine in Fig. 2. It operates at 3600 r.p.m. on the turbine end, and at 514 r.p.m. on the generator end. The turbine takes exhaust steam from the reciprocating unit at 16 lb. pressure absolute, and the throttle is controlled by a 42-in. gate valve operated hydraulically in the basement. A board at the turbine end of the machine is

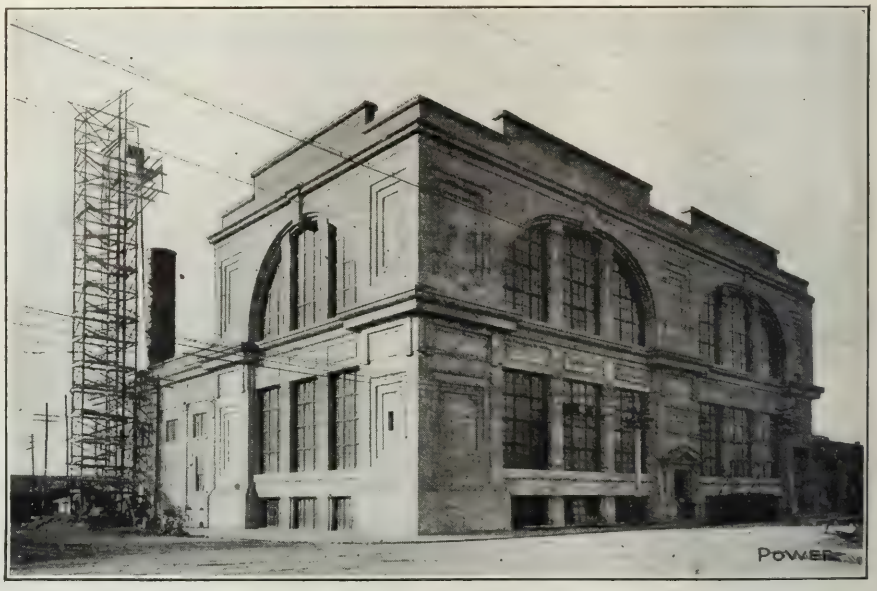


Fig. 1. Exterior View of Building

equipped with gages for steam, oil, vacuum, gland or water pressure, and a speed indicator.

As indicated in the typical load curve, Fig. 5, the turbine carries approximately 98 per cent. of the engine load during a 24-hour period, at times running as high as 1200 amp. on the machine to 1000 amp. on the Corliss unit, at a pressure of 500 volts. The turbine has been in daily service since August, 1911, giving continuously efficient performance.

Another turbine unit of the same type and size is being installed alongside the first unit, as shown in Fig. 3. Former equipment will be used to effect this change, augmented by a reduction gear. The machine will also take steam from the Corliss engine.

FIG. 2. VIEW OF ENGINE ROOM AND GALLERY

AUXILIARY ENGINE-ROOM EQUIPMENT

As shown on the plan, two 500-kw. Westinghouse motor-generator sets are opposite the new turbine installation and two similar machines will later be located on the gallery floor. One 15-kw. G. E. horizontal steam-turbine lighting set is also installed.

A 65-ton Shaw electric traveling crane spans the engine-room main bay, while a 20-ton crane is installed beneath the gallery; both run the length of the building.

SALT WATER CIRCULATING SYSTEM

Condensing water is taken from the Bay of San Diego, about 1000 ft. distant. The salt-water system is particularly interesting and has many original features.

As shown in Fig. 3. the system comprises two 60-in. lines of reinforced-concrete pipe, each 766 ft. long, with intake and terminal chambers. The two lines of pipe, in 3-ft. lengths, were laid on 7-ft. centers, affording a clearance of 12 in. The pipe was placed directly upon the sand, where the nature of the soil permitted, with the inside bottom 6 ft. below grade. For additional foundation as required, 3x3-in. mudsills of redwood were laid on 3-ft. centers, with two 8x8-in. stringers, beveled on one edge to support each line of pipe. Concrete manholes were placed at two street intersections along the route.

The intake chamber, near the bulkhead line at the foot of E St., is divided into two compartments, one for either line of pipe. These are 17 ft. high or approximately 2 ft. above the highest water line ever recorded at that point. The footing of the chamber is 24 in. thick; the interior walls vary in thickness from 24 in. at the base to 18 in. at the top, and are heavily reinforced with twisted rods. The partition wall is 12 in. thick and is similarly built. The pipes have cast-iron sluice-gate valves, operated from wheel-stands on top of the chamber. The chamber is covered on top with 2x12-in. planking and has an iron-pipe guard railing.

A channel 7 ft. wide and of the same depth has been dredged for each pipe line, beyond the intake chamber and flanked with sheet piling, and 4x6-in. strips have been placed for a distance of 50 ft. out into the bay, as indicated, terminating about 15 ft. apart. This piling provides against the bypassing of any water from the discharge to the intake pipe. On the outer side of the dredged area, a wooden grill-work, similar to a picket fence, is built to keep out seaweed and other substances.

The terminal chamber is about 29 ft. from the power station, adjacent to E St. It is 18 ft. 3 in. by 20 ft. 3 in. inside and 24 ft. high. The footing is of the same size as the intake chamber and the exterior walls vary from 24 in. at the bottom to 12 in. wide at the top. The chamber is entirely covered on top, except for a manhole section, with a reinforced-concrete slab flooring. Openings similar to those in the intake chamber are provided for the entrance of the pipe lines and sluice gates are installed to operate from a wheel-stand on a platform 17 ft. above the floor of the chamber. The terminal chamber is connected to the plant by four tunnels.

The entire system has been designed with a view to interchanging the service of the respective lines. Either line may be employed as intake or discharge. The terminal chamber has been constructed to effect this condition and also that either line may be closed temporarily for repairs, etc. The sluice gates additionally provide that the suction and discharge pipes from the plant enter the respective

sea line as desired. This alternation affords a means of keeping the line clear of seaweed and obstructions, and it is for this reason, primarily, that the concrete pipes are laid at the same level.

Through the tunnels between the terminal chamber and the power station, connection is made by two 20-in. suction lines, an 8-in. suction sump line and a 30-in. discharge line, all of cast-iron pipe. An additional 28-in. suction line is to be installed later.

The two 20-in. suction lines are operated by 26-in. Worthington horizontal centrifugal pumps, directly connected to a 125-hp. American Ball engine, both being in duplicate. The suction lines connect with three Wheeler surface condensers. The other auxiliaries include a 4-in. Worthington centrifugal pump directly connected to a 30-hp. Westinghouse motor; a pump of the same size, type and operation is used in connection with a hotwell adjoining the 5000-sq.ft. Wheeler condenser and permits its operation as an auxiliary to the condenser for the low-pressure turbine or Corliss unit.

An 18-ft. drop at high tide creates a siphon effect on the discharge end of the condenser, giving an approximate vacuum of 10 in., proportionately reducing the load on the engines driving the pumps. The circulating discharge water is maintained at 80 deg. temperature and is supplied to a neighboring bath house, as required, by an 8x8x10-in. pump.

Electrical Equipment

The switchboards are temporarily located on the engine-room floor, as shown in Fig. 3, but soon will be moved to the

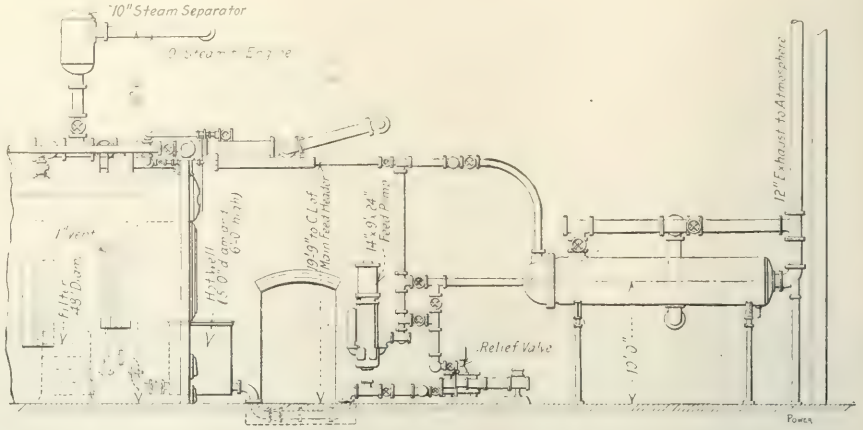


Fig. 4. Section through Boiler Room

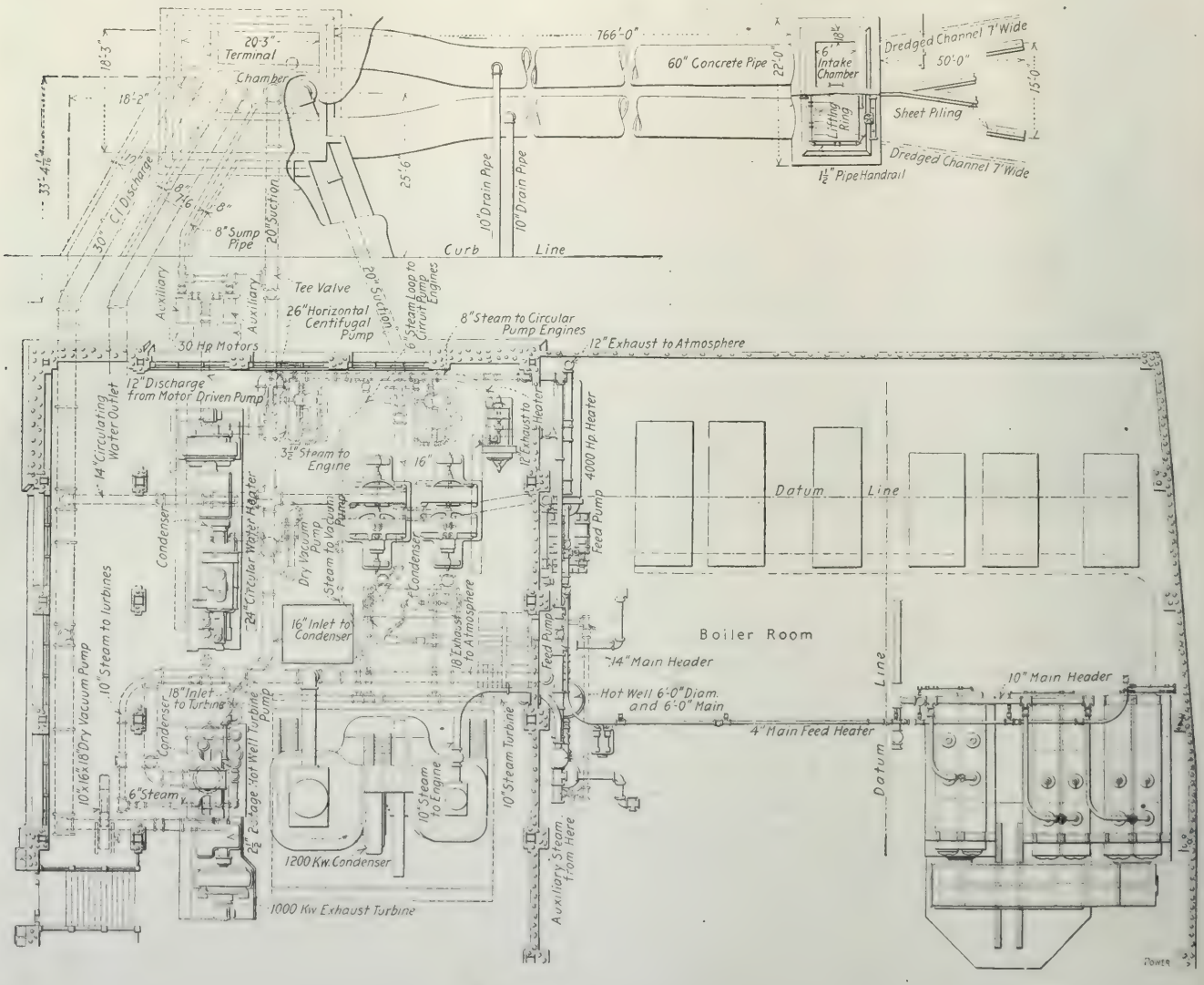


Fig. 3. General Layout of Plant Showing Engine and Boiler Rooms and Intake and Discharge Tunnels

gallery floor, which has been constructed primarily as a high-tension gallery.

The boards are of slate and comprise 19 panels in two sections. Sixteen of the panels are for direct-current service and the others for alternating current. Of the former, five are generator panels, eight, double-feeder panels, and three, single-feeder panels. On the alternating-current side, two are generator-feeder panels and the third a tie-in panel for outside service. The board is fully equipped with the usual electrical-measuring instruments of General Electric and Westinghouse type. Direct current at 500 volts is supplied for the railway service.

The outside leads, shown in Fig. 1, are temporary, and underground conduits leading from the basement, where the busbars and similar apparatus are located, are now in course of construction. To comply with a recent city ordinance the company has just completed the construction of underground conduits for its feeder wires in the business district.

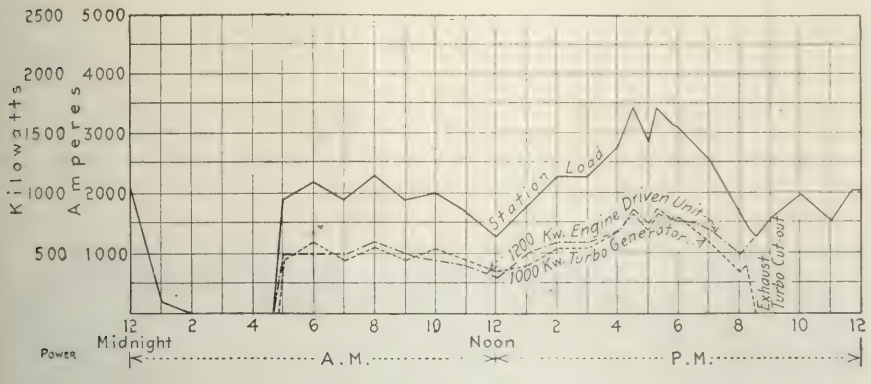


Fig. 5. Typical Load Curve

Station Auxiliaries

The plant is equipped with a Richardson gravity-oiling system and a Bowser self-measuring oil-storage outfit is installed. A blowoff tank is in the basement, with a drain to the sewer, and a 9½x9½x12-in. air pump is used for cleaning the generators.

The company is at present constructing an underground fuel-oil storage tank on the opposite side of E St. This will be of reinforced concrete with a ¼-in. steel lining; it will have three compartments and a total capacity of 12,000 bbl. A pipe line will lead directly thereto from a private wharf of the company in the Bay of San Diego, permitting it to purchase its fuel from tank steamers at an exceptionally low rate. The oil-storage tank will connect with a day storage plant at the power station.

A city franchise has been granted the company to construct an underground heating line through the business section, principally on D St., where the Union Office building, the Spreckels Theaters and the Spreckels and Hanam Hotels, the property of the Spreckels Companies, are located. As soon as this line is built these buildings will be heated by exhaust steam from the power plant.

The station has cost approximately $350,000 and the future extensions contemplated will involve a considerable sum. Much credit is due the management for maintaining regular service throughout the entire reconstruction of the plant and causing no interruptions in the transportation system.

The building was designed by Andrew Ervast, chief engineer for the company, and was erected under his direct supervision. The power-plant design and the installation of machinery was carried on under the direction of Hunt, Mirk & Co., engineers and contractors, San Francisco.

Variable-Speed Motor Drive for Stokers

The Detroit United Ry. has installed compound-wound variable-speed motor drives on all of the stokers in its main power houses, with the exception of those upon four boilers. The motors were put upon the opposite ends of the shafts from the steam engines, which were installed with the original apparatus, and the motors for two batteries, consisting of four boilers, require from 19 to 21 kw.-hr. daily.

The engines operate very nicely at fixed loads, but under certain conditions have to be slowed down so much that they get beyond the control of the governor. Then they are likely to stop when the mechanism passes the center and, unless the attendant notices it, the upper part of the grate becomes bare and incomplete combustion takes place. When the fireman comes back up the line and finds the engine stopped, he opens it wide, forcing in a lot of green coal, which produces smoke, etc. The motors under reduced-speed conditions operate very nicely.

Engine Connected up Backward

By J. H. Stratton

Last summer I was asked by the superintendent of a neighboring company to see what I could do with his lighting plant, which consisted of a small dynamo direct-connected to a vertical engine, of the shaft-governed, piston-valve type, running at 650 r.p.m. He said that he had been unable to keep a man for more than a month and had to keep a hose turned on the crankpin every night and babbitt the brasses every two weeks, the engine sounding like a rock crusher.

We drove over to the plant, and when the engine was started I fully agreed with the superintendent that it sounded like a rock crusher.

I decided to try it for a few nights and while looking things over, the superintendent gave me the history of the plant, telling me that it had been installed by one of the best electricians in the oil fields, who, after trying for three days to get the lamps to light without any visible success, had left for parts unknown. The next man finally produced light and decided that the pounding was caused by the crankshaft being bent, so he took it out and sent it to a machine shop (10 m. away), telling them to straighten it. They placed it in a lathe and found it to be straight and true and sent it back with a note to that effect, but the engineer sent it to another shop to be straightened, as he said he knew "when a crankshaft was bent and when it was not"; the second shop came to the same conclusion and sent it back, whereupon the engineer got offended and quit.

The next man said the cause of the crankpin getting so hot was lumps on the pin, so one morning after shutting down he loosened the brasses and put in some powdered emery "to grind off the lumps" and then he quit by special request.

Several others had similar experiences with the engine during its 12 months of operation and finally it was up to me. I noticed that with a steady load the voltage would vary from 10 to 15 volts when the engine was running on a wide-open throttle. The superintendent told me that the governor never did work right, as the only way a steady voltage could be obtained was by cutting out all the rheostat resistance and running the engine on the throttle.

The first night I began exploring. The exhaust sounded like that from a 100-hp. gas engine. I noticed that the engine had a direct valve motion but the steam pipe was connected to the top of the steam chest. About this time I had discovered also that the steam and exhaust pipes were the same size (1½ in.). Following up this clew, I found that the

inlet to which the steam pipe was attached, was bushed from 2 to 1½ in. This told me at a glance where the trouble was; the steam pipe had been connected where the exhaust pipe should have been and the engine was running

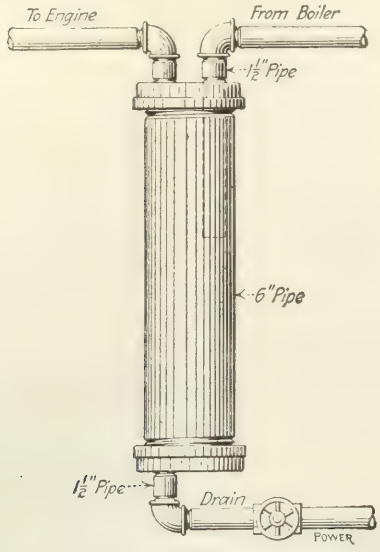


FIG. 1. DRAIN

backward, and had been doing so for 12 months.

The next morning I reported having located the source of the trouble and upon explaining it to the superintendent he laughed at me and said it was impossible, as an engine would not run with the pipes connected that way.

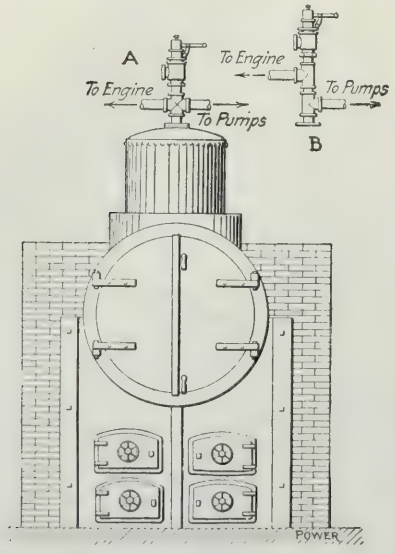


FIG. 2. ORIGINAL AND FINAL PIPE CONNECTIONS

I suggested that he walk down to the engine, whereupon I took a steam gage off one of the boilers and connected it to the drain valve of the cylinder. I knew that if the connections were wrong the gage would show more pressure on the cylinder than was on the boiler. Sure enough, when the engine was started the gage showed 180 lb., while there was only 120 lb. on the boiler and the pressure on the cylinder did not drop below 120 lb. The hand of the gage just traveled back and forth from 120 to 180 lb.

Well, I got the necessary fittings and that afternoon made the change in the pipes. I also had to change the brush-holders around as the direction of rotation was reversed.

When all was ready to start up, a number of the "oil-field mechanics" came around to see the engine refuse to start, and to have a good laugh on me, but it started off quietly and ran perfectly cool. With a wide-open throttle the voltage did not vary 1 per cent., even with large variations in the load, and the exhaust could hardly be heard when standing at the end of the exhaust pipe.

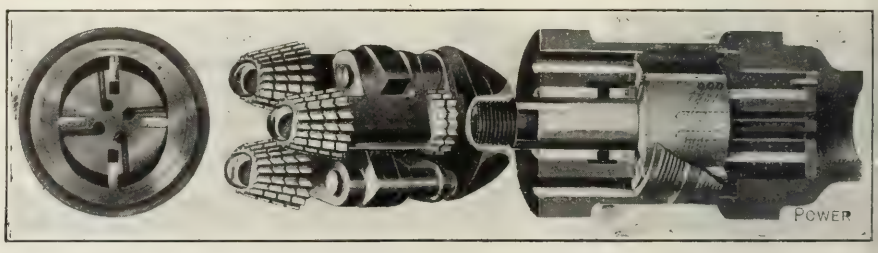
While I had the steam-chest cover off, I took the valve out and made a diagram of it and measured the parts. Then I put the valve back and turned the engine over by hand in the direction it had been running, and noticed how the different events took place. Steam had been admitted for full stroke, cutoff occurring just as the crank passed the center; it was then compressed about ½ stroke, when the release took place and it had to exhaust in the remaining half stroke. I did not have the good luck to own an indicator at that time and I shall always regret it, as a diagram from that engine would have been a souvenir worth framing.

On the second night I made some more discoveries. The boilers also supplied steam for one well and two large pumps for pumping the oil to the railroad, 10 miles away. These were started whenever the tanks were full, by an attendant who had to look after twelve or fifteen pumps scattered all over the surrounding oil fields. No warning was given when he was ready to start a pump, but when it had made about four strokes, I knew it, as the engine slowed down and stopped. I noticed before it stopped that there was a steady stream of water from the exhaust. I tried to keep the engine running by carrying just one gage of water in the boilers, but it would pull over and eventually stop the engine.

The next day I got a piece of 6-in. pipe threaded at both ends and about 5 in. long; also, two 6-in. caps, one with two holes drilled and tapped for a 1½-in. pipe, and the other for one 1½-in. pipe. These caps were put on the 6-in. pipe and the latter connected in the steam line to the engine, as shown in Fig. 1. The drain valve was left cracked, but the water came over so fast that the engine slowed down no matter what was done. Therefore, I decided to change the connection at the top of the boilers. In Fig. 2 are shown both the original and altered connections, the former at *A* and the latter at *B*. After the change was made no more trouble was experienced from wet steam at the engine.

New Lagonda Air Driven Boiler Tube Cleaner

The sectional cuts of this cleaner, shown herewith, give a good idea as to its construction. The compressed air or steam passes through two ports in a plate in the rear end of the cleaner, then through transverse openings in the rotor, and out through branch openings to the space behind the paddles. There are but two ports opening into the air chamber and only the two paddles doing the work are under air pressure.



LAGONDA AIR DRIVEN BOILER TUBE CLEANER

To secure economical operation under various air pressures and in different hardnesses or thickness of scale, two interchangeable rear plates are provided having different-sized port areas. Where a limited amount of air is available, the plate with the smaller holes can be used, securing plenty of power with a small amount of air. If the scale is heavy and plenty of air can be furnished, the plate with larger holes can then be used and more power will be developed.

The cleaner is manufactured by the Lagonda Manufacturing Co., Springfield Ohio.

It is announced from Toronto that the government of Saskatchewan has made a contract by which the Canadian Pacific Canadian Northern, and Grand Trunk railways will spend $20,000,000 diverting the South Saskatchewan River to supply water to Regina, Moose Jaw and other cities, and also in developing 30,000 electrical horsepower. A commission has been appointed to prepare plans.

Development of Large Steam Turbines

Discussion before the American Society of Mechanical Engineers of the paper by Prof. A. C. Christie on the above subject.

An abstract of this paper appeared in the July 23 issue, page 144, and an extract "Economy Tests of Steam Turbines," June 11, page 831.

In the discussion of the paper upon the present state of development of large steam turbines presented by Prof. A. E. Christie, at the recent meeting of the American Society of Mechanical Engineers, Carl George DeLaval, general manager of Henry R. Worthington, called attention to the new radial-flow reaction turbine designed by the Messrs. Ljungstrom, of Stockholm, a description of which will be published in an early issue. The steam enters between two disks, passing from center to circumference between concentric blading rings. The two disks revolve with their shafts at the same rate of speed, but in opposite directions. At the end of each shaft an electric generator is attached. With this arrangement the relative speed of each blade becomes double that in the ordinary reaction turbine of the same rotative speed and diameter with the result that the number of blade rows becomes only one-quarter as great. The general result is a turbine of small dimensions for the power and several have been and are being built. The first was a 500-kw. turbine running at 3000 r.p.m. which gave an efficiency of 71.8 per cent. and a steam consumption of 8.75 lb. per b.hp.-hr. with steam at 175 lb. absolute and 250 deg. F. with a vacuum of 28½ in. Sizes from 100 to 7500 kw. with a speed of 3000 r.p.m. have been designed on this double-rotating principle. Single-rotating turbines of the same design have also been brought out, but with slightly less efficiency and about 10 per cent. increased steam consumption.

The highest recorded efficiency is 71.8 on an Erste Brünner turbine with steam at 156.2 lb., 1500 r.p.m., 15.82 lb. of steam per kw.-hr. The Ljungstrom 1000-kw. turbine shows 74.7 per cent. efficiency, taking the steam in front of the throttle valve, and 76.9 per cent. behind the throttle valve, with steam at 162 lb., speed 3000 r.p.m., pounds of steam per kilowatt-hour 11.55, or 8 lb. per b.hp. In starting the double-rotating turbine the two sides may not run at the same speed, but when a certain speed is reached and the exciting currents of the generators are strong enough, they will automatically synchronize and run as one unit. In case of accident one generator can be left idle and the other continued in use with, of course, a loss of efficiency.

CLARENCE P. CHRISSEY, of the General Motor Co., Detroit, suggested the following classification:

(*A*) Impulse turbines with only one velocity stage in each pressure stage.

(*B*) Impulse turbines with two or more velocity stages in each pressure stage.

(*C*) Impulse turbines in which all pressure stages do not have an equal number of velocity stages.

Classification of impulse turbines might be carried further to illustrate a point of construction often overlooked; i.e., the method of conducting the steam leaving the last buckets to the succeeding nozzles. Various constructions are:

(*a*) Impulse turbines in which the steam leaving the last moving bucket passes directly from the bucket into the succeeding nozzles without great loss of velocity.

(*b*) Impulse turbines in which the steam leaving the last moving bucket passes into the casing and after losing practically all of its velocity enters the next set of nozzles.

(*c*) Impulse turbines in which the steam leaving the last moving bucket is utilized in some pressure stages, according to the methods described in (*b*), and in other pressure stages, according to the method outlined in (*a*).

It follows from this classification that

Rateau turbines fall in class *Ab*.

Zoelly turbines fall in class *Ac*.

Curtis turbines of General Electric Co. fall in class *Bb*.

Curtis turbines of A. E. G. (large size) fall in class *Cc*.

Professor Christie says that the single-row stages of the A. E. G. turbines are of the Rateau construction. The Allgemeine Electricitäts-Geselleschaft, however, states that these stages are constructed in accordance with the 1896 patent description of Curtis, which represents the immediate forerunner of the Rateau and Zoelly construction. The difference between the A. E. G. single-row stages and those of Rateau consists in the manner in which the steam leaving the moving bucket is utilized, i.e., the A. E. G. single bucket row stages are of type *a* in which the steam leaving the moving bucket passes directly into the succeeding nozzle without great loss in velocity, while the Rateau falls under type *b* in which the steam is brought to rest before entering the next set of nozzles. The large A. E. G. turbine, therefore, appears to be of the Curtis type throughout, although not altogether of the type popularly known as Curtis.

The moving buckets of reaction turbines in practice have been supported upon drums and those of impulse turbines usually upon disks. The manner of construction does not, however, change the fundamental type, and therefore the author's designation of the Belliss and Morcom turbine as a Curtis-Parsons construction is confusing because this turbine operates entirely upon the impulse principle.

The paper conveys the impression that high steam velocities are inherent to impulse turbines having two or more velocity stages. Such is not necessarily the case; if low bucket velocities are employed the spouting velocity of the steam is also low. Marine turbines of the impulse type illustrate this point.

The theoretically correct action of the steam in any impulse turbine appears only at one set of conditions. For any variation in conditions, or quantity of steam flowing, a variation from the theoretically correct action must occur. The flat economy curves of impulse turbines prove that this is not important from the standpoint of efficiency, and practice shows that the end thrust is not appreciably increased by departure from the theoretical conditions.

The drum construction in impulse turbines was probably first used by the A. E. G. in the turbine for the steamship "Keiser," which was tested in 1905. The firm's experiments led them to the conclusion that the drum construction was advantageous for the low bucket speeds existing in direct-connected marine turbines when the steam had expanded to fifty times its initial volume.

While the efficiency ratio comes nearer than any other single feature to expressing the economic results, it must be used with caution because

1. The steam conditions must be taken into account.

The same steam turbine operating with an equal amount of available energy will not have the same efficiency ratio if the energy is obtained in one case by a moderate vacuum and high superheat, and in another case by high vacuum and low superheat.

A noncondensing turbine may have the same or a better efficiency ratio than a condensing machine, but this is no indication that the noncondensing turbine is as suitable or better for a given plant than a condensing unit.

2. It is more difficult to obtain a high efficiency ratio with a large amount of available energy than with a smaller amount.

Referring to the table of tests,* it is seen that the 6000-kw. machine built by Erste Brünner, M. F. G. has an efficiency ratio of 71.3 and that the heat available per pound of steam is

*Power, June 11, p. 832.

380.7 B.t.u. The A. E. G. turbine of about the same rating gives an efficiency ratio of 68.7, but the heat units available per pound of steam are 434.2 or about 14 per cent. greater. If the latter machine were operated with steam having 380.7 B.t.u. per lb., there is little doubt that the efficiency ratio would equal or approach closely that of the Erste Brünner turbine.

3. The efficiency of the electrical generators is included in the efficiency ratio.

This has considerable importance if the ratios are used to compare turbine types.

It is stated in the paper that European builders generally guarantee better generator efficiency than American builders.

4. The results may be affected by the inclusion or exclusion of the power necessary for steam excitation and auxiliaries, as well as other variations in testing practice.

Speed is a factor which cannot be neglected in making comparisons upon which to base opinions as to type. Either type is benefited by an increase of rotative speed, and it is almost certain that in the future the speed of all types will not differ materially for machines of a given rating and service.

E. D. Dreyfus, of the West Penn Railways Co., urged the establishment of some method of turbine rating, which would be recognized by, and satisfactory to, the engineering fraternity, also rules of conducting tests of this type of power-generating unit, including the turbine and generator as a whole. There is a wide dissimilarity in present practice.

Prof. James A. Moyer, of the University of Michigan, said that such a tabulation as Professor Christie had presented would be incomplete if it did not include the very elaborately detailed and accurate test of a Sulzer turbine of the combined impulse-reaction type reported by Probessor Stodola in the *Zeitschrift des Vereines deutscher Ingenieure,* Oct. 28, 1911, page 1799. At the load showing the best economy (2058 kw.) the steam consumption was 15.21 lb. per kw.-hr. Speed was 1501 r.p.m., absolute pressure 178.15 lb., superheat 182 deg. F., vacuum, referred to 29.92 in. barometer, 28.41 in. (0.740 lb. per sq.in. abs. pressure). Calculated from these data the B.t.u. per kw.-hr. are 18,800, the heat utilized per pound of steam 224.5 B.t.u., the available heat per pound of steam 380.1 B.t.u. and the efficiency ratio 59.1 per cent. The generator efficiency was low for a machine of this kind. This is certainly a reliable test of a presumably well designed turbine of the Curtis-Parsons type and the results are not particularly favorable as compared with the data ordinarily published. It should be clear, therefore, that the adoption of a design consisting of both impulse and reaction stages along with excellence of construction details, does not necessarily give higher efficiencies on the Rankine cycle than other types. The Sulzer turbine was laboring under the disadvantage that it had three rows of moving blades and two rows of stationary blades instead of the modern Curtis construction, employing two rows of moving and one row of stationary blades per stage. It has never been shown that anything is gained by attaching to a blade wheel more than two rows of moving blades or buckets.

Professor Moyer felt that tests showing Rankine efficiencies greater than about 68 per cent. should be accepted only with a great deal of caution. He did not feel justified in accepting without reservation the data of the first three and the fifth tests reported in the table. The second of these tests does not bear the earmarks of a test made with very much accuracy. Practically all the data are given in round numbers, usually indicating approximation. The load is exactly 6000 kw. The steam pressure is given roughly at 13 atmospheres absolute. The steam temperature averaged apparently exactly 300 deg. C. at the throttle throughout the test, and the steam consumption per kw.-hr. was calculated not much closer than the nearest half pound. He was inclined to agree with Mr. Emmett in his statement at a previous meeting of the society that the data for the fourth test were "not altogether convincing." There is very little erosion of the blades in either the Curtis or the so-called Curtis-Parsons type, and the constant erosion of continuous operation for five years will have very little effect on the economy of the turbine as compared with the corresponding deterioration of a reciprocating engine. He criticized Mr. Zoelly's claim that he got the best results on his turbine by throttle governing. The Zoelly method of governing is cheap and simple and that, as he understood it, is the reason for its adoption. He would like to be shown, however, by authentic tests that it is even as good as some other types.

George A. Orrok, of the New York Edison Co., stated that he had been sufficiently interested in the test which Professor Moyer questioned to correspond with the maker of the test and had thoroughly satisfied himself as to its authenticity.

Francis Hodgkinson, of the Westinghouse Machine Co., said that for 60-cycle work from 3000 to 10,000 kw., 1800 r.p.m., is the speed employed by the Westinghouse company. For larger powers than this, 1200 r.p.m. is used. For 25-cycle work 1500 r.p.m. is employed up to a rating of 15,000 kw. normal, or 20,000 kw. maximum. A speed of 750 r.p.m. is not likely to be again employed for capacities less than 30,000 to 40,000 kw. The impulse blading is operated at speeds of from 350 to 500 ft. per sec., according to the particular type or size of machine. The reaction or Parsons blading occasionally reaches a speed of 500 ft. per sec. at the extreme low pressure end in cases of high powers at the given speeds. The higher speed being necessary because of the large steam volumes involved, the last low-pressure rows varying in condensing machines from 300 to 500 ft. per sec. In non-condensing machines, of course, the velocities are lower.

Considerably higher journal velocities are employed than Professor Christie has shown; 80 ft. per sec. and 100 lb. per sq.in. of projected area are commonly employed. There can be no question that such pressures and velocities may be materially exceeded without encountering wear of the bearings, if proper bearing design, and flooded lubrication are employed. The higher the journal velocity, the greater is the rate of shear of the oil, and hence more cooling capacity must be employed. However, it is undesirable to operate with cold oil, as this only increases the bearing losses. No difficulties are encountered with temperatures approximating 150 deg. F.

Professor Christie quotes the opinion of European engineers that higher economy is to be obtained from impulse elements of the Rateau or Zoelly types, than with impulse elements of the Curtis design. Mr. Hodgkinson thought this to be a generally accepted fact, the greater losses in the case of the latter type being doubtlessly due to the low efficiency of the last or third row of moving blades, probably because of eddies produced by the steam passing through the previous rows. A previous speaker had stated that the third moving row in a Curtis element had never been found of value. This is not true for certain high steam and low blade velocities. It is certain, however, that the efficiency of each succeeding moving row rapidly decreases.

By the same token, it would seem that the reaction turbine may be more economical than either, unless it be subjected to some disability. This would seem to be plain, for with an impulse turbine two complete conversions of energy are necessary—the first expanding the steam through nozzles, the steam doing work in giving itself velocity, the second energy transformation being in the blades, where the velocity of the steam is abstracted. Both of these transformations of energy are subject to losses, the second greater than the first. If the nozzles could be permitted to revolve, and there were no blades at all, obviously such a turbine would be the more efficient, if, as said before, it does not suffer from some disability or disadvantage. The reaction or Parsons turbine is a machine of this type, but unfortunately is subject to some leakage. However, where the speed of the turbine permits a de-

sign of blading which has an inappreciable ratio of leakage, such a turbine is highly efficient, which is exactly the feature of some of the later high-speed turbines, and is also the feature of the combination type of turbine, comprising a Curtis element for the high-pressure, and a reaction element for the low-pressure section. The Curtis element, as compared to an all Parsons design, replaces a large number of rows of reaction blading of small diameter, with the attendant great length and weakness of structure, or a smaller number of rows of larger diameter in which case the blades will be smaller and of poor proportions, subject to high rates of leakage.

Of late, the tendency has been toward higher speeds, which, within reasonable limits, is conducive to better results, not only from the standpoint of economy, but also from the standpoint of operativeness. As Professor Christie has stated, congestion at the low-pressure end may be entirely avoided by making the last portion double-flow. Higher rotational speed involves smaller diameters and hence better blade proportions at the high-pressure end with the attendant lesser leakage. In addition to this, the turbine cylinder structure as a whole is much less gigantic, which tends toward less distortion of structure with temperature changes, thus permitting, if it were desired to do so, running the blades with smaller clearances. Higher rotational speed does not necessarily mean higher stresses in the rotating parts.

Professor Christie very properly points out that there is a lower leakage ratio in a single-flow turbine, than in one of double-flow construction. However, the choice of rotative speed is limited on account of the standard frequencies employed. While reducing the speed to the next lower one available may permit of single-flow construction, the advantages may be entirely offset by the increased clearances which must be employed, simply because the size of the machine has so materially increased with the attendant greater amount of distortion. Therefore, the turbine having the low-pressure sections double-flow is frequently more economical. It is convenient to be able to double up the steam passage at the low-pressure end, because of the tremendous rate with which the steam volume increases when turbines are designed for high vacuum.

Concerning blade troubles and blade construction, Mr. Hodgkinson ventured the statement that no turbine built by a reputable manufacturer in this country, ever came to grief because of the blades becoming detached, due to centrifugal force, per se. Accidents have happened because of misalignment or distortion which have resulted either in blades colliding with turbine parts or with one another, which of necessity resulted in injury. The principal source of blade trouble is due to the blades buzzing or vibrating in the steam current. The natural period of the blade may be raised by means of a shroud or other form of lashing, which, of course, offers some protection. One thing to be avoided in blade construction is the reduction of section where the blade is attached to the blade-carrying member, which obviously permits of deflection due to vibration taking place at that point, which if material, results in ultimate fracture.

Mr. Hodgkinson exhibited with the aid of the lantern slide reproduced herewith, standard construction for reaction blading as employed by the Westinghouse Machine Co. The feature, as will be seen, is an absolute interlocking system, the strength of which is equal to that of the blade itself. The blade sections are stronger at the root than at the tip. The section tapers slightly from the root, thus eliminating chance of injury due to vibrating or buzzing, should it exist.

As shown in the illustration, the bottom end of the blade is upset, by a truly upsetting process and not any mere bending, and the die for holding the blade at this time, is so arranged that the section of the blade is increased in thickness throughout its lower portion, this latter feature, however, not being clearly shown in the illustration.

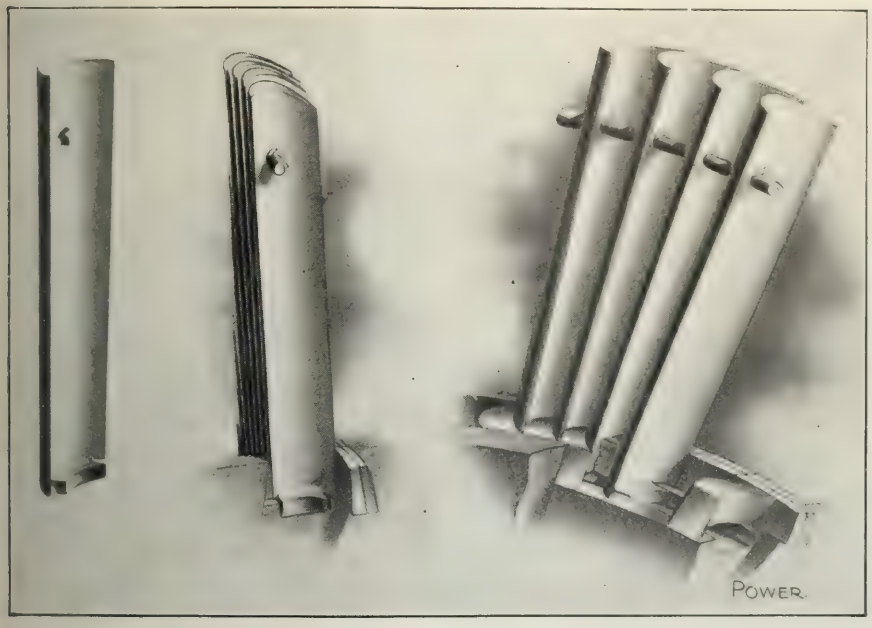
The dovetailed groove in the blade-carrying member has a supplementary groove in the floor of the main groove into which fits the upset end of the blade. The latter locks under the packing pieces which occupy the main groove. The larger blades have compound wedges which go between the walls of the groove and the blades and packing pieces, effectively filling the groove more satisfactorily than a calking strip. With the smaller blades, the wedges are found unnecessary, owing to the lighter sections of the packing pieces, which permits them to fill the groove, when driven up into place. Inasmuch as no calking of any kind is resorted to, blades may be taken out and replaced an indefinite number of times.

Professor Christie discusses overload capacities of machines as practiced in this country and Europe, and suggests that the society fix a rating for turbines, defining their overload capacity. This may be difficult unless the turbine rating be entirely disassociated from that of the generator. The speaker cordially concurred with Professor Christie in that much confusion exists. In discussing capacities of machines, one is obliged to go into some explanation as to what one means. In fact, the rating of machines needs more describing nowadays than does the load factor. It causes particular confusion in statements of the cost, per kilowatt, of installations.

There are essentially two kinds of ratings required today, in modern power plants.

One case is that of the large plant containing many units, where a unit if running at all, is carrying its maximum load. The unit best adapted for such a plant is one the generator of which has a maximum continuous rating with say a 50-deg. rise, the turbine to give its best steam consumption at this load. It need have no overload capacity, or at most but 10 or 15 per cent.

The other case is that of smaller plants, containing fewer units, and having to encounter certain daily peaks, when it is convenient to have units capable of carry-



FORM OF BLADING USED IN WESTINGHOUSE TURBINE

ing loads for limited periods at some sacrifice of economy. Here, while the generator may still be given a certain maximum continuous rating, the turbine should be most economical at some fraction of this load. The old system of rating a machine at a normal full load at 40 deg. rise, the turbine giving best economy at that load, the generator being capable of carrying 25 per cent. overload continuously, and both being capable of carrying 50 per cent. overload for a limited time, is a system of rating, admirably adapted for the plant containing few units. This system of rating was used in this country in the early days of turbine work, and had not been previously used in Europe. The system was brought about by the turbine having to compete with the Corliss engine of that day. Latterly, "bargain-counter methods," as Professor Christie calls them, have caused such units to have their turbines sometimes given a maximum continuous rating corresponding to the 50 per cent. overload just referred to. It would seem entirely possible to have uniformity of generator ratings on a maximum continuous load basis—the turbine, however, to have its best steam consumption at some fraction of this rating, and overload capacity to suit the generator. Plainly, the generator and turbine should be considered separately.

W. L. R. EMMET, of the General Electric Co., criticized the turbine tests as tabulated by Professor Christie, which he claimed do great injustice to the Curtis type and convey very incorrect impressions.

The first machine mentioned in this table is a 2000-kw. Curtis-Parsons machine operating at 1500 r.p.m. and is reported to give an efficiency of 71.9 per cent. with a moderate degree of superheat. Everybody familiar with the design of either Curtis or Parsons machines knows perfectly well that no such result has ever been produced under such conditions.

The efficiencies assigned to the five General Electric Curtis machines mentioned range from 63.6 down to 61 per cent. The first, which operates at 3464 kw., is a representative result of the performances of the machine in question and is a good performance when the very high vacuum used is considered. Of the other four only one of the tests is representative of the performances of the machine in question when correctly tested and in good condition, and in that case also the extremely high vacuum is the cause of a relatively low efficiency.

In discussing this table the writer says, "It is therefore apparent that the efficiency ratio alone will best express the degree to which the designer has approached ideal results in his turbines." To this statement exception should be taken for the reason that it is naturally possible to produce higher efficiencies in turbine construction with moderate degrees of vacuum than it is with very high degrees of vacuum, the limitations to the effective use of high vacuum being many and their extent being affected by speed and capacity. Therefore the designer who produces a relatively low efficiency with a very high vacuum may have accomplished his purpose much more creditably than another who has produced a higher efficiency with a low vacuum.

None of the General Electric machines referred to in Professor Christie's tabulation, except the first mentioned, was designed later than 1904, while many of the other makes are of recent production. Mr. Emmett presented a tabulation of tests (Table I) of a few representative Curtis machines. The first two tests in this list apply to machines in Boston, very carefully and repeatedly tested when they were comparatively new. These are of the same date and type as most of those to which the figures in Professor Christie's tabulated report could apply, and are representative of the General Electric's original large five-stage vertical units when correctly tested and in good condition. The tests were made by the Boston Edison Co.'s engineers with very perfect facilities.

The third test applies to an 8000-kw. machine of the same date and type in Chicago. One of the tests in Professor Christie's paper also purports to refer to this machine under different conditions.

The next three items given refer to recent six-stage Curtis machines and the last to a three-stage Curtis machine recently built by the British Thomson-Houston Co. This machine produces a very remarkable result, considering its small capacity and extreme simplicity.

The accurate testing of steam units requires great care and is susceptible of many kinds of error. Many figures concerning such tests, which are entirely incorrect, are being constantly circulated. It is only by comparing many tests and carefully observing their consistency as to results and characteristics that one can ever be sure as to turbine economy. The figures given in the accompanying table have been so analyzed and com-

TABLE I. TESTS OF CURTIS TURBINES

Customer	Date of Test	Load K.W.	R.p.m.	Steam Press. Abs.	Superheat Dg.F.	Vacuum	Water rate lb. per kw. hr.	Efficiency per cent.
Boston.....		7526	720	184.40	134	28.65	13.732	66.6
Edison Co.....	1907	7481	720	185.10	129	28.58	13.866	66.6
Commonwealth Edison Co., Chicago.....	1907	12000	750	200.00	125	28.00	14.220	66.2
Boston Edison Co.....	1912	12460	720	207.50	191	27.52	13.630	68.2
Stock.....	1911	4000	1800	189.00	0	28.00	16.050	64.3
Louisville Ltg. Co.....	1911	6500	1800	180.00	100	28.00	14.780	66.10
British Thomson-Houston (maker) Rugby.....	1912	2000	3000	165.00	150	28.00	14.400	67.3

TABLE II. ECONOMY TESTS OF HIGH PRESSURE STEAM TURBINES
EFFICIENCY RATIOS BASED ON E.H.P. MARKS & DAVIS STEAM TABLES USED
References ÷ Zeit. D.V.D. Ing.—Zeitschrift Des Vereines Deutscher Ingenieure

Maker of Turbine	Type	Date of Test	Load K.w.	R.p.m.	Steam Pressure Lbs. Absolute	Temperature at Throttle–deg.F.	Vacuum Referred to 29.92" Bar	Condenser Pressure Lb. Absolute	Lbs. of Steam per Kw.-hr.	B.t.u. per Kw.-hr.	Heat Utilized per lb. of Steam	Heat Available per lb. of Steam	Efficiency Ratio	Reference
Escher, Wyss & Co.....	Zoelly	1912	1,275	3000	196.3	588	28.99	0.455	12.59	15,940	268.6	418.1	64.3	Zeit. D.V.D. Ing. 4/13
Escher, Wyss & Co.....	Zoelly	1912	5,418	1500	180.7	588	28.74	0.580	11.95	15,060	285.5	402.1	71.0	Zeit.D.V.D. Ing. 4/13
Elsassische Maschinenbau Gesellschaft.....	Zoelly	1912	10,006	1250	167.8	565	28.65	0.625	12.01	14,975	284.2	389.1	73.0	Zeit. D.V.D. Ing. 4/13
Belliss & Morcom.....	Drum Impulse	1911	3,000	1500	192.5	558	27.92	0.981	15.0	18,420	227.3	372.7	61.0	Manufacturer's Data
Belliss & Morcom.....	Drum Impulse	1911	2,500	1500	164.7	466	26.92	1.471	16.95	19 815	201.2	323.6	62.2	Manufacturer's Data
Aktiebolaget Ljungstrom Angturbin	Radial Reaction (Ljungstrom)	1911	1,000	3000	181.7	669	28.55	0.671	11.68	15,130	292.0	414.9	70.4	Engineering 4/19 12
		1911	972	3000	176.7	667	28.58	0.657	11.55	14,960	295.3	414.2	71.3	

pared as to be unquestionably correct, and it will be observed that they give an impression concerning the value of Curtis turbines which is very different from that produced by Professor Christie's tabulation.

C. V. KERR, chief engineer, McEwen Bros., said with regard to wear of the blading due to wetness of the steam, that in his double-bucket turbine they had found the wear of the blading in the upper stages. That may be due to one or both of two things. In the first place they allowed a greater proportion of the available energy to produce velocity in the upper stages with the idea that part of that would be transferred to the lower stages and thus equalize the wear further down. This may have been overdone.

Another reason is that the steam is denser in the upper stages and that its erosive power is greater. With regard to the rating of steam turbines, there is one point which is a sort of corollary of the rating question. The small steam-turbine builders drop into the habit of selling their machines on the power basis. A turbine is a peculiar machine. The casing can be made for a certain manufacturing cost and by adding nozzles the power can be doubled, or trebled, or quadrupled, but the small steam turbine competes with the steam engine and the electric motor and they are generally sold on the power basis; hence it is the practice now with at least two of the small steam-turbine builders, to establish a minimum price for the turbine, based on manufacturing costs and then rate upward on the basis of additional power development.

In his closure the author added the following tests as supplementary to the table printed with his paper:

At a Salt Works Power Plant

BY E. J. SAXE

I was employed as an expert engineer by a large salt company which operated a number of widely scattered plants. My job was to get the largest amount of salt from a given amount of coal.

The works to which I first went contained 11 grainers, or shallow wooden tanks, each 150 ft. long, 14 ft. wide, and 18 in. deep, lined with tile, and containing 1500 ft. of 4-in. pipe, laid lengthwise and connected by return bends; they were suspended from timbers laid crosswise of the tanks so that they were about 4 in. under the surface of the brine when the tanks were full.

The boiler room contained twelve 72-ft. by 18-in. horizontal return-tubular boilers, each having a 60-ft. steel stack; they were hand fired, using slack coal for fuel. In an adjoining room there were the necessary pumps for feeding the boilers and pumping the brine; nearby were two ancient engines. The plant was making about 7 bbl. of salt per ton of coal, but the first expected 10 bbl.; the possibilities were 13 bbl.

For the first week I stood by and watched the process and soon discovered that the only way to make salt was the way it was being made; any change would mean: "You can't make no salt."

The draft gage showed 0.4 in. at the base of the stack, and from 0.25 in. to about nothing in the firebox. The Orsat apparatus showed 3 to 4 per cent. CO_2; the stack temperature averaged 650 deg. F.

Metering the water showed that the boilers were doing 200 hp. when the firemen were feeling good, and I found that one grainer would condense all the steam one boiler could make.

The firemen received $1.80 per day of 12 hours, and the plant ran 24 hours per day, 365 days per year. I got a mason to stop up the cracks in the boiler setting, and put in arches and wing walls under the boilers, and got a better showing in the draft readings in the furnace and a much higher CO_2. I tried to stir up the firemen to do better work. But not much enthusiasm could be obtained on a 12-hour shift paying $1.80 per day.

Matters had begun to look up and as less coal was being burned and more steam made I turned my attention to the grainer room. The grainers were set higher than the boilers, and an 8-in. pipe ran the length of the room and supplied the 4-in. pipe with steam. A 1¼-in. pipe drained the condensation into a 4-in. return pipe and thence into a return tank, from which it was taken by the feed pump and fed to the boilers. The condensation flow was regulated by hand, and to make sure that all the water was drained from the coils, steam was blown through them and discharged to the atmosphere through the vapor pipe on the return tank.

I proposed the use of traps, but was informed that, "If you do we can't make no salt." However, I got the traps, and then ran against a problem. The brine was kept at a temperature of about 138 deg. F. When the coils were clean, 10 lb. of steam would give the desired temperature, but as the coils accumulate scale, the pressure was increased until, when the scale became ½ in. thick, 100 lb. pressure was required to do the work. This condition would occur in from two to three weeks, when the brine was drawn off and the scale removed. The problem was to get a trap that would handle the water with this variable pressure, but a trap with a large balanced valve was selected, and did the work satisfactorily. Steam was saved and salt was still made.

My next discovery was that while the return water came to the return tank at 212 deg. F., the feed water was only heated about 120 deg. An investigation showed that a 2-in. pipe from the bottom of the tank was wide open to the sewer, and cold water was making up the supply; I was told that the feed pump would not handle the water any hotter. The pump should have handled the water as the steam tank was 18 in. higher than the pump, and there was some reason if it would not. Investigation disclosed that the water end had been designed for valves ½ in. thick, but someone had substituted ⅝-in. valves, with extra-heavy springs, and the valves had a scant ⅛-in. lift. New valves were secured, light springs used, and the pump did pump hot water. The result of all these small economies was the desired 13 bbl. of salt per ton of coal.

The fuel used was a very volatile, long flaming coal, and the smoke that poured from the stacks was a constant eyesore, but with various arrangements of brick arches and wing walls, I could not stop it without using steam jets, which I would not do. I could not persuade the firm to pull down the stacks, and put in induced draft, which would have eliminated the smoke, and made a saving of at least 10 per cent. of the fuel.

While making the changes I ran boiler tests to determine the advantage of each. My first test was plain furnace versus arch and wing walls; result, 13 per cent. economy and 13.5 per cent. increased capacity in favor of the wing walls and arches.

Next I picked out a fireman who was willing to do as he was told, and had him fire the boiler as carefully as possible; result, 27 per cent. increase in capacity and 10 per cent. in economy.

Then I told the manager that he was burning 90 tons of coal per day, at a cost of $225. I said: "If I can give those firemen $1 per day to stand up to the boilers and fire them with three shovelfuls at two-minute intervals, they will be glad to do it. It will save 10 per cent. or a profit of $22.50 at a cost of $8; net $14.50 per day. Do away with the smoke, and have the firemen looking for the job instead of the job always looking for the firemen." But he said: "It can't be done."

With such decisions on the part of those higher up, the engineer does not find much encouragement or opportunity for introducing economical methods of operation.

In 1910 the mines of the Transvaal employed 257 steam-driven air compressors having 100,291 hp., there being an average of three 50-drill compressors on each mine. Formerly, all the air compressors were steam-driven, but now electric energy is furnished to all the mines by the Victoria Falls & Transvaal Power Co., Ltd., and electricity for air compressors has superseded steam.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Inspecting and Testing Electrical Apparatus—II

By A. L. Cook

Alternating-current Systems

Alternating-current generating systems generally may be divided into three classes:

1. Generation at 220, 440, or 550 volts for supplying power and light to small areas or to a single building. The system may be three-phase, quarter or two-phase, or single-phase; the frequency generally 60 cycles.

2. Generation at 1100 or 2200 volts for supplying power and light a distance of several miles from the station. As in class 1 the system may be single or polyphase, generally with a frequency of 60 cycles.

3. Generation at 6600, 11,000 or 13,200 volts for supplying power and light to distances not exceeding 15 miles. The three-phase system is generally used for power systems, for this class, at a frequency of 60 or 25 cycles.

To transmit power more than 15 miles, still higher voltage is necessary for economy, and step-up transformers are required, since generators are not wound for voltages higher than those of class 3. All alternating-current generators require direct-current exciters and these should be inspected and tested as described in the previous article for direct-current machines.

Low Voltages

In class 1, the switchboard is often similar to that for direct-current, laminated-brush, carbon-break circuit-breakers and standard knife switches being employed, with the busbars mounted at the rear of the board. The present tendency is to use oil circuit-breakers with the higher voltages, such as 440 or 550, particularly if the station is of large capacity. The switchboard should be inspected like that for direct current, as already described, either a magneto or a direct-current voltmeter being used to test for grounds.

Oil circuit-breakers should be examined for good contact when the switch is closed, and automatic circuit-breakers should be tripped several times by raising the plunger by hand. Circuit-breakers which are tripped by current transformers should preferably be tested with current passed through the trip coil. Usually these coils require from 5 to 10 amp. to trip the switch.

Sometimes the switches are closed and tripped by direct current, in which case the current transformers would be connected to a relay which closes the tripping circuit. Usually all the trip coils or relays are wound for the same current. The current in the feeder, causing the circuit-breaker to open, depends upon the ratio of the current transformers connected in the feeder circuit. Thus, if the transformers have a ratio of 1500 to 5 amp., 5 amp. would flow in the trip-coil circuit when 1500 amp. is flowing in the feeder. Usually an ammeter is in series with the trip coil so that it receives 5 amp. at the same time, although the scale generally indicates the actual current in the feeder. Usually the trip coil is marked for the actual current in the feeder the same as the ammeter.

If a source of alternating current of the same frequency as that normally used on the feeder, is available, the trip coil may be tested by passing current through the secondary circuit. Fig. 3 shows connections for current transformers and trip coils as used in two-phase and three-phase circuits. To make the test, the secondaries of the current transformers should be first short-circuited, as indicated by the dotted lines; then the circuit should be opened at a convenient point, such as *A*, and the test circuit inserted. The trip coil, feeder ammeter and test circuit must all be in series. The alternating-current supply may be at 110 volts, or any other convenient voltage, and the amount of current required usually will not be more than 10 amp. The resistance in the test circuit may consist of incandescent lamps or a small water rheostat. The trip-coil plunger should be set at the desired point and the current gradually increased with the rheostat and the feeder-ammeter reading noted when the circuit-breaker opens. This is the current which must flow in the feeder to trip the circuit-breaker.

Medium Voltages

For voltages of 1100 or 2200, oil circuit-breakers are nearly always used.

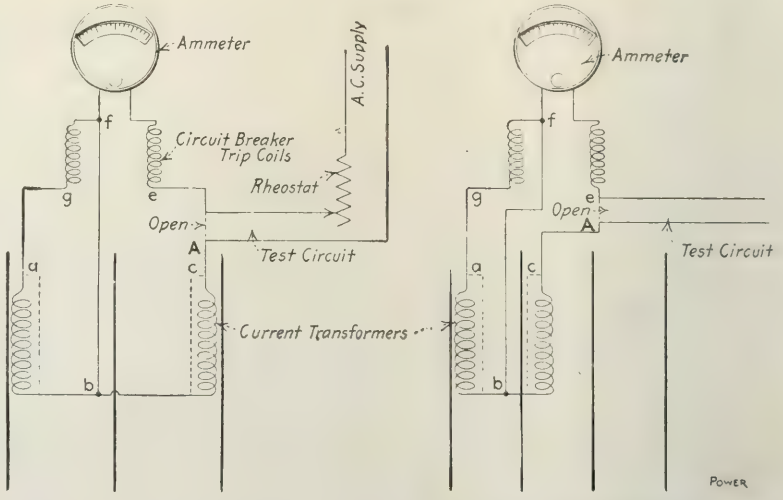


Fig. 3. Connections for Current Transformers and Trip Coils for Three- and Two-phase Circuits

The busbars are frequently at the back of the switchboard, although it is better to mount them and circuit-breakers away from the board, with only the operating handles on the panels. The circuit-breakers should be inspected as outlined under class 1, and if opened and closed electrically, should be operated several times to make sure that they close and trip satisfactorily and that the proper indicating lamp lights each time. The relays used with these breakers may be tested as described for ordinary circuit-breaker trip coils.

Tests for grounds on this class of equipment cannot be made very satisfactorily with a magneto or an ordinary voltmeter, but, if nothing better is available, either may be used to indicate the condition of the insulation. If possible

the wiring should be tested with high voltage. For important systems a test with double the normal voltage, applied for 15 min. between each conductor and the ground is satisfactory. This test voltage is to be applied only to the wiring and not to the generators or the voltage transformers. The oil circuit-breakers and switches, however, can be tested with the wiring. Since the current transformers are connected in with the wiring, they would also be subjected to this test voltage. The current transformers usually furnished for this service are insulated to withstand a test of this character, but if there is any doubt about it the manufacturer of the transformers should be consulted before making the test.

Voltage tests generally can be made only on new systems, for extensions to stations in operation as usually arranged make this impracticable. If the new part of the equipment can be isolated from the old, tests should be made. If no apparatus is available, giving the required voltage for testing the new equipment, as described, a simple test can be made with one of the main generators. The wiring to be tested should be connected to a terminal of one of the generators which is disconnected from the busbars. A second terminal of the generator is then grounded. As a precaution against a rush of current if the insulator breaks down it is sometimes advisable to ground this terminal through resistance. The generator voltage is raised slowly until the voltmeter connected to the two generator terminals reads about 10 per cent. above normal at which point it should be held for about 15 min. This also subjects the generator to a voltage slightly greater than normal, but this voltage is not dangerous for machines of standard construction.

High Voltages

In the third class as in the second, oil circuit-breakers are usually employed with single-pole knife switches in addition for isolating the circuit-breakers and other apparatus for inspection and cleaning. The busbars and circuit-breakers are seldom mounted on the switchboard, but are installed remote from the board, usually with the individual circuit-breakers and the busbars inclosed in masonry compartments. The inspection of the circuit-breakers should be made as outlined for the medium-voltage system.

For high-tension apparatus, it is desirable to test the station wiring for faults, which can be done satisfactorily only by subjecting the equipment to a high-voltage test. This requires a testing transformer unless regular transformers of the proper voltage are available. A test at double the normal voltage applied for 15 min. between the wiring and the ground is sufficient. In such tests, as previously mentioned, the wiring, busbars, switches, oil circuit-breakers and current transformers, constitute the apparatus to be tested; the other equipment, such as generators and transformers, should not be subjected to this test since they are either tested in the factory or, if wound at the plant, would be given special tests by the erectors.

To make a voltage test on the wiring, a relatively low voltage is first applied, such as 10 per cent. of the final value, and gradually increased until the proper value is reached. If a regular testing outfit is used, generally some method is provided for adjusting this voltage, but if an ordinary transformer is used special provision must be made for controlling the voltage. Most convenient is to use a generator supplying only the testing transformer, regulating the generator voltage by its field rheostat. Another method is to employ a water rheostat in series with the primary of the testing transformer, but this regulation is more difficult and less satisfactory for testing purposes.

The actual voltage applied to the apparatus may be determined; by reading the voltage on the low-tension side of the transformer with a voltmeter and multiplying by the ratio of the transformer as marked on the name-plate; or by measuring the actual testing voltage with an electro-static voltmeter or a spark gap. The first method gives only approximate results but is sufficiently accurate for ordinary tests, especially if the testing load is not more than one-fourth the full-load rating of the transformer. The second method, although more accurate, requires more careful manipulation. If a high-voltage test can be made as described, the wiring at least should be tested by using the generator alone as described for medium voltages.

The bearings should be examined and the air gap checked as explained for direct-current machines. The brushes used on the slip rings should be fitted and the spring tension adjusted. The machine should then be freed from dirt, using air under pressure, if available.

Testing the Generator

If possible, the generator insulation should be tested by the voltmeter method before being put in service. With the machines belonging to class 1 in apparently good condition, this test is not very important, but for the other two classes is essential. Voltmeters which operate either on direct or alternating current are not suitable for making insulation tests, the only proper kind being the permanent-magnet type which can be used only on direct current. The insulation resistance required for alternating-current armatures is given by one manufacturer as approximately 1,000,000 ohms per 1000 volts of rated voltage, when the machine is at its full-load temperature. The field winding, however, should show a resistance higher than this. Large machines will have a lower resistance than small machines of the same voltage rating. For machines in the first and second classes, the ordinary 125-volt circuit and the 150-volt instrument can be used to test the insulation, but for the third class a high-resistance voltmeter is generally required. This class of machine is frequently given a high-voltage test, but this should be made only under the direction of the manufacturer.

If the generator has been exposed to moisture, it must be dried out and some provision made for testing the insulation as outlined before. To dry out the generator, the armature should be short-circuited beyond the main switch and current transformers, and the field current adjusted to give sufficient armature current so that the temperature shall not exceed 160 deg. F. on any exposed part of the windings. Anyway, the current should not exceed 150 per cent. of full-load current. If sufficient temperature rise is difficult to get at normal speed, owing to the small iron loss, the speed may be reduced, thereby decreasing the air circulation through the ventilating ducts; or with inclosed turbo-generators, the duct may be partly closed. High-voltage machines take longer to dry out than low-voltage machines, due to the greater amount of insulation. Arrangements should be made to measure the insulation resistance from time to time as described for direct-current machines. (Article I.)

Before starting a polyphase generator the main connections should be checked, if possible, for proper phase relation, so they will not have to be changed when the generator is "phased out." With two machines of the same make, corresponding terminals should connect to the same busbar, if the generators rotate in the same direction when viewed from the collector-ring side. If they rotate in opposite directions, any two of the leads of a three-phase machine, or the two leads of one phase of a two-phase machine, should be interchanged.

Before starting the generator, the main switch and field switch should be open and the field rheostat all cut in. The generator can then be started slowly. The oil rings should be inspected and the generator then can be gradually brought to speed. The field switch should be closed and the voltage raised to normal by cutting out the field rheostat. In a three-phase generator the voltages of each phase should be the same. With a two-phase four-wire machine, the voltage of the two phases should be the same. For a two-phase, three-wire generator, the voltage between either outside terminal and the common terminal

should be the same, and that between outside wires should be 1.41 times this value.

If the station has only one generator, the load can now be thrown on, but if one must operate in parallel with other

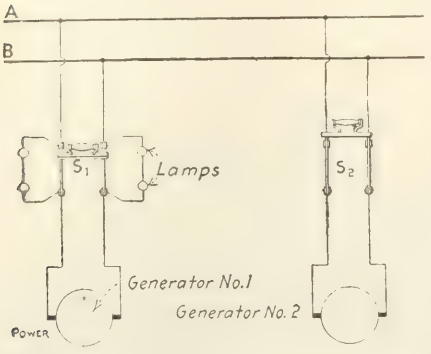


FIG. 4. SHOWING METHOD OF SYNCHRONIZING LOW-VOLTAGE GENERATORS

machines, it is necessary to "phase out" the new machine and also check the synchonizing connections.

PUTTING GENERATORS IN PARALLEL

For two alternating-current generators to operate properly in parallel, the voltages must be approximately equal, the frequencies the same, and the voltages in phase, that is, the corresponding maximum and minimum values of the voltage must occur at the same instant; under these conditions the machines are in synchronism.

The fundamental operation of synchronizing is illustrated for a single-phase generator by Fig. 4. Generator No. 2 has its main switch S_2 closed and is therefore connected to the busbars A and B; generator No. 1 has switch S_1 open and is to be synchronized with the busbars. Incandescent lamps are connected across the terminals of the switch, as shown, the number in series on each side being such that the sum of their voltages equals the voltage of the generator. When generator No. 1 is in synchronism the lamps will be dark, since there will be no difference in potential across the corresponding terminals of the switch. If the voltage of generator No. 1 is approximately the same as that of the busbars, but the frequency is not the same, the lamps will vary at intervals from bright to dark and the rapidity of these pulsations will be an indication of the difference in frequency between the busbars and generator No. 1. The speed of the generator can then be changed until the pulsations of the lamps occur quite slowly, and when this condition is reached and the lamps are dark, generator switch S_1 can be closed and the generator is thus put in parallel with the busbars.

For low-voltage machines, such as are included in class 1, lamps could be used in this way, but for higher voltages, such as those of classes 2 and 3, potential transformers must be used. Fig. 5 is a diagram of connections for this case. It will be seen that the conditions are similar to those in Fig. 4. Usually the transformers have such a ratio that normal voltage on the generators gives about 110 volts on the secondary; therefore two 110-volt lamps are sufficient. If the frequency of generator No. 1 is not the same as that of the busbars, the lamps will pulsate, and when the lamps are dark for some time the generator is in phase and switch S_1 can be closed. If the secondary connections are crossed, the lamps will burn brightly when the generator is in phase, instead of being dark as before; this arrangement is preferred by some operators.

For polyphase machines, the conditions outlined should occur on each phase simultaneously. Thus for a three-phase machine, if three sets of lamps are connected across corresponding terminals of the

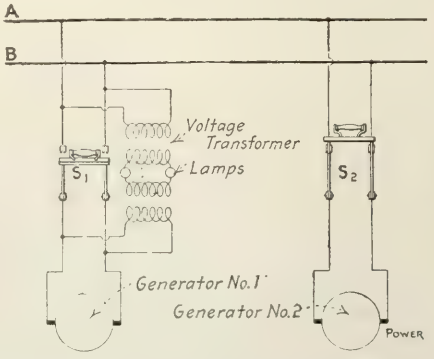


FIG. 5. SHOWING METHOD OF SYNCHRONIZING HIGH-VOLTAGE GENERATORS

main switch, all three sets of lamps should be dark and bright together. Having once phased out a polyphase machine, the generator thereafter need only be synchronized on one phase, unless there is some way by which the main connections of the generator or the busbars can become changed.

it to another machine which is known to have correct connections and to bring the two machines up to speed together. If the station can be shut down for a time, the two generators may both be connected to the busbars by closing the main switches; if not, a spare generator can be used with temporary connections. These temporary connections should be made so that when the switches are closed, the machines are paralleled the same as if connected to the busbars. The fields of each should now be excited, adjusting the field current to about that required to give normal voltage at no load when the generator is up to speed. With the two armatures thus connected the generators should be started slowly, keeping both as nearly as possible at the same speed. At first heavy current will flow in the machines, but as they speed up, this should decrease and finally fall to a small amount indicating synchronism. This should occur before half speed is reached. If the heavy current continues and seems to increase, the connections are wrong and the two leads of one phase, in the case of a two-phase machine, or any two leads of a three-phase machine, should be interchanged, and the test repeated. The generators should fall into synchronism as indicated by the ammeters and the machines can be brought up to normal speed together.

If the machines are both connected to the busbars in the foregoing test, the permanent synchronizing connections can be tested at the same time. When the main connections have been found correct, the synchronizing lamps or synchroscope should indicate that the machines are in synchronism, and if not, the synchronizing connections should be changed. If, however, temporary connections were made in phasing out, the synchronizing connections must be checked separately. To do this, the main

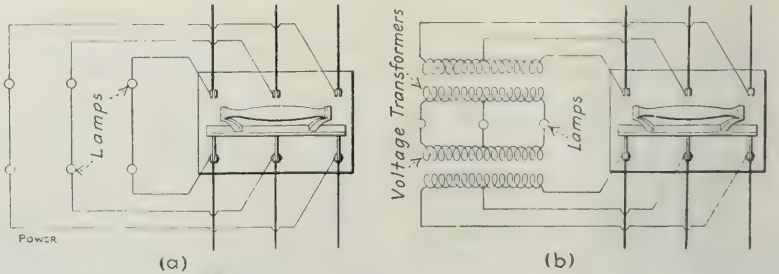


FIG. 6. SYNCHRONIZING CONNECTIONS FOR THREE-PHASE GENERATORS

At the time that the machine is phased out, the synchronizing connections may be checked. With single-phase generators, if the synchronizing connections can be easily traced, no further test is necessary, since there are only two main leads and they can be synchronized as previously described. For polyphase generators, especially where potential transformers are used, the easiest way to phase out the new machine is to connect leads of the new generator should be disconnected at the terminals, and the ends of the leads fastened clear of each other, and the main generator switch closed, thus connecting the leads to the busbar. This will also connect the synchronizing circuits of the machine, and the lamps should indicate synchronism. If they do not, the synchronizing connections should be changed. Care should be taken to see that these synchronizing

connections are made to the corresponding phase of the busbars and machine.

When it is not convenient to bring two machines up to speed together to phase out, temporary synchronizing connections may be made. This method requires connections for each phase similar to those shown in Figs. 4 or 5. For three-phase generators the proper connections are shown in Fig. 6, *a* being for low-voltage machines and *b* for high voltage. The connections are made across the main switch of the generator to be tested. If the generator is brought up to speed and its voltage adjusted to normal, the three sets of lamps should be bright and dark simultaneously, the rapidity of changes from dark to bright depending upon the difference in the frequency of the machine and busbar. If the lamps are not

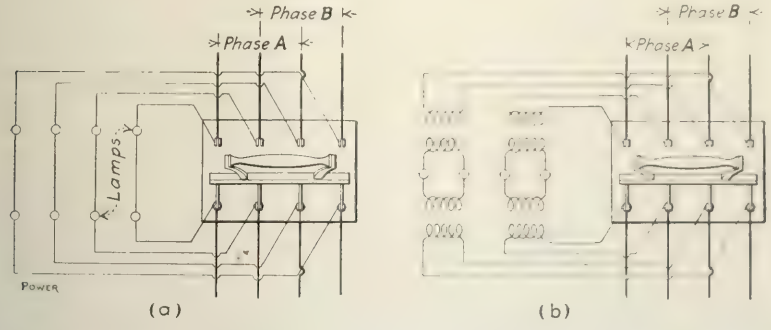


FIG. 7. SYNCHRONIZING CONNECTIONS FOR A TWO-PHASE, FOUR-WIRE GENERATOR

all bright simultaneously, any two of the main leads should be interchanged, keeping the synchronizing connections on the same terminals of the switch as before, and the test repeated. The lamps should now all fluctuate together, indicating that the main connections are correct. Fig. 7 shows similar connections for a two-phase, four-wire generator. The leads corresponding to each phase are first determined by a voltmeter as previously described. When the generator is operated, the lamps should all burn brightly at the same time; if not, the two leads of either phase should be reversed.

A polyphase induction motor may be used to phase out a generator, in which case the motor is first connected to the terminals of the main switch on the busbar side, through transformers if necessary, and the direction of rotation noted. The motor leads are marked and then connected to the generator side of the switch, taking care that the leads in each case are connected to the same pole. If the motor rotates in the same direction the connections are correct. Having phased out the main generator connections by either of these methods, the permanent synchronizing connections must be checked by disconnecting the generator leads at the terminals of the machine, and closing the main switch as previously described.

Each time that the machine is put in service during regular operation the generator must be synchronized before it is paralleled with the other machines. To do this only requires adjusting the speed of the incoming machine until the regular synchronizing devices indicate that it is in proper phase relation with the busbars, and then the main switch can be closed.

Incandescent lamps are frequently used to indicate synchronism, as described, but many stations have synchronism indicators, or synchroscopes, which have a pointer mounted in a suitable case, with a dial marked at one point indicating synchronism. Usually in these instruments, the pointer is free to rotate and has no springs to control its action. When one pair of terminals of this instrument is connected to the incoming generator and the other pair to the busbars, the pointer will rotate if the two generators do not have the same frequency, and the speed of rotation will depend upon the difference in frequency. If the incoming machine is slightly slower than the other generators, the pointer will rotate in one direction, and if faster, will rotate in the opposite direction, clearly indicating what adjustment of the machine is necessary. The instrument is usually connected in the secondary circuit of voltage transformers, one pair of terminals being connected to each transformer. The lamps in Fig. 5 can be retained as a check, even if the synchroscope is used.

If the generator is in multiple with other generators, the load can be transferred to the new machine by increasing the speed of the latter slightly. A change in the field current would have practically no effect upon the amount of load that the generator would carry, but would change the current in the generator armature. In shutting down an alternating-current generator, it is preferable to transfer as much of the load as possible to the other machines, but frequently this is not done; the main switch being opened with the generator carrying its share of the load. The field switch should *never* be opened while the generator is under load and connected in parallel with other generators, as a very heavy flow of current would result.

Chicago Electrolysis Ordinance

The city council of Chicago passed an ordinance July 15 requiring the railways to so equip and operate their systems that damage due to electrolysis caused by stray currents will be prevented.

The ordinance requires that all uninsulated electrical return circuits must be of such current-carrying capacity, and so arranged, that the difference of potential between any two points on the return will not exceed 12 volts, and between any two points on the return 1000 ft. apart, within one-mile radius of the City Hall, will not exceed 1 volt, and between any two points on the return 700 ft. apart, outside of this one-mile radius limit, will not exceed 1 volt. In addition, a proper return conductor system must be so installed and maintained as to protect all metallic work from electrolysis damage.

The return-current amperage on pipes and cable sheaths must not be greater than 0.5 amp. per lb.-ft. for calked cast-iron pipe; 8 amp. per lb.-ft. for screwed wrought-iron pipe, and 16 amp. per lb.-ft. for standard lead or lead-alloy sheaths of cables.

The railways must equip their uninsulated return-current systems with insulated pilot wire circuits and voltmeters so that accurate chart records will be obtained daily showing the difference of potential between the negative busbars in each station and at least four extreme limits on the return circuit in its corresponding feeding district.

The system must also be equipped with recording ammeters, insulated cables and automatic reverse load and overload circuit-breakers which will record and limit the maximum amperes drained from all the metallic work (except the regular return feeders) to less than 10 per cent. of the total output of the station.

The chart records must be so kept as to be always accessible to city officials.

Failure to comply with the ordinance within 9 months from its passage will incur a fine not less than $50 nor more than $200 for each offense; and each day's operation of such equipment contrary to the provisions shall constitute and be regarded as a separate and distinct offense.

The ordinance is to take effect at once.

Ray Palmer, electrician for the city, after extensive observation and numerous careful tests, has estimated that the damage to the pipe lines of the city and the gas company and to the cable sheaths of the city, sanitary district of Chicago, and the telephone company due to stray currents from the tunnel, surface and elevated railway systems, represent an annual maintenance cost of upwards of $600 000. In addition vast damage is done to the steel and iron work of office buildings and other structures which cannot be estimated.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

American Practice in Rating Internal Combustion Engines—IV*

By T. C. Ulbricht and C. E. Torrance, Jr.

Inlet and Exhaust Valves

In this article equations will be given for the relations between valve diameters, cylinder diameters, valve lifts, etc., as derived from the accompanying curves.

Curve 2, Fig. 29, represents the average values of inlet valve diameter plotted against cylinder diameter, for horizontal engines up to 30 in. cylinder diameter. Above this size the ratio of valve to cylinder diameters increases more rapidly and approaches curve 1, which represents maximum values for the smaller engines.

This is because in sizes above 30 in. the ratio of stroke to cylinder diameter increases. This is especially true from 28 or 30 in. to 48 in. diameter, where the piston speeds are high; since the diameter of the inlet valve is plotted against the cylinder diameter, which does not increase as fast as the piston speed, the inlet-valve diameter must increase more rapidly in proportion to the cylinder diameter, to supply a greater quantity of fuel. Letting d equal the diameter of the cylinder and d_v the diameter of the valve, the curves in Fig. 29 may be represented by the formulas:

$d_v = 0.45\ d$ (maximum) curve 1;
$d_v = 0.345\ d$ (average) curve 2;
$d_v = 0.25\ d$ (minimum) curve 3.

Fig. 30 shows similar curves for the exhaust valves of horizontal engines. The equations are:

$d_v = 0.40\ d$ (maximum) curve 1;
$d_v = 0.35\ d$ (average) curve 2;
$d_v = 0.29\ d$ (minimum) curve 3.

Fig. 31 shows average curves for the inlet and exhaust valves of vertical engines, the equation $d_v = 0.34\ d$ being the same for both inlet and exhaust valves.

Average curves plotted between valve lifts (inlet and exhaust) and corresponding valve diameters, for horizontal and vertical engines, are shown in Figs. 32 and 33.

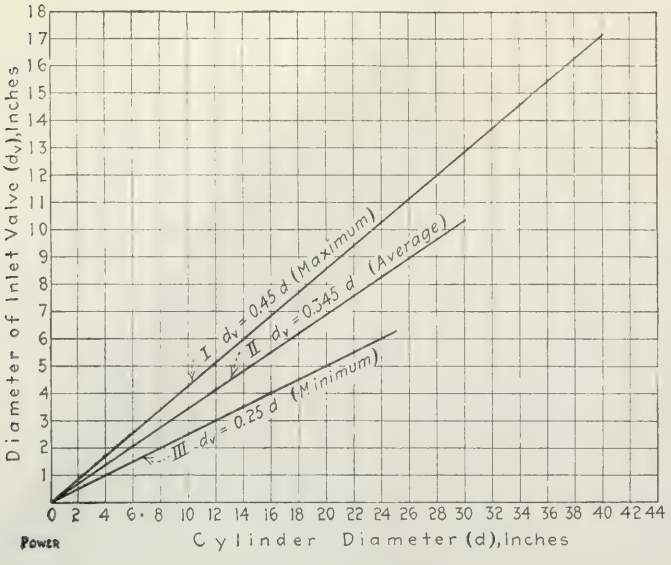

Diameter of Inlet Valve (dv), Inches
Cylinder Diameter (d), Inches
I dv = 0.45 d (Maximum)
II dv = 0.345 d (Average)
III dv = 0.25 d (Minimum)
POWER

Fig. 29. Inlet Valve Diameter vs. Cylinder Diameter for Horizontal Engines

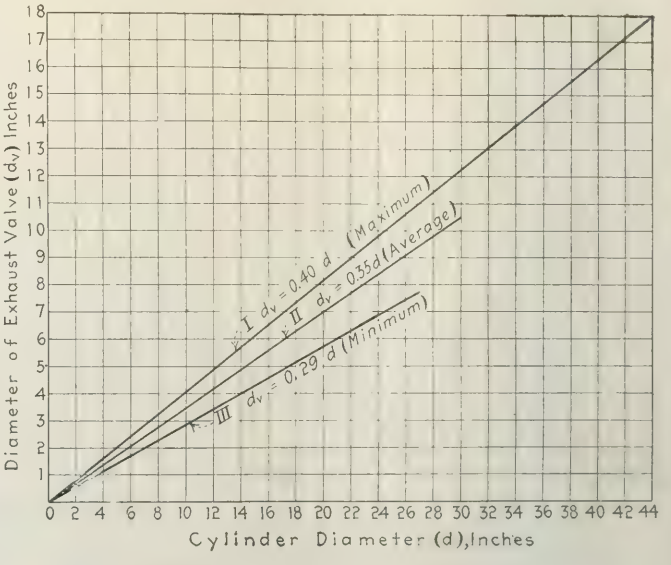


Fig. 30. Exhaust Valve Diameter vs. Cylinder Diameter for Horizontal Engines

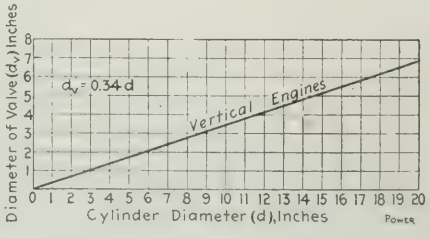


Fig. 31. Cylinder Diameter vs. Valve Diameter for Both Inlet and Exhaust Valves of Vertical Engines

Gas Velocities through Inlet and Exhaust Ports

When a quantity of gas Q passes through an opening of area F with a velocity V,

$$Q = FV$$

where Q is in cubic feet per minute, F in square feet, and V in feet per minute.

The total charge taken into and exhausted from the engine per minute equals the quantity of gas passing through both the inlet and exhaust ports per minute; that is,

$$Q = \frac{\pi d^2}{4 \times 144} \times \frac{2\,ln}{12} = \frac{3.14\,d^2 \times l \times n}{2 \times 1728} = \frac{d^2 ln}{1100}$$

where d is the diameter of the cylinder in inches, l the length of stroke in inches, and n, the revolutions per minute.

The area of the port opening in square inches is $F = \pi\, d_v\, l_v$, in which d_v is the valve diameter in inches, and l_v the valve lift in inches. Substituting these values,

$$F = \frac{3.1416\, d_v l_v}{144} = \frac{d_v l_v}{45.83}\ sq.ft.$$

but as previously found,

$$Q = FV = \frac{d^2 ln}{1100}$$

hence,

$$\frac{d^2 ln}{1100} = \frac{d_v l_v}{45.83} \times V$$

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and,

$$V = \frac{d^2 ln \times 45.83}{1100 \times d_v l_v} = \frac{d^2 ln}{24\, d_v l_v} \text{ ft. per min}$$

Table 3 contains the velocities calculated by this method for various assumed values:

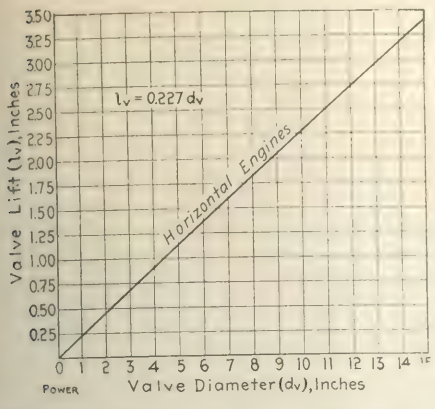


FIG. 32. VALVE LIFT VS. VALVE DIAMETER FOR BOTH INLET AND EXHAUST VALVES IN HORIZONTAL ENGINES

In each case for the assumed brake horsepower rating per cylinder per end, the corresponding d^2ln value was found from the average d^2ln versus *b.hp.* curves of article I. The revolutions per minute were then read from the *r.p.m.* versus *b.hp.* curves. Also, $d^2ln \div n = d^2l$ from which d and l can be found as follows:

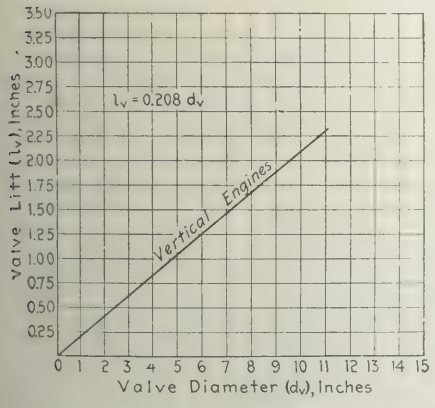


FIG. 33. VALVE LIFT VS. VALVE DIAMETER FOR INLET AND EXHAUST VALVES IN VERTICAL ENGINES

TABLE 3. GAS VELOCITIES THROUGH VALVES

SINGLE-ACTING HORIZONTAL (GASOLINE)

Rated b.hp.	R.p.m.	d^2ln	d^2l	d	l	Area Inlet	Area Exhaust	Lift Inlet	Lift Exhaust	Gas Velocity Inlet	Gas Velocity Exhaust
10	270	170,000	630	7¼″	10¼″	2.68	2.68	0.56	0.56	4400	4400
20	224	333,000	1,492	10½″	14¾″	3.5	3.5	0.78	0.79	5080	5020
30	200	500,000	2,500	12″	17½″	4.12	4.16	0.93	0.94	5430	5350
40	185	660,000	3,570	13¾″	19½″	4.7	4.76	1.07	1.08	5570	5450
50	183	825,000	4,500	14½″	21¼″	4.98	5.05	1.12	1.15	6160	5920
Avg....									Ft. per min.	5340	5294

SINGLE-ACTING HORIZONTAL (PRODUCER GAS)

20	290	380,000	1,310	9½″	13½″	3.35	3.38	0.76	0.77	6220	6080
50	225	920,000	4,080	14½″	20½″	4.87	4.90	1.10	1.12	7150	6980
100	185	1,810,000	9,780	18½″	27½″	6.5	6.57	1.47	1.49	7900	7720
125	176	2,250,000	12,800	20½″	30″	7.10	7.20	1.61	1.63	8200	8000
150	171	2,700,000	15,780	22½″	32″	7.60	7.71	1.725	1.76	8580	8290
Avg.									Ft. per min.	7610	7414

SINGLE-ACTING HORIZONTAL (NATURAL GAS)

20	290	370,000	1,275	9½″	14″	3.32	3.33	0.76	0.77	6120	6010
50	225	830,000	3,690	13½″	20″	4.70	4.74	1.06	1.07	6940	6820
100	185	1,590,000	8,600	18″	26½″	6.20	6.25	1.42	1.43	7530	7420
125	176	2,040,000	11,580	20″	29″	6.90	6.98	1.56	1.58	7960	7710
150	171	2,330,000	13,620	21″	30¾″	7.22	7.32	1.65	1.68	8140	7900
Avg.									Ft. per min.	7338	7172

DOUBLE-ACTING HORIZONTAL (PRODUCER GAS)

50	192	1,100,000	5,730	16″	22½″	5.5	5.58	1.25	1.27	6670	6470
100	146	2,120,000	14,500	21½″	32″	7.3	7.4	1.66	1.68	7300	7100
125	134	2,640,000	19,700	23½″	36″	8.1	8.2	1.83	1.87	7420	7170
150	125	3,160,000	25,250	25½″	39½″	8.7	8.8	1.98	2.00	7650	7480
200	113	4,200,000	37,100	28¾″	45½″	9.84	9.95	2.23	2.25	7900	7820
Avg....									Ft. per min.	7388	7208

SINGLE-ACTING VERTICAL (GASOLINE)

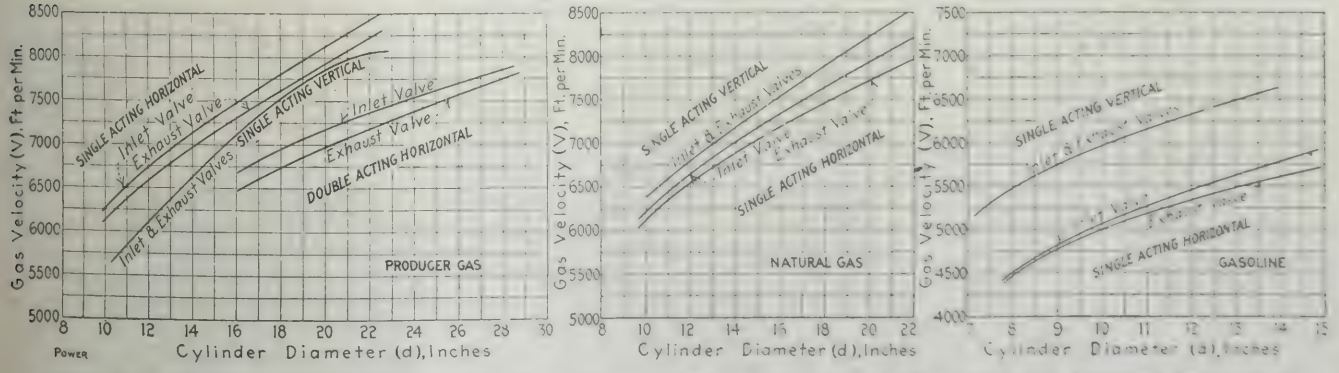
10	397	170,000	428	7½″	8½″	2.46	2.46	0.56	0.56	5140	5140
20	332	333,000	1,000	9½″	11½″	3.25	3.25	0.73	0.73	5850	5850
30	296	500,000	1,690	11⅝″	13″	3.90	3.90	0.876	0.876	6110	6110
40	275	660,000	2,400	12½″	14½″	4.36	4.36	0.98	0.98	6440	6440
50	260	825,000	3,170	14″	16″	4.80	4.80	1.08	1.08	6620	6620
Avg....									Ft. per min.	6032	6032

SINGLE-ACTING VERTICAL (PRODUCER GAS)

20	332	380,000	1,215	10½″	11½″	3.52	3.52	0.80	0.80	5630	5630
50	260	920,000	3,540	14⅝″	16⅝″	5.02	5.02	1.13	1.13	6760	6760
100	223	1,810,000	8,120	19½″	21½″	6.65	6.65	1.48	1.48	7780	7780
125	215	2,250,000	10,450	21″	23½″	7.21	7.21	1.63	1.63	7980	7980
150	208	2,700,000	13,000	22¾″	25¼″	7.82	7.82	1.78	1.78	8080	8080
Avg..									Ft. per min.	7246	7246

SINGLE-ACTING VERTICAL (NATURAL GAS)

20	332	370,000	1,113	10″	11½″	3.41	3.41	0.71	0.71	6380	6380
50	260	830,000	3,190	14½″	15.7″	4.80	4.80	1.00	1.00	7200	7200
100	223	1,590,000	7,130	18½″	20.8″	6.35	6.35	1.33	1.33	7840	7840
125	215	2,040,000	9,480	20½″	23″	7.00	7.00	1.47	1.47	8275	8275
150	208	2,330,000	10,600	21½″	23½″	7.30	7.30	1.53	1.53	8700	8700
Avg....									Ft. per min.	7679	7679



FIGS. 34 TO 36. SHOWING THE RELATION BETWEEN CYLINDER DIAMETER AND GAS VELOCITIES FOR ENGINES RUNNING ON PRODUCER GAS, NATURAL GAS AND GASOLINE, RESPECTIVELY

For present purposes let the value of $d^2l = a$ or $l = \frac{a}{d^2}$. Then substitute this in the equation showing the relation between diameter and stroke for the particular type of engine and kind of gas as derived from Fig. 12, article I. For a single-acting horizontal engine using any fuel this is

$$d = 0.677l + 0.4$$

Substituting for l,

$$d = 0.667\,\frac{a}{d^2} + 0.4$$

or

$$d^3 = 0.667\ a + 0.4\ d^2$$
$$d^3 - 0.4\ d^2 = 0.667\ a$$

Assume that a or $d^2l = 630$, then

$$d^3 - 0.4\ d^2 = 0.667 \times 630 = 420$$

whence

$$d = 7¾ \text{ in.}$$

and since $d^2l = 630$, $l = 10½$ in.

After finding d and l, the corresponding inlet and exhaust valve diameters and lifts for the given cylinder diameters are read from the curves in Figs. 29 to 33, which show the relations between d_v and d, and d_v and l_v. These values are then substituted in the equation for V, namely,

$$V = \frac{d^2\,ln}{24\,d_v l_v} \text{ ft. per minute}$$

The curves in Figs. 34, 35 and 36 show the foregoing calculated velocities in feet per minute plotted against cylinder diameter for the various gases.

Attention is called to two typographical errors appearing in the first of the series of articles "American Practice in Rating Internal Combustion Engines" in the July 23 issue. The tenth line of the third column, page 121, should read

$$b.hp. = \frac{d^2\,ln}{16{,}400} - 0.5 \text{ (Curve 2, Fig. 5)}$$

Also on page 123, near the bottom of the third column, the diameter of the cylinder should be 19 in. instead of 16 in.

Observations on Cooling Cylinders

By James H Beattie

A good gas engine will convert into work from 25 to 30 per cent. of the heat in the fuel burned, 70 to 75 per cent. being radiated or lost in the exhaust. About 30 per cent. of the total heat of the fuel is radiated from the cylinder jacket and this must be carried away by some cooling medium, usually water, but sometimes air or certain grades of oil. Engines having jacketed cylinders for water-cooling are by far the most common, air-cooling being used only for small cylinders, while oil is not so well adapted as a cooling medium on account of its inability to take up and give off heat as rapidly as water.

Several types of water-cooled engines are in use; the most common is the ordinary closed jacket, in which the water is circulated either by a thermo-siphon system, or by a pump. When the pump is used, it is employed to assist the natural tendency of the heated water to rise in the system and allow the cold water to replace it. Among the arrangements for cooling the water after it is heated by passing through the engine cylinders, the oldest and most common, for stationary engines, is the circulating tank, but this has the disadvantage that it must be very large for a comparatively small engine. A 25-hp. engine requires a tank of at least 2000 gal. capacity. Other devices for cooling the water more rapidly include cooling sprays, radiators, such as used in automobile engines, and motor-driven fans, which force a strong blast of air through the spray.

An improperly cooled cylinder will wear rapidly and interfere seriously with the operation of the engine, for lubrication is made difficult and preignition, with pounding and excessive speed fluctuation, is likely to occur. The thermal efficiency of an engine is higher the hotter the cylinder; hence, for economy, the cylinder should be kept near the highest allowable temperature. For the ordinary closed-jacket engine of moderate size, the discharged water should be at 160 to 180 deg. F. With hopper-cooled, or open-jacket engines, the temperature is usually about 212 deg. F. The temperature of air-cooled engines is higher still, but all the engines designed to run at high temperatures have more allowance in the fit of the pistons and rings to permit greater expansion.

Accidents due to improper cooling are extremely common, especially with small engines, where the attention given them is limited. The following cases illustrate how lack of attention to these details often leads to serious results. A small horizontal portable engine had been in constant use for several weeks, and on the day the accident occurred had been in operation for 14 hr. As it was some distance from the machine it was driving, it received only an occasional visit from the operator. The stud bolts holding the cylinder to the frame sheared off, and the cylinder blew back, breaking the connecting rod and the piston. Examination showed that the jacket was nearly filled with lime, the oil cup was empty and the piston and cylinder perfectly dry. The piston had seized the walls of the cylinder so strongly that portions of the metal between the piston rings had been torn off. Fortunately the cylinder was uninjured and by fitting new stud bolts and a new piston and connecting rod, the engine was again in good condition. The lime was removed by placing a 10 per cent. solution of muriatic acid in the jacket and allowing it to stand over night. After this accident, rain water was substituted for the well water previously used, and more attention was given the oil cups.

The operator of another engine complained that it overheated badly. It had been in constant operation for several years and had always given excellent satisfaction. The cooling system consisted of a large tank and a belt-driven circulating pump, taking make-up water from a nearby stream. Investigation proved that the jacket was clogged with leaves and sediment drawn into the tank from the stream. The pump was able to keep up some circulation, but not sufficient to properly cool the engine.

In a similar instance, the trouble was from practically the same cause, but occurred in a different way. The engine was a small portable type, used for operating a sawmill, and overheated badly, burning out the cylinder-head packing. Leaves falling from nearby trees, it was found, had entered the jacket, clogging up the space so that water could not circulate, and causing the overheating. For those not familiar with the open-type jacket, it should be stated that it has an extension, or hopper, large enough to hold a gallon or two of water per horsepower. This water evaporates, the steam carrying away the heat, and the water is replaced from time to time as required. This system is used only on engines up to 25 or 30 hp., and is very economical and convenient, owing to the small amount of water necessary.

Another operator complained that the exhaust pipe of his engine and the exhaust-valve chest kept very hot, but the cooling system was found to be all right. The engine used excessive fuel for the work which it was doing, and the trouble proved to be in the carburetor. The needle valve regulating the flow of gasoline to the cylinder had been jammed so hard that the brass valve seat was burred until the opening to the air intake was larger than originally; consequently, the opening of the needle valve remaining the same, more fuel was allowed to pass than when the engine was new. An exceedingly rich mixture resulted and accounted for the large fuel consumption and also for the exhaust-valve casing and exhaust pipe overheating, as the mixture was still burning when the exhaust valve opened near the end of the power stroke. Substituting a new seat for the needle valve remedied the difficulty.

The adjustments of the valves, the carburetor and the ignition of the air-cooled engine are important. The owner of an air-cooled engine complained that it would run but a short time without overheating. His exhaust valve was found to open too late in the stroke, which was responsible for the overheating. The valve should have opened with the crank at least 30 deg. from the end of the power stroke, but was opening almost on center.

Other cases have been found where overheating was due to a combination of all three factors, improper valve timing, improper adjustment of the carburetor and too late ignition.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Means for Accumulating Refrigeration—II

By Charles H. Herter

To date the effect of air agitation upon k, the coefficient of heat transmission per degree wall temperature difference, has received scant attention, and it is to be hoped that those having the knowledge and facilities to solve this interesting problem will soon attack it.

In April, 1909, Walter Kennedy, Pittsburgh, tested Nonpareil corkboard 2 in. thick, using a fan in and outside of the test box and obtained k = 3.2 and 3.3 B.t.u. for the 2-in. thickness.

In 1911, John E. Starr tested 2-in. Union corkboard, using ice in the box and no fan, and derived k ranging from 3.31 to 3.48 B.t.u.

In 1910, Prof. G. F. Gebhardt, Armour Institute of Technology, Chicago, tested Nonpareil and Union corkboard under identical conditions and found their conductivity c to be exactly alike.

In 1910, Prof. F. L. Pryor, Stevens Institute of Technology, Hoboken, N. J., made tests on Star corkboard. A box built of 2-in. corkboard, with incandescent lamps inside, was placed in a room at 31 deg. F. Using the fan in the box, k averaged 3.595 B.t.u.; and without the fan, 3.428 B.t.u. When the 2-in. box contained ice, and was placed in a room at 80 deg. F., k without a fan averaged 3.495 B.t.u. That these tests are not conclusive as to the effect of air agitation on k, is shown by results from tests on a box of 4-in. Star corkboard which were conducted at the same time: With the box in the cold room, using heat and a fan in the box, k was 1.74 B.t.u.; and without the fan, 1.736 and 1.772 B.t.u. through the 4-in. thickness.

From the above it would seem that the difference between c and k might be neglected, but the following additional data show that a difference exists that should not be neglected.

H. Recknagel's investigations on stone walls show a_1 to coincide approximately with the values obtained from hot-water and steam-heating pipes, or as in Table 2.

In analyzing other tests, where both the surface and the air temperature were actually taken and no effort was made to agitate the air, the writer found a_2 over a vertical plastered wall to increase with the insulating value of the wall, as follows:

c B.t.u.	k B.t.u.	a_2 B.t.u.	$t_1 - t_4$ deg. F.
5.68	3.904	24.95	7.1
3.99	3.102	27.97	5
1.939	1.739	33.6	2.42

Prof. Pryor's tests furnish a direct contradiction to this, his values of a_1 and a_2 (combined coefficients of radiation and convection at the warm and the cold side of wall respectively) with 4-in. cork figuring out to about half of the values found with 2-in. cork. In his test on 2-in. Star corkboard, where k = 3.495 B.t.u., the rate of heat flow H was

$$3.495 \times 26.4 \text{ deg.} = 92.27 \text{ B.t.u.}$$

and as the air was still, a_1 may be taken

TABLE 2. RECKNAGEL'S INVESTIGATIONS ON STONE WALLS

Velocity of air at 1 ft. from wall		a_1 in B.t.u., per sq.ft. per deg. diff., $t_1 - t_{2'}$ per 24 hr.
Ft. per sec.	Miles per hr.	
0	0.0	30
1	0.68	37
3	2.05	58
5	3.40	79
10	6.82	120
15	10.22	150

at 30 B.t.u., and a_2 at 30 B.t.u. Thus $t_1 - t_2$ and $t_3 - t_4$ will each be

$$\frac{92.27}{30} = 3.076 \text{ deg.}$$

and the wall temperature difference, $t_2 - t_3$, will be

$$26.4 - (2 \times 3.076) = 20.25 \text{ deg.}$$

therefore

$$c = \frac{92.27}{20.25} = 4.556 \text{ B.t.u.}$$

for 2-in. or 9.112 B.t.u. per inch thickness per 24 hr., or 0.38 B.t.u. per hour. In this case c is 30 per cent. larger than k.

In the tests on the 4-in. box without a fan, k averaged

$$\frac{1.736 + 1.772}{2} = 1.754 \text{ B.t.u.}$$

Multiplying by $t_1 - t_4$ = 57.8 deg., H = 101.4 B.t.u. Using the same coefficient for inner conductivity as above, namely

$$\frac{9.112}{4 \text{ in.}} = 2.278 \text{ B.t.u.}$$

$t_2 - t_3$ must have been equal to

$$\frac{101.4}{2.278} = 44.51 \text{ deg.}$$

and $t_1 - t_2$ equal to

$$\frac{57.8 - 44.51}{2} = 6.645 \text{ deg.}$$

Thus a_1 and a_2 equal

$$\frac{101.4}{6.645} = 15.26 \text{ B.t.u.}$$

and

$$\frac{c}{k} = 1.3$$

Table 3, giving the comparison of figures from tests on the same boxes, using air agitation inside, shows a proportionate reduction in the values of a_1 and a_2 between 2-in. and 4-in. corkboards, the temperature drop at the surfaces of the thick board apparently remaining undiminished.

In agitating the air, $t_1 - t_2$ and $t_3 - t_4$ are decreased, the result being that k approaches the value of c; but with a fan it is hardly possible to eliminate entirely the temperature drop at the surfaces. If, with the 2-in. corkboard, the air could be agitated inside and outside with a speed of 15 ft. per second, $t_1 - t_2$ would probably equal

$$\frac{199.1}{150} = 1.327 \text{ deg.}$$

$t_3 - t_4$ also 1.327 deg., making a total of 2.654 deg., which, added to 46.8, makes $t_1 - t_4$ equal to 49.454 deg., and

$$k = \frac{199.1}{49.454} = 4.026 \text{ B.t.u.}$$

Then

$$\frac{c}{k} = \frac{4.253}{4.026} = 1.056$$

Slight differences in the material and treatment of compressed corkboard cause

TABLE 3. RESULTS OF TESTS ON CORKBOARD BOXES

Symbols	2-in. Corkboard Box	4-in. Corkboard Box
k	3.559 B.t.u.	1.74 B.t.u.
$t_1 - t_4$	55.95 deg.	60.6 deg.
H	199.1 B.t.u.	105.4 B.t.u.
a_1	79 B.t.u.	35.15 B.t.u.
$t_1 - t_2$	2.52 deg.	3 deg.
a_2	30 B.t.u.	13.18 B.t.u.
$t_3 - t_4$	6.64 deg.	8 deg.
$t_2 - t_3$	46.8 deg.	49.54 deg.
c	4.253 B.t.u.	4.253 / 2 B.t.u.
$\frac{c}{k}$	1.195	1.221

the conductivity to vary from 3 to 5 per cent. either way from the average. Straining for extreme accuracy is therefore useless.

Adopting the value of c = 0.38 B.t.u. per hour, and reverting to the formula given in the first article for finding the value of c for a compound wall, the heat transmission through the 20-in. wall of the building per square foot per degree of wall temperature difference is

$$\frac{1}{\frac{16}{5.56}+\frac{2\times\frac{1}{2}}{4.2}+\frac{3}{0.38}}=0.0909\ B.t.u.$$

per hour, or 2.18 B.t.u. per 24 hr.

While the air circulation inside a room may be very feeble, wind might prevail outside. When the air velocity is about 10 ft. per second (a gentle wind) the radiation and convection will increase to 5 B.t.u. per hour per degree difference between the air and wall, and then k, the heat transmitted through the 20-in. wall per degree air temperature difference, will be

$$\frac{1}{\frac{1}{5}+\frac{16}{5.56}+\frac{2\times\frac{1}{2}}{4.2}+\frac{3}{0.38}+\frac{1}{1.25}}=\frac{1}{12.01}$$

$$=0.0833\ B.t.u.$$

per hour, or 2 B.t.u. per 24 hr., as against 2.18 B.t.u. when the air was quiet outside as well as inside.

This clearly shows the influence air movement has upon the transmission coefficient k, misleading one unless properly defined. On the other hand, the coefficient per degree wall temperature difference is a fixed quantity and is the only proper value to use.

Another common error is that in most tests made in this country the air temperatures only are taken, the resulting coefficient representing k and not c.

The heat transmission for 1-in. corkboard, for example, is then simply taken as double that of 2 in., because heat conduction varies inversely as the thickness. However, the latter rule applies only with true conductivity from point to point inside of the material, not when referred to temperatures obtaining outside of the test piece. For instance, with the above 2-in. corkboard, the heat actually transmitted through 1-in. corkboard per degree air-temperature difference, assuming 25 deg. wall temperature difference, is

$$25\times 9.13=228.25\ B.t.u.$$

per 24 hr. Dividing by 30, the coefficient of total radiation with still air, the surface difference is 7.6 deg., and the whole air-temperature difference $t_1 - t_i$ is

$$7.6+25+7.6=40.2\ deg.$$

Hence the heat transmission actually desired (for instance, when computing temperature curves) is

$$\frac{228.25}{40.2}=5.68\ B.t.u.$$

not 2 × 3.5 or 7 B.t.u. However, c, figured here at 9.1 B.t.u. for 1-in. thickness is the value actually required.

Having determined the heat transmission through the walls—namely 2 B.t.u. per square foot per 24 hr., or 0.0833 per hour per degree air-temperature difference—calculations can be made as to their neutralizing effect. Evidently the outer wall surface will average only a trifle cooler than 79.6 deg., the average air temperature outdoors at 7:52 a.m. If the refrigerating machine has been kept running until then, the curve of temperature through the walls will be as in Fig. 1. The heat transmission per square foot per hour will be

$$(79.6-32)\times 0.0833=3.965\ B.t.u.$$

The outside surface-temperature drop

$$(a-b)=\frac{3.965}{5}=0.793\ deg.$$

The inside surface-temperature drop

$$(f-g)=\frac{3.965}{1.25}=3.172\ deg.$$

Thus b will be

$$79.6-0.793=78.807\ deg.$$

The points c, d and e can be obtained

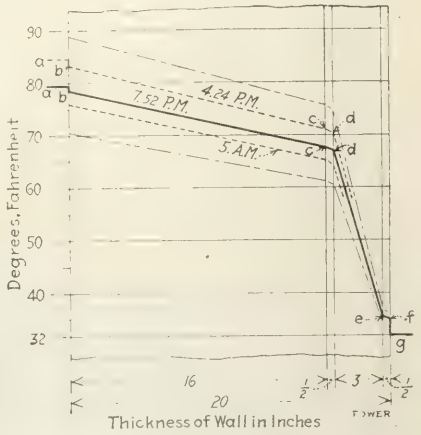


FIG. 1. TEMPERATURE CURVES

from the inner heat conducted through the materials as follows:

The flow of heat through each layer being 3.965 B.t.u. per hour,

$$c=78.807-\frac{3.965}{\frac{5.56}{16}}=67.397\ deg.$$

$$d=67.397-\frac{3.965}{\frac{4.2}{\frac{1}{2}}}=66.925\ deg.$$

$$e=66.925-\frac{3.965}{\frac{0.38}{3}}=35.705\ deg.$$

$$f=32+3.172=35.172\ deg.$$

From these data the mean temperature of the several layers comprising the wall is, for the 16-in. brick wall, 73.102 deg.; first ½ in. of cement, 67.161 deg.; 3-in. corkboard, 51.315 deg.; second ½ in. of cement, 35.438 deg.

The purpose here in calculating intermediate temperatures to an accuracy of several decimals is to have the various temperature drops agree in total with the entire wall-temperature difference. In actual measurements, probably none of the readings would come within 1 deg. of the figures given.

The table of hourly air temperatures given in the previous article shows an increase as high as 90 deg. during the day, which would seem to add materially to the heat transmission. It is therefore interesting to investigate the influence of this higher temperature upon the temperature curve of the wall. The average air temperature between 7:52 a.m. and 7:24 p.m. is obtained at 4:24 p.m. and is 84.2 deg. At the latter temperature the heat flow per hour will be, say

$$(84.2-32)\times 0.0833=4.348\ B.t.u.$$

per square foot. The dotted line in Fig. 1 labeled 4:24 p.m. represents the curve of temperatures obtaining under this condition, namely: a = 84.2 deg., b = 83.33 deg., c = 70.82 deg., d = 70.302 deg., e = 35.985 deg., f = 35.48 deg., g = 32 deg.

The mean temperature of the several layers will be, for the 16-in. brick wall, 77.075 deg.; first ½ in. of cement, 70.561 deg.; 3-in. corkboard, 53.143 deg.; second ½ in. of cement, 35.733 deg.

The amount of heat required to thus increase the temperature of one square foot of insulated wall is:

For 16-in. brick wall,

$$123.5\times\frac{16}{12}\times 0.22=36.2\ B.t.u.,$$

$$\times 3.973\ deg.=143.8\ B.t.u.$$

For first ½ in. of cement,

$$78\times\frac{\frac{1}{2}}{12}\times 0.2=6.5\ B.t.u.,$$

$$\times 3.4\ deg.=22.1$$

For 3-in. corkboard,

$$12\times\frac{3}{12}\times 0.25=0.75\ B.t.u.,$$

$$\times 1.867\ deg.=1.4$$

For second ½ in. of cement,

$$6.5\ B.t.u.\times 0.333\ deg.=2.16$$

Total 169.46

The average air temperature of 84.2 deg. occurs at 4:24 p.m.; hence just nine hours were available for increasing the temperature of the wall.

$$\frac{169.46}{9}=18.83\ B.t.u.$$

per hour. In the meantime, heat has entered the room at a rate of 4.348 B.t.u. per hour, or a total of 23.178 B.t.u. per hour. Dividing by 5, the coefficient of heat transfer at the surface, the average difference between air and surface must have been 4.63 deg. and

$$84.2-4.63=79.6\ deg.$$

the air temperature at 7:52 a.m. This calculation shows that when the machine is in operation, the outer-wall temperature does not increase more than

$$83.33-78.81=4.52\ deg.$$

above its normal; therefore the flow of heat into the room during the day cannot exceed

$$(83-35.2)\times 0.0909=4.345\ B.t.u.$$

per square foot per hour.

Before the heat flowing into the room during the night can be determined, the extent of heat abstraction must be known which takes place at the outer surface of the wall, due to the declining air tem-

perature. To determine this quantity accurately involves a rather intricate calculation; it will be close enough to figure again on cooling at the rate of 5 B.t.u. per hour per degree temperature difference between the outer surface and the air. At 7:24 p.m. and 5 a.m. the air temperature averages 74.5 deg. The wall surface during this interval may fall to 76 deg., thus the average difference will have been

$$\frac{83 + 76}{2} - 74.5 = 5 \text{ deg.}$$

and the heat abstraction

$$5 \times 5 \times 9.6 \text{ hr.} = 240 \text{ B.t.u.}$$

Between 7:24 p.m. and 5 a.m. the heat entering the room is about

$$4 \times 9.6 = 38.4 \text{ B.t.u.}$$

per square foot. Thus there are

$$240 - 38.4 = 201.6 \text{ B.t.u.}$$

to be abstracted from the 20-in. wall. To lower the mean temperature of this wall 1 deg. requires

$$36.2 + 13 + 0.75 = 49.95 \text{ B.t.u.}$$

In abstracting 201.6 B.t.u. from the wall its mean temperature will be lowered by

$$\frac{201.6}{49.95} = 4 \text{ deg.}$$

Fig. 1 gives the temperature curves for 4:24 p.m. and 5 a.m., and the dash-and-dot lines the curves which obtain if there is sufficient heat to raise the outer wall temperature close to 90 deg., and enough heat is abstracted to cool it at night down to almost 70 deg. Accordingly the flow of heat into the room may be taken at 4.345 B.t.u. per square foot per hour during the day, and about 4 B.t.u. per hour during the night. Just at 5 a.m. it is

$$(76 - 35) \times 0.0909 = 3.727 \text{ B.t.u.}$$

A 3-deg. rise in temperature of the room contents is obtained after an inflow of

$$\frac{57{,}800 \text{ B.t.u.}}{1800 \text{ sq.ft.}} = 32.11 \text{ B.t.u.}$$

per square foot of wall surface. During the day a stop could therefore be made for

$$\frac{32.11}{4.345} = 7.39 \text{ hr.}$$

and during the night for about eight hours, neglecting such sources of heat as doors, lights, etc.

Fig. 1 also shows that only very little refrigeration can be stored up when the insulation is placed at the inner or room-side of the brick walls. A 16-hour stop would let the room temperature rise to 38 deg.

By placing the insulation at the outside of the walls, as in Fig. 2, their average temperature is kept quite low and from their proximity to the cold air they influence the temperature rise. The material used is the same as before with a ½-in. finishing coat of portland-cement plaster added on the room side, making the entire wall 20½ in. thick.

The 20½-in. wall might be expected to exert its retarding influence directly refrigeration is discontinued, but while the room contents are colder than the inside wall surface there is no reason for a diminution in the rate of heat supply to the room. How closely these theoretical considerations apply to practice remains to be determined by experiment.

The heat transmission c through this wall per square foot per hour per degree wall-temperature difference equals

$$\frac{1}{\frac{3 \times \frac{1}{2}}{4.2} + \frac{3}{0.38} + \frac{16}{5.56}} \quad \frac{1}{1 \quad .125}$$

$$= 0.0899 \text{ B.t.u.}$$

and k per degree air-temperature difference, still assuming wind on the outside of the building, equals

$$\frac{1}{\frac{1}{5} + 11.125 + \frac{1}{1.25}} = \frac{1}{12.125} = 0.0825 \text{ B.t.u.}$$

This is 1.98 B.t.u. per 24 hr., as against 2 B.t.u. with the 20-in. wall. The air-temperature difference is

$$79.6 - 32 = 47.6 \text{ deg.}$$

and the wall-temperature difference

$$47.6 \times \frac{0.0825}{0.0899} = 43.68 \text{ deg.}$$

the difference between the two being 3.92 deg., one-fifth of which, or 0.784 deg.,

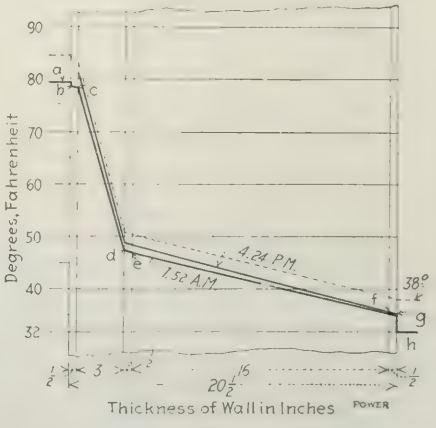


FIG. 2. INSULATION PLACED OUTSIDE OF WALLS

is outside of the wall, and $\frac{1}{1.25}$ or 3.136 deg., inside. The lowest temperature curve in Fig. 2 can now be readily computed: $a = 79.6$ deg., $b = 78.816$ deg., $c = 78.348$ deg., $d = 47.354$ deg., $e = 46.886$ deg., $f = 35.586$ deg., $g = 35.136$ deg. and $h = 32$ deg.

This curve is supposed to apply at 7:52 a.m. The total heat transmission is then equal to

$$(79.6 - 32) \times 0.0825 = 3.927 \text{ B.t.u.}$$

per hour.

To show how little the temperature of the brick wall is now affected by temperature changes outdoors, a second curve is drawn in representing the conditions obtaining at 4:24 p.m., the end of the warming period. In this curve the heat transmission is

$$(84.2 - 32) \times 0.0825 = 4.306 \text{ B.t.u.}$$

per hour.

Even with this construction a 3-deg. temperature rise in the room would be reached after a stop of eight hours, because the brick wall only comes into play when the inner wall temperature rises. When the machine is in operation the room is normally at 32 deg. and the wall surface at 35.14 deg. It has been shown that with the 20-in. wall the room temperature would be 38 deg. after a stop of 16 hours. The question is, how many hours can a plant be shut down with this 20½-in. wall before the room is up to 38 deg.? The dotted line in Fig. 2 shows a curve beginning with the maximum outer surface temperature of 83.34 deg. and ending with 38 deg. inner surface temperature, the heat transmission being then

$$(83.34 - 38) \times 0.0899 = 4.076 \text{ B.t.u.}$$

per hour.

To reach 38 deg., which means a 6-deg. rise,

$$32.12 \times \tfrac{6}{8} = 64.24 \text{ B.t.u.}$$

must enter the room. On the basis of 4 B.t.u. per hour, this takes

$$\frac{64.24}{4} = 16.06 \text{ hr.}$$

plus the time required to raise the wall temperature over the curve of 7:52 a.m. The difference in mean temperatures of the two curves for the inner 17 in. of wall thickness is 3.092 deg. Multiplying by 49.2 B.t.u., the heat capacity of this portion of the wall, 152 B.t.u. is found to be required to thus heat up the wall.

As the heat is being supplied only at 4 B.t.u. per hour, it evidently takes

$$\frac{152}{4} = 38 \text{ hr.}$$

additional or a total of 54 hr. before the room temperature is up to 38 deg. The 38-hr. gain over the other construction is entirely due to placing the walls within the cold zone instead of, as usual, out in the warm zone where they can do no good. With a thinner brick wall the time would be reduced. No saving is made in refrigeration, for as soon as the machine is started again the neat accumulated must be removed and the temperature of 32 deg. regained, but in many plants a shutdown, weekly or daily, would be a great convenience, dispensing with the third or even the second shift of operators.

A plain solid wall, the heavier the better, erected at the inside of the cold room, preferably using the building walls themselves by leaving the insulation outside, is the most rational expedient for accumulating refrigeration. Hollow tile is not recommended because its heat capacity per cubic foot is less than for a solid wall, and the air spaces are of doubtful insulating value because as

built they usually form communicating air channels likely to gather moisture. Their ability to resist fire also leaves much to be desired.

Exterior insulation is still novel in this country, but deserves consideration. Aside from the valuable feature of accumulating refrigeration by means of the cold walls, as explained, the overall insulating effect will be improved because the wall is protected from wind, rain and frost, a damp wall conducting heat readily. Further advantages are decreased wear and tear of building, protection in case of fire, especially on the roof, because modern insulating materials do not support combustion.

The advantage of continuity incidental to interior insulation might be urged as an argument against exterior insulation, but when the insulation on a wall, for example, is carried up 2 ft. or more above the interior ceiling line the same effect is obtained because the heat is then separated from the interior by that amount of brick. On the other hand, the steel framework conducts a great deal of heat into the building, sometimes piercing the insulation at every beam, column and floor, while exterior insulation could be erected without a single break. It could be applied to most old and new buildings, and when laid in the forms of new concrete buildings the cost should be much lower than that of interior application.

In the next article, the application of brine-storage tanks and the piping therefor will be gone into, these being in common use and necessary where there are no other means for accumulating refrigeration.

Troubles in an Absorption System and Remedies

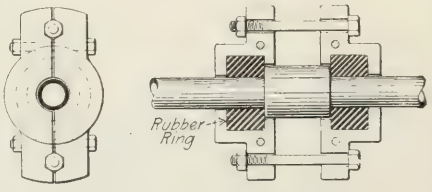
By W. S. Luckenbach

In the issue of July 30, some of the troubles that may perplex those handling an absorption refrigeration system were discussed. The following takes up other troubles and their remedies:

Failing Capacity

It may happen that a machine has been making its full capacity, but is gradually failing, although it is apparently working as usual with the rich and poor liquor the same, no leaks and the temperature of the cooling and distilled water the same. To locate the trouble and apply a remedy several points must be considered. There may be more lids off the freezing tank than usual, thereby exposing more ice cans to the warm atmosphere, or the temperature on top of the bath may have increased, due to additional warm air from the machine room or elsewhere. The liquid ammonia pipe may be subjected to additional heat that would not be detected unless the pipe had a thermometer close to the expansion valve. Referring to a standard table of properties of saturated ammonia gas it will be found that a difference in the increase of temperature from 70 to 75 deg. F. decreases its heat-absorbing power nearly six times.

The importance of maintaining as low a temperature as possible of the liquid anhydrous ammonia at the expansion valve cannot be overestimated, and the writer would no more think of running without a thermometer inserted near the expansion valve than to discard the thermometer in the brine tank. For instance, if certain results are obtained with a certain temperature at the expansion valve and the temperature increases, say 5 deg. (which no one can detect by simply grasping the pipe) the thermometer will show it at once. Otherwise one might wait until the brine had gone up several points and then wonder what was the cause. With the thermometer one would know at a glance that probably either the condensing and cooling coils needed cleaning or needed more water, or, if neither of these reasons accounted for the decrease in capacity, then the proper amount of liquid ammonia was not passing the expansion valve, or moisture had



Split Flange Collar for Stopping Leaky Threads

gotten into it by the liquor having been too high in the generator.

The presence of moisture can be readily ascertained by drawing some of the ammonia into a piece of 1-in. pipe about 1 ft. long capped at one end and having a handle securely attached so as to hold it off some distance when drawing. A little poured into a shallow vessel to evaporate, will show by the amount of water left the proportion of moisture. If the liquid ammonia was originally dry it will become so again by carrying the liquor in the generator a trifle low for a day or two.

Another cause for an ammonia pump not pumping its full capacity is its becoming partly gas bound. To correct this it is well to have a connection from the poor liquor coil at the absorber to the pump suction pipe; then, by partially closing the valve on the suction pipe at the absorber and opening the poor liquor connection wide, the poor liquor passing into the pump will absorb all the surplus gas in a few minutes. Too much brine agitation will increase its temperature. Enough agitation to keep a fairly even temperature is all that is required; more than that is a loss.

Decreasing Liquor Density

Another perplexing thing is to have the densities of both the rich and poor liquors decrease without any apparent cause. If the loss amounts to, say, 1 deg. in a few days, without any odor being noticeable, gas is evidently escaping through leaks in the pipes or joints of either the condensing, cooling or absorber coils or pipes, and as the cooling water passes over, or through them, it absorbs the escaping gas, so that the latter is not detected unless the water is tested with the usual solution for that purpose.

If the condensing coils are of the atmospheric type, the leak can be easily located by a whitish substance which in a few days will form a crust. It is plain that these leaks must be stopped, and without shutting down. The illustration shows an excellent split-flange collar, recessed for two pure gum rings ¼ in. thick, bolted to a pipe coupling for stopping leaky threads. It can also be used on any fitting or return bend. The flanges are simply rough cast iron with no machine work except drilling one bolt-hole; the other one is cored. The writer has used these for years, always successfully. It is well to keep on hand a few for the various pipe sizes, in case of an emergency.

Importance of Temperature Variations

A 5-deg. variation in temperature of the distilled water has more effect in the production of ice than may at first be realized. Raising or lowering the temperature 1 deg. means adding or abstracting 1 B.t.u. for each pound of water. Hence, by lowering the temperature 5 deg. in a 50-ton machine (2000 lb. per ton)

$$2000 \times 5 \times 50 = 500{,}000 \text{ } B.t.u.$$

of heat less need be extracted by the expanded gas. This is equivalent to about one ton of ice per 24 hours on a 50-ton machine, allowing also for meltage in removing the ice from the cans. This again demonstrates the importance of paying attention to seemingly unimportant details.

Concerning Tank Paints

The writer would be glad if some of the readers of this department who have had experience along the lines indicated, would answer the following questions:

Are paints containing an excessive amount of oil deleterious to iron coils used in refrigerating work, or to tanks containing salt or calcium chloride brine, or to water tanks?

Is black asphaltum paint the best or the only good one to use on new coils, or coils that have been in the salt brine 12 years and then cleaned to be used in calcium chloride brine?

B. A. Warren.

Rochester, N. Y.

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POWER

Superheated Steam

In our issue of July 23 is a resumé by Professor Dwelshauvers-Dery of a remarkable treatise by his friend and pupil, Dr. Armand Duchesne. Professor Dwelshauvers-Dery is professor emeritus at the University of Liége and founder of its steam-engineering laboratory where Dr. Duchesne's experiments which form the basis of the treatise were made.

One point brought out by the experiment is of special interest. The temperature of steam was measured simultaneously by two mercury thermometers, one with its bulb immersed in the steam itself, the steam passing out through a stuffing-box, one in a thermometer cup filled with oil, and by a very sensitive thermocouple. The current of steam flowed very sluggishly past the temperature-measuring devices in the order named, and when the steam was saturated their indications were substantially the same. But when the steam was superheated, a very considerable difference developed between the mercury thermometers, especially that in the oil well, and the thermocouple, amounting in some cases to over 80 deg. C., or about 150 deg. F.

The thermocouple is very sensitive to impact, and it is difficult to get with it reliable indications of the temperature of the mass in which it is immerged if that mass is in rapid motion. Efforts to explore, for temperature, the axis of a jet of expanding air and steam with a thermocouple consisting of a fine wire stretched through the jet axially, resulted in the indication of 80 or 90 deg. for the final temperature, when by the relation of pressures and temperatures with adiabatic expansion the temperatures should be below zero, and the jet showed by refraction the presence of ice particles. In this case the junction was surrounded by steam flowing with a high velocity, and its impact and friction evidently produced an indication of a temperature certainly above that of the mass. But in the Duchesne experiments the couple was in a current of steam moving very sluggishly and hardly subject to serious disturbance from impact. Furthermore, its indications coincided with those of the thermometer and with the temperatures given by the steam tables for saturated steam of the observed pressure.

Why did it read so much higher than the thermometer for superheated steam?

The reason given by Dr. Duchesne, and it is interesting as bearing upon the behavior and usefulness of superheated steam, is that while the steam is saturated its temperature is the same throughout. It cannot get any higher than that due to its pressure, and if it gets any less the steam will condense and more steam of the fixed temperature will take its place. The temperature of the mass immediately surrounding the thermometer and the thermocouple is in this case the same, and the readings are practically identical.

Superheated steam, however, can cool off without condensing, and is a very poor conductor. A film of such steam about the thermometer cup becomes cooled by transmitting some of its heat to the metal. It clings around the cup and the thermometer records its temperature and not that of the hotter mass. The thermocouple was made of wires as fine as hair and of so little heat capacity that they were not similarly affected.

This explanation furnishes a reason why superheated steam, notwithstanding its greater temperature, loses less by radiation when passing through a steam main than does saturated steam. It appears to be a well authenticated fact that less heat will be radiated from a pipe filled with superheated steam at a high temperature than from one filled with saturated steam at a considerably lower temperature. The explanation also accounts for the well known reluctance of superheated steam to give up its heat to the cylinder walls, whereas the instant the steam becomes saturated the heat exchange is prompt and vigorous. Many of the experiments upon superheated steam have been made with mercury thermometers. It will be interesting to see

what the effect of this phenomenon has been upon their results, and whether its exploitation may not explain some of the inconsistencies which exist between the results of different experimenters.

Value of a Sketch

In commenting on an article which appeared in the July 19 issue under the heading "Preparing for License Examination," a contributor in this issue, page 236, adds a suggestion relative to the value of a sketch.

Ability to convey ideas by making a sketch is very valuable to an engineer, and the importance of acquiring this art cannot be too strongly emphasized. Some are fortunate enough to be gifted with this ability; others acquire it only by tedious effort. Most of us are in the latter class. This is because sketching is, contrary to the general belief, difficult, unless it is a natural gift. It is a fact that the majority of highly proficient draftsmen cannot make good freehand drawings and yet can produce very accurate work with tools.

As an instance of where a sketch was valuable, the following is interesting:

A German engineer unable to speak English was being examined by an American board unable to speak German, so the board and the applicant could not understand each other. The German, a technical graduate familiar with the proper procedure of examinations, conceived the idea of showing what he knew of boilers, design of joints, furnace construction, and engine and valve design, etc., by sketches made on a blackboard. He received a license without further questioning because he proved his familiarity with power-plant equipment by his ability to sketch it.

To the young engineer the sketching of things about the plant is one of the best means of acquiring a lasting knowledge of the things which he operates. In examining the object to be sketched he discovers details that otherwise would have escaped attention.

The value of the sketch does not end here. As the contributor suggests, a sketch will often tell what words fail to convey. But if one finds it difficult to express himself in words there is no better way of overcoming that trouble than by drawing sketches. To produce an object on paper he must first study it analytically. Unconsciously he acquires the coveted attribute of being logical and soon finds that his mind can direct his tongue as well as it guides his hand.

One Schedule is Wanted

A comparison of the two standards of flanged fittings and flanges is made on other pages of this issue and a statement issued by the manufacturers giving their reasons for not adopting the "1912 U. S. Standard" is also published. A study of the article first mentioned will show the exact differences between the two and furnish a basis on which to judge the contribution supplied by the manufacturers.

For the standard fittings it may be noticed that most of the variations occur in the dimensions for long-turn elbows and those for reducing fittings. The societies' schedule maintains uniform face-to-face dimensions for both the straight and reducing fittings of a particular diameter, while the manufacturers make the length of the reducing fitting the same as the old standard, and it is thus shorter than the straight fitting. With the manufacturers' new standard the fittings now in a plant may be replaced with no change in the piping. This, of course, would mean a temporary convenience for the consumer, but in the long run would it not be of greater convenience to be able to interchange straight and reducing fittings, or *vice versa,* as the case required? Apparently this is one of the strong points of the "1912 U. S. Standard."

There is some difference between the two standards on Y-fittings and reducers, and in the extra-heavy fittings and flanges the dissimilarities are the same as those mentioned for fittings of standard weight. For extra-heavy flanges for piping nine inches in diameter and above, the societies increase the bolting beyond the requirements of German or English practice with which the manufacturers' new standard closely agrees. In the opinion of the manufacturers the increase is excessive, and on the larger flanges the bolts are so large as to be cumbersome and inconvenient from the steamfitters' standpoint.

Convenience is a factor that should be considered, but above everything else is safety. With any doubt concerning the strength of the bolting it is better to err on the safe side. Both parties no doubt have good reasons for the adoption of the individual schedules, but the differences are not so great as to prohibit settlement by a joint committee. Personal opinion or prejudice should not influence the issue, nor should *temporary* expense to the manufacturer or inconvenience to the consumer be given primal importance. One schedule, and only one, with convenience giving first place to safety, is wanted.

A joint committee is the solution. A little give and take on both sides would settle all differences. It is up to the society committee and the manufacturers to throw aside all personal considerations, to get together, and to agree on a schedule that will last as long as the present materials and pressures are employed.

The Massachusetts Formula for Circular Segments

There has been considerable speculation as to the derivation of the formula prescribed by the Massachusetts Board of Boiler Rules for the areas of circular segments. It is said that the chief inspector dug it up out of an old note book, but that neither he nor anybody else knew how it happened nor why it gave the right answer. Some of the applicants for license have been asked lately how this formula was derived, possibly as a test of their frankness. The derivation is explained by Prof. Marshall, of the Sheffield Scientific School, on page 240.

How about that heating system? Don't be like the man who couldn' shingle his roof when it rained and didn' shingle it when it didn't rain because i didn't need it. Now is the time, when i is out of commission, to comb out of th system all of those kinks and snarl which unattended to will make th winter's work harder and the service les satisfactory.

If *The Engineer* had given Professo Christie credit for the sage remarks o European turbine practice in relation t bearings given in its issue of July 2 page 93, it would not be quite so respo sible for a misstatement made. Turbin bearing surface speeds of 60 ft. per se ond (not minute) are common practic The same mistake appears in the origin paper by the professor.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Providing for Pipe Expansion

An engineer was piping a large vertical boiler from which a long pipe was designed to discharge steam into a header. When this pipe was ready to make the final connection it was 3 in. too short. He turned steam into it, and when heated it was just the right length. This shows how a pipe expands, but when this pipe is cooled, it will make a strong effort to regain its former position. Unless changing lengths are provided for, a pipe line is sure to be subjected to heavy strains, that may cause ruptures.

Another engineer was laying a long line of pipe, the end of which was 7 in. from a brick wall. When steam had heated this pipe he was surprised to find an ell on the end of it, hard against this wall. When cool it went back to its original position.

Fig. 1 illustrates the main steam pipe of a plant, showing where it comes through a brick wall and enters a separator just above the throttle valve. Four ells were used and ample provision was thus made for expansion, but its appearance is very crude and unsatisfactory. The main part coming through the wall might have been continued as shown by dotted lines, then brought forward and turned downward to connect with the separator shown. Two ells would have sufficed and expansion would do no harm, as the pipe could spring enough to compensate for it.

Fig. 2 is the exhaust-steam piping from the cylinders of a double engine. The piping, as will be seen, is so arranged that the pipe from each cylinder can expand without damaging the other. The Y-fitting shown in connection with a 45-deg. ell is recommended because it allows exhaust steam to escape freely, with less friction and back pressure than if a common tee was used. Gate valves allow either exhaust pipe line to be shut off, if one cylinder is not used.

Fig. 3 illustrates a section of piping in another plant which has proved satisfactory. The upper pipe is carried straight through, as there is ample room for it to expand. This expansion does no harm to the branch lines, as they are long and free to move. In the lower pipe the expansion must be divided, hence the ells and extra piping used cause steam to follow an indirect course where it passes through the brick wall shown, but this arrangement admits of expansion without causing leaks or other evidence of excessive strains.

Fig. 4 shows the provision made for expansion in a long line of 8-in. pipe, carrying a light pressure. Two long sweep bends are used to carry the pipe 10 ft. out of a straight line, two more bring it into line again. This constitutes a 10-ft. offset, and the pipe extends beyond these points in both directions. It works well in practice, as no leaks have developed in 10 years' service.

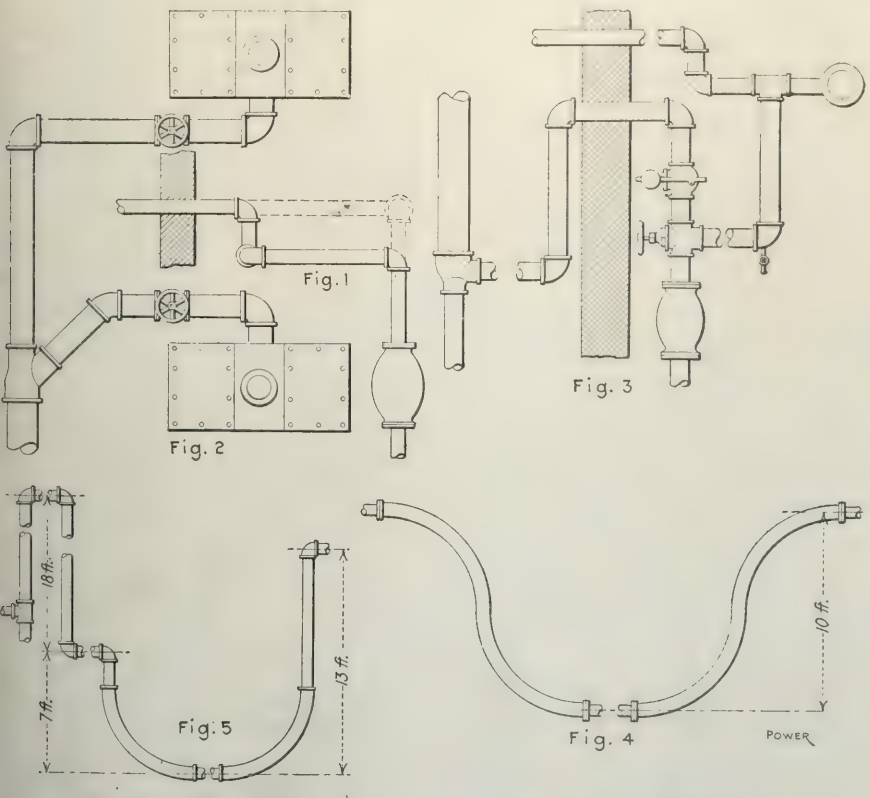
Fig. 5 shows a line of 10-in. pipe which might have been carried in a straight line (as indicated by dotted lines), and fitted with an expansion joint to provide for variations in its length, but it was decided to adopt offsets instead.

Starting at the right, an ell followed by a piece of pipe carries it 8 ft. out of line, then a long sweep bend gives 5 ft. more. Then come several lengths of straight pipe, followed by another bend and a short pipe which carry it back 7 ft. A right-angle turn and more pipe follow, until there was a suitable place for another ell. The next length is 18 ft., then it is extended 14 ft., followed by a turn at right angles to the main line, which gives ample chance for expansion without causing leaks.

This arrangement has given satisfactory service, hence the principles involved are recommended for application to similar conditions. The piping shown passes through four brick walls. In all such cases the pipe should rest on substantial rollers well anchored, so as to move easily when expanding and contracting.

W. H. WAKEMAN.

New Haven, Conn.



STEAM-PIPE CONNECTIONS, SHOWING PROVISION FOR EXPANSION

The Massachusetts License Law

The new clause in the Massachusetts license law, that went into effect Jan. 1, 1912, seems to have filled a long felt want for men seeking a second- or first-class license.

Applicants are now sure of an impartial examination with three of the examiners to pass judgment. Under the old system there was only one examiner which dissatisfied many engineers with the result of their examination. With the new order of doing things, the applicants say that they are satisfied with the examina-

tion. Some making this remark did not receive any license, and in one case the applicant was one of the "hard knockers" of the old administration.

Does not this show that the engineers are pleased with the new law?

W. L. NESBIT.

Springfield, Mass.

Repairs with Plastic Metal

For many repair jobs about the plant, plastic metal* is useful. It will run into

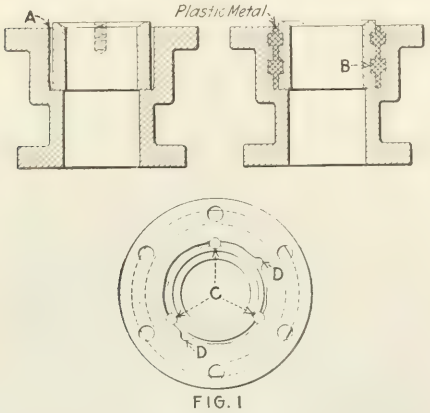


FIG. 1

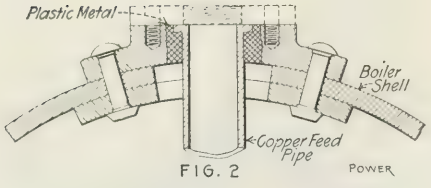


FIG. 2

SHOWING CHECK VALVE AND REINFORCING FLANGE REPAIRED WITH PLASTIC METAL

almost any shape at a low heat and is hard and tough enough to hold under high pressure. For lining bearings, thrust collars, bushings, glands, coating plungers, etc., it has few equals. Price is the drawback, but where time is money the metal is cheap at any cost.

It helped me out in several cases where the job seemed hopeless, as the following instances will show: The boiler-feed check-valve seats had been loose for some time, but no stop could be made for overhauling them, until the body of the castings had worn as at *A*, Fig. 1. The worn seat was turned, as shown at *B*, and holes drilled and tapped vertically in the casing and seat to admit screw pins *C* for reinforcement. The seat was centered in place and heated, after which the metal was run in through the openings *D*. The metal was then lightly calked. It made a good job and works well under 165 lb. pressure.

The next case was a repair to a nipple for the top feed pipe in a return-tubular boiler. The feed line to the boiler was of ordinary steel pipe, and the down pipe in the boiler was of 2-in. gas pipe connected to a reinforcing flange, which was riveted to the shell, as shown. The whole outfit pitted badly and the outside lengths of pipe were replaced several times. I suspected that the internal feed pipe was in bad condition, and when dismantled it was found that it had been eaten off as if by sulphuric acid.

The solid steel flange was eaten away almost to the stud holes, as shown in Fig. 2. A temporary job was made with a separate false flange (shown dotted) into which a new pipe was screwed. The internal copper pipe tinned and slightly flanged, as shown, was passed into the boiler and held flush with the ring. The metal was then poured in and well calked around the copper pipe, after which the temporary flange was connected. The boiler pressure in the case was 140 lb. and the job has held without any trouble.

J. McA. HOWDEN.

Melbourne, Australia.

*Any alloy of ordinary fusing temperature sufficiently high to safely withstand the temperature of the steam would do for such work.

Pressure Chart for Comment

This recording steam-gage chart shows the pressure maintained in our plant for 24 hours. We have one 500-hp. and one 250-hp. water-tube boilers, hand fired with buckwheat mixed with duff* coal of which about 2300 lb. are burned per hour. Forced draft of 0.3 in. of water is maintained, and a damper regulator is also provided. The fires are cleaned three times during a 24-hour run.

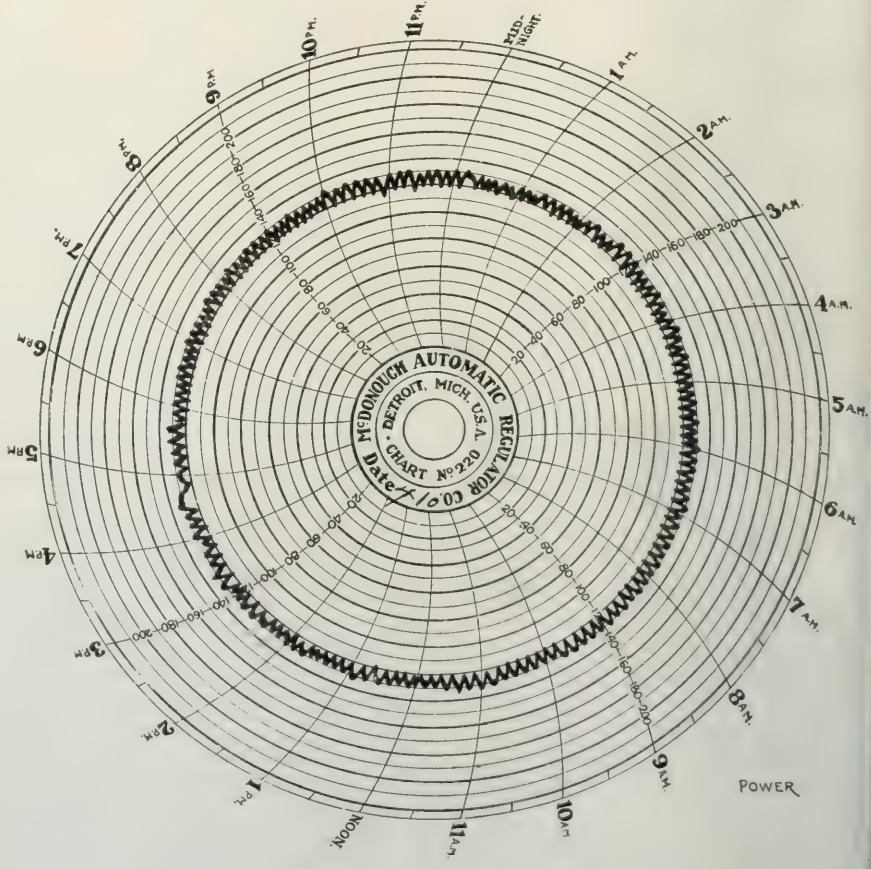
Until recently the boilers had automatic stokers which were dispensed with on account of poor economy and other troubles. We now burn nearly 20 per cent. less coal, keep the furnace walls in better condition and obtain a more uniform steam pressure than with the stokers. The load carried is the same as when the stokers were in use.

I would appreciate comment on the chart and would like to see more about results with stokers and comparisons of charts obtained with automatic and hand-fired furnaces.

BURT M. SEYMOUR.

Delphos, Ohio.

[*"Duff" is a fine coal, the name being given to washed bituminous slack.—EDITOR.]



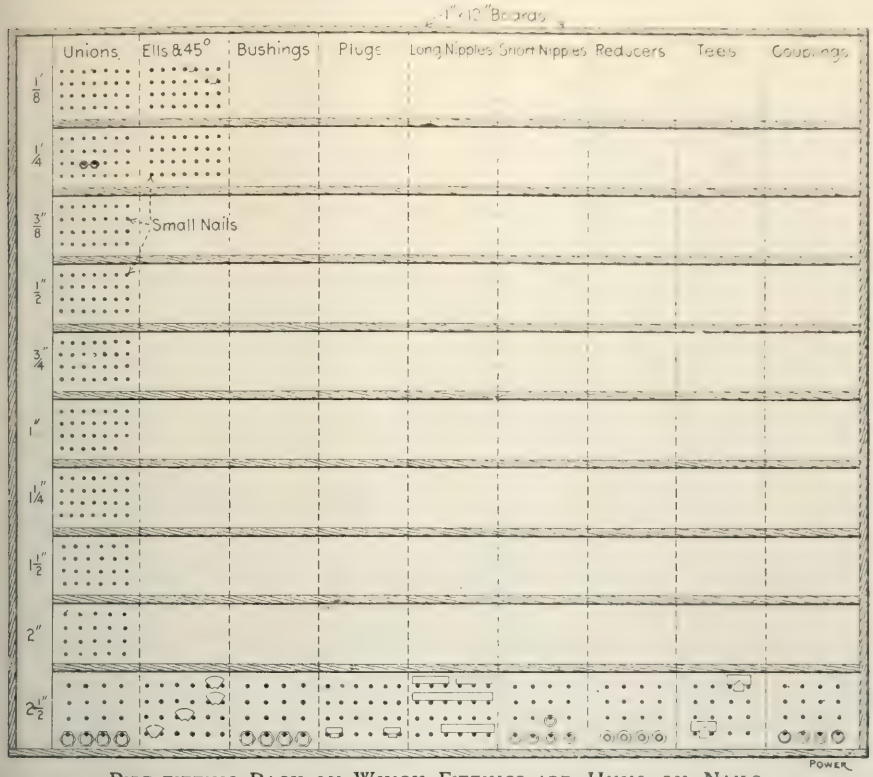
STEAM CHART FROM HAND-FIRED BOILERS

Rack for Pipe Fittings

The pipe-fitting rack illustrated is designed to take the place of those which consist of nail kegs, barrels, old boxes and very often floor space. The design may not be new, but the rack has proved very valuable in our plant. It consists of a number of spaces arranged in rows and labeled for the kinds of fittings which are to be hung in each division. Instead of the usual rows of boxes or bins, nail

are driven and the fittings hung on them

It can be cheaply constructed, and will not take up much room. It has all the desirable features that go to make for simplicity and time saving, when properly arranged and a place for each fitting designated. Generally provision for ⅛ to 2½-in. fittings will be found sufficient. All of the fittings are out in plain view, which is also advantageous.



PIPE-FITTING RACK ON WHICH FITTINGS ARE HUNG ON NAILS

Dirt does not accumulate in this rack as it does in boxes, and no boards are needed to build it. Any wooden wall will answer, but if appearance is a consideration, a smooth board facing can be added if needed and painted to make it more attractive. With smaller nails for small fittings driven close together, the maximum capacity for a given space can be realized. Care should be taken in driving nails for hanging ells and tees and two nails instead of one should be used for each fitting.

Other fittings, such as short nipples up to 3 in. long, can be hung on single nails driven slightly slanting. For longer nipples the nails should be in a row close together, as illustrated.

It is well not to allow odds and ends of pipe with fittings attached to lay around, but to strip all that are useful and hang them on the rack, throwing the others into a barrel or box kept for that purpose. It is a bad habit to save everything from a damaged flywheel to a broken hacksaw blade, thinking that they may be "good for something." Junk is junk and should be treated as such by selling it when it accumulates. In this way the "good for something" is realized to the best advantage.

The orderly storing of stock should be carried farther by arranging labeled boxes, or bins to contain screws, bolts, washers, nuts, valve wheels, disks, etc.

JOHN J. RAMSEY.

Wellington, Kan.

Sawdust Stops Belt Slipping

Our air compressors are belted to motors, the pulleys of which are of paper. These compressors would always give trouble when the riveting floor was full of bolted-up material to be riveted. The belts would slip, and down would go the air pressure until it was impossible to drive a tight rivet. Rosin and a dozen kinds of belt dressing were tried, but they only seemed to work while they were being put on. It was also difficult to apply dressing on account of the location of the belts.

This trouble became so great that we decided to coat the pulleys with anything that would correct the slippage. The sawdust under a crosscut saw in the template room looked good so we decided to try it. Glue was suggested as a sticking medium and Saturday afternoon as the time, as it would set over Sunday.

The motor belt was taken off and the machine turned over by hand until an even coat of glue covered the entire face of the pulley. A pan full of sawdust was held under the pulley, which was kept turning until its entire surface was covered. The result was gratifying as all the next week the machines served by that compressor kept going, while the other compressor "fell down" as usual. In fact, the sawdust did so well that a couple of extra hammers were supplied by the compressor without reducing the pressure.

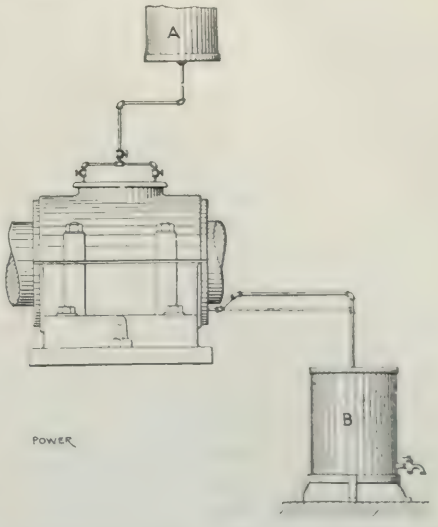
The next Saturday the other compressor-motor pulley was sawdusted, and the riveter boss who had been kicking for more air wanted to know what happened. Not only was it an easy and cheap way to correct the complaint, but the cost of power to run the compressors with the slippage eliminated, was less than it had been.

A. E. DIXON.

Cleveland, Ohio.

Lubricating a Large Bearing

The illustration shows a ring-oiling bearing for a 6⅛-in. shaft driven from a 500-hp. engine by a 48-in. belt. The distance between centers is very short, requiring a very tight belt to develop the required power. The tight belt caused excessive friction, resulting in hot boxes. Oil was fed to the bearings in a continuous stream from an elevated tank *A*. The end near the bottom of the box was tapped for a ⅜-in. connection, which was piped to a can *B*, as shown by the dotted lines. This arrangement kept the oil recess in the bottom of the box from filling, as the oil coming in at the top would run right through and out at the bottom.



ARRANGEMENT FOR OILING LARGE BEARING

After having considerable trouble with heated bearings it was suggested that the pipe be raised, as shown by the full lines, allowing about 1½ in. of oil in the bottom of the box before it would run out, thus giving the rings a deeper pool of oil to turn in. This furnished sufficient lubrication, as no further trouble was experienced.

J. W. DICKSON.

Memphis, Tenn.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Value of a Sketch

To supplement W. Nelson's article on "Preparing for License Examination" in the July 19 issue, I would like to offer a suggestion that seems right in line with his idea. It is that the young engineer should learn early to sketch. He should learn to draw, but he must at least learn to make an intelligible free-hand sketch, and rapidly, too, if he is to succeed in his work. He may never be called upon to produce a carefully finished drawing, but every day, and many times a day, he will meet occasions when a few deft strokes of a pencil will convey ideas, in an instant, which words could hardly have made distinct. A subordinate wants instructions—make him a sketch. The manager wants to know just how you propose overcoming a difficulty—let the lead pencil answer him.

It is not necessary to wait till one reaches a position of responsibility before he reaps any benefits. They come immediately. Every time one makes a sketch of an object, he learns something about that object. Putting the lines on paper makes a lasting impression on the memory.

Begin with something easy—say a pump valve—and work gradually up to more and more difficult things. Try to get the idea with the fewest lines. Try to get in every necessary dimension—and no more. After the sketch is made, put yourself in the other fellows' place and see if you can make the article from the data given.

WILLIAM E. DIXON.

Malden, Mass.

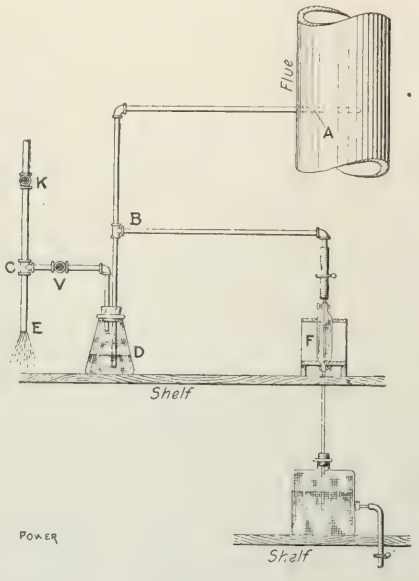
[Editorial reference to this letter is made on page 232.—EDITOR.]

Flue Gas Sampler

Charles M. Reed's description of a flue-gas sampler, given in the July 16 issue, is a little misleading as to the reason for collecting the gas over mercury, and is in error as to the manner of obtaining the sample.

If it is for immediate analysis, as generally, a sample taken over water, previously saturated with chimney gas, can be measured accurately enough for practical purposes. The gas will be saturated with moisture when the analysis is made, the percentage of moisture depending on the temperature of the gas when analyzed, amounting to 1.16 per cent. at 62 deg. F. and 5.56 per cent. at 92 deg. F., thus reducing the dry gas contained in the sample, and causing a difference in the sum of the oxygen contents.

If the sample is allowed to stand for an hour or so, the CO_2 will be materially reduced by absorption in the water over which it is collected. For this reason the mercury sampling bottle as described by Mr. Reed, has been adopted by the Bureau of Mines. This form of sampling was described in a bulletin issued by the Bureau of Mines, and in an abstract by J. C. W. Fraser and E. J. Hoffman, in POWER, Aug. 22, 1911.



PIPE CONNECTIONS FOR FLUE-GAS SAMPLER

Mr. Reed's method of sampling the gas is not practical, for the reason that if steam were allowed to blow through flask *D* the water there contained would soon be evaporated, and there would be only a steam discharge from *E* and no indication of a suction from the flue.

The proper connections are shown herewith. The pipe *A* leading from the flue has one end immersed in the water in the flask *D*, while a branch leads to the sampling bottle *F*. From a tee connection *C* in the steam pipe *K*, a branch is led to *D*, with its end well above the water level. The object of the flask *D* is simply to indicate a proper gas flow, while the steam aspirator is to insure a fresh sample being always drawn from the point of sampling, so that there will be no lag, due to the excess capacity of pipe *A* over that of the bottle *F*.

To operate, regulate valves *K* and *V* so that a moderate agitation is noted in *D*, due to the gas being drawn through the water. When the pipe *A* has been exhausted of all dead gas, open the top cock on the bottle *F*, and regulate the lower cock so that the rate of flow of the mercury will empty the bottle in the time over which the sample is to be taken. The sample is obtained by the suction of the mercury leaving *F*.

O. Z. HOWARD.

New York City.

I wish to thank Mr. Howard for clearing up the reasons for taking flue-gas samples over mercury, and also for calling attention to and correcting the error in the method of obtaining these samples.

While it is true that samples collected over water will absorb moisture, the error in the CO_2 content will, as Mr. Howard says, be materially affected. The Bureau of Mines, in the bulletin referred to by Mr. Howard, says in part:

"If water is saturated with furnace gas containing 12 per cent. of carbon dioxide and is then used in collecting a sample from gas having a carbon dioxide content of 8 per cent., the water saturated with gas containing 12 per cent. of carbon dioxide will give up some of the dissolved gas, and the sample will be found to contain more than 8 per cent. of carbon dioxide. The magnitude of the error introduced in this way will vary with conditions, but it will depend largely on the length of the sampling period, and the total time of contact of the sample with the water."

Upon reading Mr. Howard's criticism I made a very thorough examination of a plant to which I have access, and to my chagrin discovered that the connections were made essentially as he has shown them, and not as shown in my drawing. I am very glad that he called attention to the error.

CHARLES M. REED.

Annapolis, Md.

Taking Steam from Water Column Connections

Mr. Wakeman's letter in the June 3 issue wherein he shows, in Fig. 1, a bad arrangement of water-column connections, reminds me of a similar case. I

was recently called as an expert witness in a court trial to tell what I thought of such an arrangement where, however, the connection was made to the top water-column connection for a stack blower. The boiler was of the portable firebox type and would not generate enough steam without the blower. There is hardly any doubt that the explosion which occurred and killed the fireman and injured two others was caused by this arrangement. At any rate it cost the owners of the boiler about $12,000 to satisfy the courts.

C. R. SUMMERS.

Chattanooga, Tenn.

Results of Burning Screenings

Mr. Fieux's comment under this heading in the June 18 issue on my article on burning coke screenings greatly interested me, and I will be glad to learn the results he has obtained in detail, if he will furnish them.

If he will refer to the article in question he will find under "Average Pressures and Temperatures" the pressure in the ashpit in inches of water for the first test to be 0.877 and for the second 0.972. These pressures are the actual average of observations taken throughout the tests and are not, as he suggests, the results of computations. The point I wished to make in the paragraph from which he quoted, was that the rate of combustion determines the steam pressure required at the blowers. That is, an increase in the amount of coke fired calls for an increase in the amount of air supplied, which, in turn, since all the air is supplied by the blowers, demands an increased steam pressure at the blowers.

Mr. Fieux states that "this (frequent leveling of the fire) may be good in testing for maximum efficiency, but is extremely poor boiler-room practice"; it is our experience that unless the fire is leveled frequently, a hot fire cannot be maintained. The inconsistency in our experience may be due to the physical properties of the coke itself. While the analysis of the coke burned by Mr. Fieux may be identical with mine, the size may be entirely different. With screenings running as large as ¾ in. in diameter, it is comparatively easy to produce good results. On the other hand, I am anxious to learn how coke that is so fine that at least two-thirds would pass through a $^{8}/_{16}$-in. screen, approximately half through a $^{8}/_{32}$-in. screen, and which contained very little that would go over a ½-in. screen, can be burned without frequent leveling. These figures are merely an estimate, but they show approximately the size of the coke used in the tests. Most of the stuff looks more like gray sand than anything else.

If Mr. Fieux has had experience with coke as fine as this, I would like to learn how his methods of firing differ from those outlined in my article.

The steam used by the blower was obtained by a calibration of the blowers actually used in the tests. This calibration was made by immersing the blowers in a barrel of water on a platform scale and weighing the steam condensed per unit of time for different pressures at the nozzle. From these observations a curve was plotted showing the steam consumed per unit of time corresponding to the different steam pressures at the nozzle. As there was little variation in the steam pressure, which was measured during the course of the test, the steam used was calculated from the point on the curve corresponding to the average pressure at the blowers.

One significant statement is that he "obtained a rate of combustion of 25 lb. per sq.ft. of grate area." He does not say whether with dry or natural coke. Compared with this, in the first test we obtained a rate of combustion of 45.88 lb. per hr. per sq.ft. of grate as fired, or 36.98 lb. per hr. per sq.ft. of grate, and in the second 48.25 lb. per hr. per sq.ft. of grate as fired, or 39.52 lb. per hr. per sq.ft. of grate, dry. This may be the reason he obtained an efficiency 10 per cent. greater than mine.

Another reason why my efficiencies were not higher was that the moisture in the coke in each case was nearly 20 per cent., and no correction was made for this.

In making our calculations, too, the steam used by the blowers in each case was deducted before calculating the efficiencies. I suppose that Mr. Fieux also made this correction.

J. F. MOWAT.

Joliet, Ill.

The accompanying data of three tests made at various localities with various kinds of coke breeze may interest Mr. Mowat. The worst kind of coke breeze being used is the Solvay, a 72-hr. coke, which is slow burning and generally considered impossible to burn with any efficiency and capacity.

Test of Horizontal Return Tubular Boiler

Method of starting and stopping	Alternate
Grate surface, sq.ft.	25.21
Heating surface, sq.ft.	1321.1
Kind of furnace—Regular—Pin hole grate	
Percentage of air space	.9
Kind of coal. Solvay breeze—(fine) 72-hour coke.	

Total Quantities

Duration of trial, hr.	7.3
Weight of coal as fired, lb.	6,648.0
Percentage of moisture	10.49
Total dry coal consumed, lb.	5,950.
Total ash and refuse, lb.	1,784
Percentage ash and refuse	29.98
Total water fed to boiler, lb.	28,358.7
Factor of evaporation	1.07042
Equivalent evaporation to dry steam, 100 deg. C., lb.	29,293.2

Average Temperature and Pressure

Steam pressure, lb.	96.4
Feed water temperature, deg. F.	181.5
Stack gases, temperature, deg. F.	546.8
Ash pit pressure, in.	1.45
Horsepower developed	116.3
Percentage of rated horsepower	95.04

Economic Results and Efficiency

Water evaporated per pound of coal, lb.	4.405
Equivalent evaporation per pound of coal, lb.	4.
Equivalent evaporation per pound of coal, lb.	4.922
Equivalent evaporation per pound of combustible, lb.	7.[illegible]
[illegible]	48.[illegible]
[illegible]	95.91

Analyses

Coal

	per cent.
Moisture	10.[illegible]
Volatile matter	[illegible]
Fixed carbon	74.[illegible]
Ash	11.97
Sulphur	0.[illegible]74
B.t.u. per pound of coal	10,987.0

Test of Babcock & Wilcox boiler

Method of starting and stopping	Alternate

Total Quantities

Duration of trial, hr.	10
Weight of coal as fired, lb.	17,721
Percentage of moisture	19
Total dry coal consumed, lb.	14,354
Total ash and refuse, lb.	2,202
Percentage ash and refuse	12.4
Total water fed to boiler, lb.	87,824
Factor of evaporation	1.2163
Equivalent evaporation to dry steam, lb.	105,818.5

Average Temperature and Pressure

Steam pressure, lb.	119
Feed water temperature, deg.F.	43
Stack gases, temperature, deg.F.	503
Steam pressure on blowers, lb.	75
Ash pit pressure, in.	0.40
Draft over fire (suction), in.	0.1
Horsepower developed	306
Percentage of rated horsepower	87.5

Economic Results and Efficiency

Water evaporated per pound of coal as fired, lb.	6
Equivalent evaporation per pound of coal, lb.	6.14
Equivalent evaporation per pound of dry coal, lb.	7.36
Equivalent evaporation per pound of combustible, lb.	8.42
Efficiency of boiler and grate, per cent.	53.6

Analyses

Coal

	per cent.
Moisture	19
Volatile matter	10.12
Fixed carbon	60.79
Ash	10.09
B.t.u. per pound of coal	13,280

Test of By-product gas-coke breeze, Sterling Boiler

To determine general economy and capacity	Method
Method of starting and stopping	alternate
Grate surface, sq.ft.	82.9
Heating surface, sq.ft.	2,870
Kind of furnace Sterling F.B. arch. Pin hole grates	
Percentage of air space, per cent.	6–7
Kind of coal, Coke Breeze—through ¼ in. mesh—about 1 per cent. ½ in. lump.	

Total Quantities

Duration of trial, hr.	7
Weight of coal as fired, lb.	14,750
Percentage of moisture	16.3
Total dry coal consumed, lb.	12,346
Total ash and refuse, lb.	2,657
Percentage ash and refuse	18
Total water fed to boiler, lb.	90,935
Factor of evaporation	1.075
Equivalent evaporation to dry steam, lb.	97,755

Average Temperature and Pressure

Steam pressure, lb.	148
Feed water temperature, deg.F.	187
Stack gases temperature, deg.F.	473
Typhoon Turbine blowers	
Ash pit pressure, in.	0.764
Horsepower developed	404
Percentage of rated horsepower	1.41

Economic Results and Efficiency

Water evaporated per pound of coal as fired, lb.	6.16
Equivalent evaporation per pound of coal, lb.	6.63
Equivalent evaporation per pound of dry coal, lb.	7.92
Equivalent evaporation per pound combustible, lb.	10.08

Flue Gases

Carbon dioxide (Chart)	13 per cent.

Mr. Mowat fails to mention the kind of grate and air space used during the test. With a rate of combustion of from 45 to 48 lb. his ashpit pressures of 0.877 and 0.972 in. seem exceptionally low. I believe, however, his error is due to the fact that he has taken the steam pres-

sure at the blower and the rate of combustion of the fire. The thickness of the fire should also be taken into consideration. The only accurate way of taking ashpit pressures is with a U-tube, and not calculating it, as has been done.

Mr. Mowat states that the pressures are the actual averages of observation taken throughout the test. I cannot see how they can be so, for such a remarkable rate of combustion is almost equal to the rate of fine soft coal, and with a poor grade of coke breeze it would be almost impossible. The coke must have burned freely; otherwise it would have been impossible to get the high rate of combustion.

I do not think that the size of the coke breeze enters into the quality of its burning as much as the chemical analysis and the method in which its various constituents are formed. It is well known that a 72-hr. breeze is very hard to burn, whereas an 18-hr. breeze is rather easy. I have seen coke breeze that was impossible to burn without being mixed with soft coal.

I believe, if Mr. Mowat would try a pinhole grate properly designed, he would find that the fire would not need leveling as often as in his previous test.

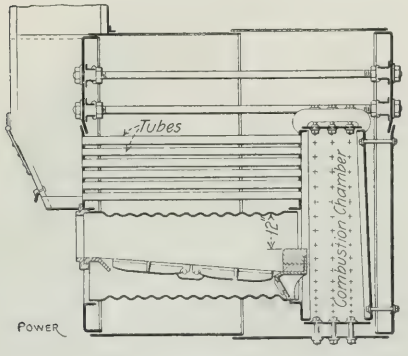
I would like to find out where the discrepancy between our results lies.

ERNEST D. FIEUX.

New York City.

Unnecessary Risks

Replying to L. A. Suverkrop's comment in the June 25 issue on my letter under this heading in the June 4 issue, I would



SCOTCH MARINE BOILER

state that the expression "crawling on their hands and knees" through the furnace of a Scotch marine boiler to plug a leaky tube without drawing the fire, is not literally true, as Mr. Suverkrop says.

It is not, however, absolutely necessary to draw the fire to get into the "back connection." On account of the sloping grate and fire, even a smothered fire and plank would be below the level of the bridgewall at that end. The bridgewall must be passed in any case, and in ordinary Scotch marine boilers it is 12 in. from the top of the wall to the crown of the furnace, as shown.

CHARLES J. MASON.

Scranton, Penn.

[Unless the fire is exceedingly heavy it is possible to crawl (not on the hands and knees) to the bridgewall on a plank laid over a smothered fire. This would be the worst kind of practice, however, as the gases would, in most cases, overcome the worker. It is questionable if this would ever be necessary and a man would be justified in disobeying an order to crawl in unless the fire was drawn.—EDITOR.]

Cylinder Condensation

If B. T. Abbott, who asks for information relative to cylinder condensation, in the issue of May 28, will look on page 88 of the catalog of the Buckeye Engine Co., he will find a rule, which, in my experience, has served well in accounting for the missing quantity. It was devised by Mr. Thompson, the designer of the original Buckeye engine, and is as follows:

12½ to 16½ times the ratio of expansion = missing quantity percentage of indicated.

The coefficient is to be varied with the range of temperature in the engines, the larger quantity with long ranges and intermediate, down to 12½ for shorter ranges. If good judgment is used in selecting the coefficient, this formula applies well to Corliss and moderate speed engines.

A formula, which takes into consideration nearly all variables, is the one devised by G. R. Bodmer, from experimental data quoted by Kent in his "Mechanical Engineer's Pocket Book," eighth edition, and is as follows:

$$W = C\frac{S\,(T-t)}{L\sqrt{N^2}}$$

in which

W = Weight of water condensed per minute;
S = Clearance surface area in square feet;
T = Mean admission temperature;
t = Mean exhaust temperature;
L = Latent heat of steam at mean admission temperature;
N = Number of revolutions per second.

The coefficient C varies according to conditions from 0.1 to 0.112.

In applying either of these formulas, experienced judgment is necessary to determine the proper coefficients to use; hence the results are likely to be misleading. Probably the practical man would get more accurate results from Thompson's empirical rule than from the more complex formula of Bodmer.

The investigation of condensation loss has gone on from the time of Hirn, and still we have no satisfactory formula for all cases. Those we have are based on variable, indefinite coefficients which are difficult to determine, and none take into account the ratio of compression which must certainly have a large influence. As to experimental data, the mere quotation of results which have been recorded with a given engine is futile, as the data could not be used in even an approximation with other than the original variables, a change in any one of which would destroy accuracy.

I have found food for thought in "Steam Using," by Prof. Smith, of Washington University, St. Louis, Mo., which quotes Hirn's experiments extensively, and also "The Steam Engine," by Prof. John Perry.

By applying Bodmer's formula with Mr. Abbott's engines it may be deduced that if clearance areas and ratio of expansion are identical in the two engines, the loss will also be identical, as in the 24x24-in. engine the surface exposed is relatively twice as large as in the 24x48-in., while the interval of time is only half as great. From a practical standpoint the short-stroke engine must have more compression and high compression produces high temperatures, so the range in the engine would probably be increased by high compression, which would increase its loss by condensation. This is probably the reason why long-stroke, low-speed engines nearly always excel short-stroke, high-speed engines in economy.

CHARLES F. PRESCOTT.

Los Angeles, Calif.

Turbine Repair

Under this heading in the June 11 issue, Mr. Martin mentions that 0.009 in. clearance between the bushing and valve spindle of the pilot valve was a contributing cause of the turbine overspeeding on light loads. This amount is not excessive. Other turbine makers use no needle valve at all, as the steam for lifting the pilot valve is taken through the clearance between the bushing and spindle. Unless the opening is nearly as large as the relief ports around the pilot valve, I do not believe excessive overspeeding would result. Under this condition the valve would open violently and close slowly, which would cause racing on light loads. The spring might also be broken, if subject to such action.

It seems that Mr. Martin's turbine has been run without sufficient lubrication for the valve piston which has become dry, and, of course, would not work freely. When he injected oil through the needle valve, the piston was lubricated and the trouble vanished.

E. A. HAWLEY.

Des Moines, Iowa.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Speed of Belt over Belt

When a belt wrapped around a driving pulley in the usual manner, carries another driving belt which is wrapped over the outside of the first belt, then in figuring the size or speed of a pulley driven by the outer belt, how much larger, if any, should the pulley be considered than its actual diameter?

H. R. C.

The pulley should be considered as increased in diameter by twice the thickness of the belt which is next to the pulley. For ordinary single leather belts allow for $\frac{3}{16}$ in. thickness, and for ordinary double belts allow for $\frac{3}{8}$ in. thickness of belt.

Object of Steam Pipe Anchorages

Is it not practicable to construct anchorages for steam pipes strong enough to prevent expansion and contraction from changes of temperature?

D. W.

The force exerted by iron when restrained from freely expanding or contracting from change of temperature is so great that it is not practicable to provide for changes in dimension due to changes in temperature of piping, except by establishing stationary points or anchorages to or from which other points will move by action of contraction or expansion, and thus control the direction of motion without resisting the natural amount of motion. When anchorages are properly employed, the strength required of them is only sufficient to resist the reaction of forces which are opposed to the free movement of the pipe in assuming a natural position with respect to the part held stationary at the anchorage.

Locating the Vacuum Line on a Diagram

Why is the vacuum line at sea level generally drawn on an indicator diagram below the atmospheric line at a distance in inches equal to 14.7 divided by the scale of the spring with which the diagram is made?

L. B.

Under average barometric conditions the pressure of the atmosphere at sea level is 14.7 lb. per sq.in. above zero, and as atmospheric pressure locates the atmospheric line at a height, which by the scale of the spring represents 14.7 lb. above zero, the zero or perfect vacuum line is to be drawn below the atmospheric line at a distance which represents 14.7 lb. per sq.in. less than atmospheric pressure.

As pounds pressure per square inch which any point on a diagram may be above or below atmospheric pressure is found in inches which the point is from that line, then, to find the distance in inches on the diagram which the vacuum line should come below the atmospheric line, we must reverse the operation; i.e., divide 14.7 lb. by the scale of the spring.

Back Water from Open Heater

In using an open feed-water heater with a back-pressure valve on the exhaust for steam heating, is a partial vacuum on the heating system likely to cause flooding of the engine cylinder; and is there not danger of water getting to the cylinder from leakage of the water-supply valve to the heater when the plant is shut down?

G. W.

Standard types of open heaters usually have float valves for regulating the supply of water so as to maintain the proper level of water carried, and glass gages for observing whether the water stands at a safe level; so that with ordinary attention there should be little danger of flooding from the heater during the time the engine is running. When the water-supply valves are leaky, there may be danger of water filling the heater so full as to back over to the engine when a plant is shut down, and to guard against such an occurrence a stop valve should be in the exhaust pipe between the engine and the heater for cutting off the connection between them when the engine is shut down for any length of time.

Resistance of Round vs. Square Wire

Each of two wires contains 100 cu.in. of annealed copper, one being round, 0.5 in. diameter, and the other a square wire 0.5 in. on a side. Which wire has the greater resistance? What is the length and resistance of each wire?

E. L.

The resistance may be found from the formula

$$R = p \times \frac{l}{A}$$

cross-sectional area in square inches, and p the specific resistance of an inch cube of the conductor.

For annealed copper wire the specific resistance in microhms per inch cube at 0 deg. C. is 0.6276. Hence for the square wire with cross-sectional area of $0.5 \times 0.5 = 0.25$ sq.in., the length would be 400 in.; the resistance would be

$$R = \frac{0.6276 \times 400}{0.25};$$

$$= 1004.16 \text{ microhms};$$

$$= 0.001004 \text{ ohms.}$$

For the circular wire the cross-sectional area being $0.5 \times 0.5 \times 0.7854 = 0.19635$ sq.in., the length will be 509.29 in. and the resistance would be

$$R = \frac{0.6276 \times 509.29}{0.19635} = 1627.85 \text{ microhms, approx.}$$

$$= 0.001628 \text{ ohms.}$$

Length of Eccentric Rod

What is considered to be the length of an eccentric rod of an engine and how is the proper length determined?

L. G.

The length of the rod is the distance from the center of the eccentric to the center of the pin which connects the end of the rod to the rocker-arm or other part of the valve motion. Set the rocker-arm, or other part to which the pin is connected, in such a position that it will be in the middle of its travel. For setting the length of the eccentric rod of a Corliss engine, the rocker should stand plumb when connected to the wristplate in its central position. If the engine has no wristplate, then with a tram scribe a mark from the pin on the engine frame or other stationary object. Next obtain the extreme forward and backward travel of the eccentric by loosening the eccentric and turning it around the shaft, or by turning the engine over, and with the same length of tram as used before, lay off the extreme distances the pin has traveled in each direction; or if a Corliss engine, mark the extreme points of travel made by the center mark of the wristplate. Find the central point between the two positions thus laid off and the distance it is from the central mark will indicate the amount the rod should be lengthened or shortened to bring the pin to the correct position for mid-throw of the eccentric and be the proper length for the rod.

Study Questions

This Week's Questions
Last Week's Answers

(66) When one of two engines in a power plant runs 20 per cent. underload and the other 20 per cent. overload their outputs are equal, but the combined output is 10 hp. less than the combined rated capacity. Under these conditions, what power does each engine develop and what are their rated capacities?

(67) The primary of a transformer contains 100 turns. How many turns must the secondary contain to increase the voltage from 110 to 2000?

(68) Each day for four days an engineer gives out 1 lb. of waste to one oiler, sends half the remaining stock to the boiler room and gives another pound to another oiler. He then has only 1 lb. left. How much had he in the beginning?

(69) How many gallons of water per minute will be raised to a height of 60 ft. by a hydraulic ram of 60 per cent. efficiency supplied with 100 cu.ft. of water per second under a head of 10 ft.?

(70) What is the absolute pressure in pounds per square inch corresponding to a manometer reading of 20 in. of water when the barometer reading is 29 in. of mercury?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(61) The heat equivalent of 1 hp.-hr. is 2545 B.t.u., and 1 hp. = 0.746 kw.; hence 1 kw. per minute would be equal to

$$\frac{2545}{60 \times 0.746} = 56.86 \text{ B.t.u.}$$

The energy lost in the form of heat by the current passing through the coil is expressed by the formula

$$W = I^2R$$

where W is watts, I the current in amperes and R the resistance in ohms.

Substituting

$W = (22)^2 \times 5 = 2420$ *watts or* 2.42 *kw.*

Since 1 kw. per minute is equivalent to 56.86 B.t.u., 2.42 kw. for 15 min. would be equivalent to

$56.86 \times 2.42 \times 15 = 2064.02$ *B.t.u.*

As 1 cu.ft. of water weighs 62.5 lb. and it takes 1 B.t.u. to raise 1 lb. of water 1 deg. F. at the temperature of the water, 2064.02 B.t.u. would raise 1 cu.ft.

$$\frac{2064.02}{62.5} = 33 \text{ deg. F.}$$

(62) The total heat of combustion of a ton of the coal is

$14{,}000 \times 2000 = 28{,}000{,}000$ *B.t.u.*

Since 970.4 B.t.u. evaporates 1 lb. of water from and at 212 deg. F., the ton of coal will evaporate

$$\frac{28{,}000{,}000}{970.4} = 28{,}854 \text{ lb.}$$

(63) The force actually exerted by the belt is the difference between the tensions on the two sides, or

$600 - 300 = 300$ *lb.*

and the distance through which it acts in 1 min. is the belt speed. This equals the circumference of the pulley multiplied by the revolutions per minute.

$14 \times 3.1416 \times 180 = 7916.832$ *in.*

or

$$\frac{7916.832}{12} = 659.736 \text{ ft.}$$

The work done is therefore

$300 \times 659.736 = 197{,}920.8$ *ft.-lb.*

and the horsepower

$$\frac{197{,}920.8}{33{,}000} = 5.9976 \text{ hp.}$$

(64) For simplicity the whole weight is considered as concentrated at the end of the mean radius. The mean velocity per second is

$$V = \frac{2 \times 3.1416 \times 5 \text{ ft.} \times 70 \text{ r.p.m.}}{60 \text{ sec.}} = 36.652 \text{ ft.}$$

The energy is

$$E = \frac{w v^2}{2g}$$

where

w = Weight;
v = Velocity;
$2g$ = Twice the acceleration of gravity = 64.4.

Then

$$E = \frac{7000 \times 36.652^2}{64.4} = 146{,}018.38 \text{ ft.-lb.}$$

(65) To save remeasuring all of the indicator cards, the horsepower can be corrected by the ratios of the lengths. It is known the mean effective pressure obtained, and therefore the horsepower, must be too large, so the correction is

$$\frac{Hp.}{3\frac{1}{2}} \times 3 \text{ or } \frac{435}{3.5} \times 3 = 372.857 \text{ i.hp.}$$

This is explained in the answer to the inquiry, "Using Averaging Planimeter" in the July 30 issue, page 167.

A new apparatus for separating oil from exhaust steam, described by the *Engineer*, of London, consists of a series of slightly curved plates standing vertically in a box through which the steam is passed. Each plate has four or five vertical baffles on the concave side. The plates are stacked close together, and their slight curvature, while sufficient to intercept oil and water in the steam, is not sufficient to cause much back pressure. An efficiency test is reported in which only $\frac{3}{10}$ of 1 per cent. of the original oil fed in passed through the separator to the condenser.

Derivation of the Massachusetts Formula for Areas of Circular Segments

By W. C. Marshall*

Some years ago, S. F. Jeter, then engineer for the Bigelow Co., sent the writer, through one of the students in the Sheffield Scientific School, a slip of paper on which was the Massachusetts formula for the area of a circular segment, viz.:

$$A = \tfrac{1}{3} H^2 \sqrt{\frac{2R}{H} - 0.608}$$

asking how it was obtained.

After some calculating the writer reached the conclusion that it was obtained from the rule for the area of a parabolic segment, which is

$$A = \tfrac{2}{3} H L$$

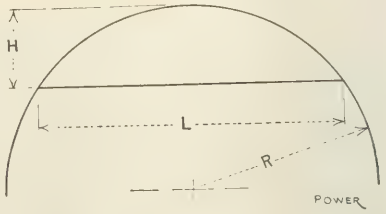


Fig. 1

where

L = Length of chord;
H = Height of segment;

as in Fig. 1.

The parabolic rule does not take into account the radius of the circle when it is used on a circular segment. Suppose it is reduced to an expression which brings in the radius. If L = one-half the chord length, then

$$A = \tfrac{4}{3} HL$$

The value of L may be obtained from the right triangle *abo*, Fig. 2, from

$$R^2 = L^2 + (R - H)^2$$

or

$$L^2 = R^2 - (R - H)^2$$

from which

$$L = H \sqrt{\frac{2R}{H} - 1}$$

Substituting this value of L in the expression

$$A = \tfrac{4}{3} HL$$

gives

$$A = \tfrac{4}{3} H^2 \sqrt{\frac{2R}{H} - 1}$$

Comparing this formula with the Massachusetts formula the only difference is found to be in the constant under the radical, which is 1 instead of 0.608. The constant 0.608 was probably obtained as follows:

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Take a semicircle, apply the parabolic rule for its area and

$$A = 1.33\ R^2$$

The area of a semicircle is

$$A = \frac{\pi\ R^2}{2}$$

or

$$A \times \frac{3.1416\ R^2}{2} = 1.57\ R^2$$

showing the parabolic rule to be in error by giving an area too small by the amount 1.33: 1.57, or if the areas are compared by the formulas

$$A = 1.33\ R^2 \quad \text{(Parabolic)} \qquad (1)$$

$$A = 1.57\ R^2 \quad \text{(Actual)} \qquad (2)$$

Equation (2) can be reduced to the *form* of (1) by dividing 1.57 by 1.33 and writing under the square-root sign the square of the quotient 1.178, or

$$\left[\frac{3.1416}{2} \div \frac{4}{3} = \frac{3.1416}{2} \times \frac{3}{4} = \frac{1.57+}{1.33+} = 1.178\right]$$

$$A = 1.33 \times 1.178\ R^2 = 1.33\ R^2 \sqrt{1.387} \qquad (3)$$

But the number under the radical in (1) is $\sqrt{2-1}$ while in (3) it is $\sqrt{2-0.61}$. Thus it is evident that the Massachusetts rule is the parabolic formula modified by changing the constant under the radical from 1 to 0.608 so as to give the true area for the largest possible segment, viz., a semicircle.

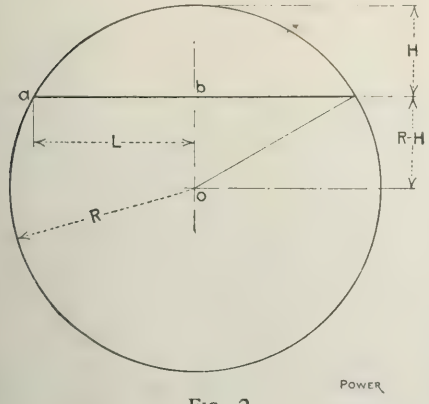


FIG. 2

The parabolic rule is less in error as the segment is smaller, for example: Take the area of the segment for a sector of ¼ the area of a circle. The area by the parabolic rule is 0.276, while the Massachusetts rule gives 0.285, which is correct. The error by the parabolic rule in this case is — 3 per cent. of the total area considered.

In the great Krupp plant in Germany, 89,430 hp. is the total steam power, and 2690 electric motors have an output of 50,491 hp. There are 68,500 employees.

It is predicted that concrete radiators for steam or hot-water heating will soon be made practical. They are a success in Germany, it is said.

The Manufacturers' Standard of Flanged Fittings

On other pages of this issue we have compared the two standard schedules for flanged fittings evolved by the engineering societies and the manufacturers. The differences between the two are clearly shown, and in this connection it may be interesting to learn why the manufacturers refused to conform with the societies' standard. Following is a statement sent out by the secretary of the manufacturers' committee.

"Without question, a universal standard is necessary and highly desirable; but any standard to be commercially successful must be designed to bring about the desired uniformity with the minimum inconvenience and expense, both to consumers and to manufacturers.

"Before the 1912 U. S. Standard was adopted there existed among the majority of manufacturers, in the United States, a generally accepted standard for flanged fittings and flanges. Most of the large flanged fittings manufacturers in this country had the entire pattern equipment necessary to build this line of goods and many had list prices and dimensions published for general distribution to the trade; in fact, these dimensions have been published in Kent's 'Pocket Book,' eighth edition.

"After having thoroughly investigated the conditions which induced the two societies to adopt the 1912 U. S. Standard, the manufacturers of flanged fittings and flanges throughout the country realized that certain changes in the generally accepted standard were desirable. The tendency of modern engineers to use higher steam-working pressures than was their practice some years ago has demonstrated the necessity for an increase in the strength of extra-heavy end flange bolting in the larger sizes. In drafting up the 1912 U. S. Standard, the American Society of Mechanical Engineers provided an increase in the bolting of large extra-heavy flanges; but this increase seems out of proportion with the necessities of steam-engineering practice. The manufacturers have, therefore, adopted a happy medium and have increased the bolting on these extra-heavy flanges, but have been more practical in this increase than the 1912 U S. Standard.

"So far as the strength of the fittings themselves, not considering the bolting, is concerned, the 1912 U. S. Standard has no advantage whatever over the Manufacturers' Standard.

"While the 1912 U. S. Standard provides longer center-to-face dimensions on many straight sizes, this change was not brought about by any attempt to better the fittings, but was an average of several published sets of dimensions which in many cases were out of date and did not correspond with the goods being made by the respective manufacturers.

"This change alone would require a large expenditure for patterns and shop equipment, without a commensurate advantage to the consumers.

"The increase in bolting in the Manufacturers' Standard compares very favorably with the table of dimensions on 'A Standard for High-Pressure Steam Lines,' issued by the Society of German Engineers in 1900; also, the report issued by the Engineering Standard Committee on British Standard Tables of Pipe Flanges.

"While the 1912 U. S. Standard provides much heavier bolting on the very large extra-heavy flanges, this excessive increase does not seem necessary, as the Manufacturers' Standard bolting is just as strong on the sizes, which include the great majority of pipe lines; namely, the 10-in. size and smaller.

"The Manufacturers' Standard makes no change in the center-to-face dimensions from the present standard, which avoids mistakes when the consumer makes changes or additions to his plant. It is also economical for the manufacturer, as he can use his present equipment, which, to a large extent, is made up in a very substantial fashion and cannot be changed without involving a high expenditure.

"Furthermore, the 1912 U. S. Standard would necessitate carrying two stocks of fittings for several years, or until the new standard is fully in effect. The Manufacturers' Standard will not cause this difficulty.

"The Manufacturers' Standard for reducing fittings is more in proportion with the straight sizes. The 1912 U. S. Standard has the same dimensions on the run of reducing fittings as the straight sizes, regardless of reduction. Consequently the proportion of a reducing fitting will not be in harmony with those of a straight fitting.

"It would seem apparent that the excessively large bolt diameters specified in the 1912 U. S. Standard for the larger flanges are much too cumbersome and inconvenient from the steamfitters' standpoint, while the bolt sizes specified in the Manufacturers' Standard are more in accordance with general practice. In a standard of this nature convenience as well as strength should be a governing factor.

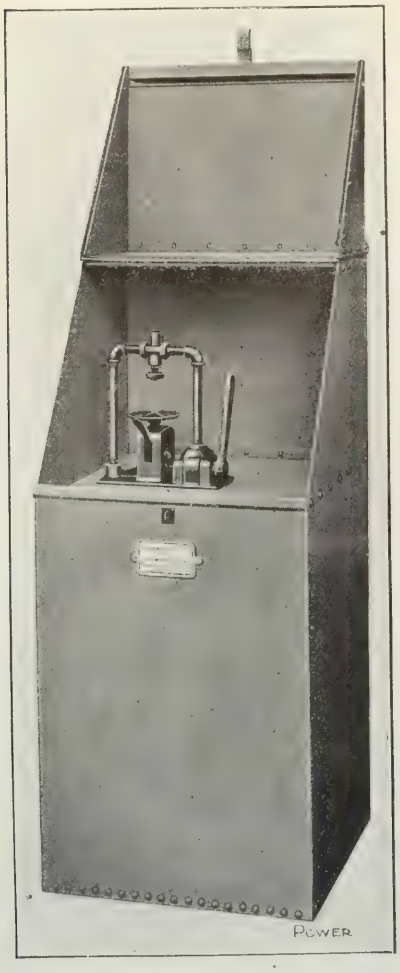
"While the manufacturers may be accused of not taking an interest in the 1912 U. S. Standard from its inception, their recent action in adopting a universal standard which is fully in accord with the requirements of the engineering trade, will demonstrate that they are acting with absolute sincerity and with a fair-minded attitude toward all interests involved."

In France a porous glass is being used for ventilation, having holes small enough to keep out drafts and dust.

Bowser Non-Overflow Oil Can Filler

S. F. Bowser & Co., Fort Wayne, Ind., are placing on the market an outfit designed for storing lubricating oil, and having a quick-acting pump which is used in filling hand oilers. The pump has an adjustable table operated with a lever. The hand oiler is placed on this table and the lever is pushed down, throwing the oiler into the proper position and fitting the nozzle into the opening of the hand oiler.

The height of the table can be adjusted instantly to accommodate it to oilers of different heights. When the oiler is in the filling position, the nozzle of the pump fits into the oil can, making an oil-tight connection. A few quick strokes of the pump fills the can. Any amount over the capacity of the oiler is returned to the storage tank through an overflow line, which prevents spilling the oil over the can.



A Non-overflowing Filler for Hand Oilers

The cabinet is provided with a metal hood. The outfit is fireproof and is made of sheet steel finished in bronze green.

During 1911, the exports of British coal amounted to 64,599,266 long tons.

The Two Schedules of Flanged Fittings

Two standards for flanges and flanged fittings are now before the public. The "U. S. 1912 Standard," evolved by the American Society of Mechanical Engineers and the National Association of Steam and Hot Water Fitters, was published in the Mar. 19, 1912, issue of Power. In the accompanying table the light figures reproduce this schedule and the heavy-face type opposite gives the variations found in the "Manufacturers 1912 Schedule," which was adopted July 10 and is to take effect Oct. 1.

For the standard fittings it may be noticed that the greatest variation occurs in the dimensions for the long turn elbow and for tees, crosses and laterals with reducing outlets.

For tees and crosses with reducing outlets from 9 in. up, the societies vary the center to face of outlet dimension from the standard only when the branch outlet is 8 in. in diameter and under. From 4½ in. to 8 in. in the laterals a variation is made for all reductions below 4 in., and from 9 in. up, a reduction to 8 in. or under calls for a different length from center to face of outlet. The face-to-face dimensions of the fittings are maintained, but not so in the manufacturers' schedule, which is given *under* the societies' figures in these cases in the heavy-face type.

No dimensions for laterals 1, 1¼, 1½, 26, 28 and 30 in. in diameter are given by the manufacturers, nor do they include any figures for Y-fittings, as these are regarded as special and are made to suit conditions. On the other hand, they give a set of dimensions for reducers, while the societies rule that the face-to-face dimension shall be equal to the diameter of the larger flange. The manufacturers' figures on reducers are given in the table. For the societies' tabulation on Y-fittings reference may be made to the Mar. 19 issue.

More variation occurs among the extra-heavy flanged fittings and particularly among the larger sizes and the fittings reducing on the branch. The same plan of tabulation was followed and the comparison should be interesting, as it shows at a glance the difference between the two standards.

Explanatory Notes

Some differences also occur in the explanatory notes accompanying the schedules. These will be evident from the following, in which the societies' notes appear in Roman type, and the manufacturers' variations, wherever they occur, follow in *italics*.

1. Standard or extra-heavy reducing elbows carry the same dimensions center to face as the regular elbows of the largest straight size.

2. Standard or extra-heavy tees, crosses and laterals, reducing on run, carry the same dimensions face to face as the largest straight size.

3. If flanged fittings for lower working pressures than 125 lb. are made, they shall conform in all dimensions, except in thickness of shell, to this standard, and shall have the guaranteed working pressure cast on each fitting. Flanges for these fittings must be of standard dimensions.

4. Where long-turn fittings are specified, it has reference only to elbows, which are made in two center to face dimensions, to be known as "elbows" and "long-turn" elbows, the latter being used only when so specified.

5. All standard-weight fittings must be guaranteed for 125 lb. and extra-heavy fittings for 250 lb. working pressure, and each fitting must have some mark cast on it indicating the maker and the guaranteed working steam pressure.

6. All extra-heavy fittings and flanges to have a raised surface $\frac{1}{16}$ in. high inside of the bolt holes for the gasket.

Standard-weight fittings and flanges to be plain faced.

Bolts to be ⅛ in. smaller in diameter than the bolt holes.

Bolt holes should straddle the center lines.

7. Size of all fittings scheduled indicates the inside diameter of the ports. For the outside diameter of pipe use the corresponding size of the inside diameter fittings.

7. *Size of all fittings scheduled indicates inside diameter of ports except for extra-heavy fittings* 14 *in. and larger when the port diameter is ¾ in. smaller than nominal size.*

8. The face-to-face dimension of a reducer, either straight or eccentric, shall be equal to the diameter of the larger flange.

8. *The face-to-face dimension of reducers, either straight or eccentric, for all pressures, shall be the same face to face as given in the table of dimensions.*

9. Square head bolts with hexagonal nuts are recommended.

10. Twin ells, double branch ells, side outlet ells, side outlet tees and four-way tees, whether straight sizes or reducing, carry the same dimensions center to face and face to face as the regular ells and tees.

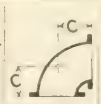
10. *Twin elbows, whether straight or reducing, carry same dimensions, center to face and face to face, as regular straight size ells and tees.*

Side outlet elbows and side outlet tees, whether straight or reducing sizes, carry same dimensions center to face and face to face as regular tee having the same reductions.

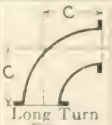
11. Bull-head tees or tees increasing on outlet will have the same center to face and face to face dimensions as a straight fitting of the size of the outlet.

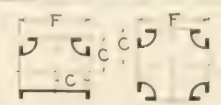
SCHEDULE OF FLANGES AND FLANGED FITTINGS

Standard Weight Flanged Fittings


90° Elbow

45° Elbow


Long Turn Elbow


Tees and Crosses

Laterals and Y Branches

Reducer

Size, Inches	All Fittings and Flanges: Diameter of Flange	Thickness of Flange	Number of Bolts	Diameter of Bolt Circle	Diameter of Bolt Holes	Size of Bolts	90° Elbow: Center to Face "C"	45° Elbow: Center to Face "C"	Long Turn Elbow: Center to Face "C"	Tees and Crosses: Center to Face "C"	Center to Face Outlet 8" and Under	Laterals and Y Branches: Face to Face "F"	Center to Face on Short End "C"	Center to Face of Outlets 8" and Under	Face to Face "F"	Reducer: Face to Face "F"
1	4	7/16	4	3	9/16	7/16	3½	2 1½	5½ 5	3½		7	6	2½	— 8	
1¼	4¼	½	4	3⅜	9/16	7/16	3¼	2	6 5½	3¼		7½	6½	2½	— 8½	
1½	5	9/16	4	3⅞	9/16	½	4	2½	6½ 6	4		8	6½	2½	— 9½	
2	6	⅝	4	4¾	¾	⅝	4½	2½	7 6½	4½		9	8	2½	— 10½	
2½	7	11/16	4	5½	¾	⅝	5	2¾ 3	7½ 7	5		10	9½	2½	— 12	
3	7½	¾	4	6	¾	⅝	5½	3	8 7½	5½		11	10	3	13	6
3½	8½	13/16	4	7	¾	⅝	6	3½	9 8½	6		12	11½	3	14½	6½
4	9	15/16	8	7½	¾	⅝	6½	4	10 9	6½		13	12	3	15	7
							Outlet 2½ in. and under							2	11 13	
4½	9¼	15/16	8	7¾	⅞	¾	7	4¼ 4	11 9½	7		14	12½	3	11½* 16½	7½
							Outlet 2½ in. and under							2	11 13	
5	10	15/16	8	8½	⅞	¾	7½	4½	12 10½	7½		15	13½	3½	12½* 17	8
							Outlet 3 in. and under							2	12 14	
6	11	1	8	9½	⅞	¾	8	5	13 11½	8		16	14½	3½	13½* 18	9
							Outlet 3 in. and under							1½	13½ 15	
7	12½	1 1/16	8	10¾	⅞	¾	8½	5½	14½ 12½	½		17	16½	4	13½* 20½	10
							Outlet 3½ in. and under							1½	15 16	
8	13½	1⅛	8	11¾	⅞	¾	9	6 5½	16 14	9		18	17½	4½	14½* 22	11
							Outlet 4 in. and under							1½	15 16	
9	15	1⅛	12	13¼	⅞	¾	10	6½ 6	18 15¼	10	9	20	19½	4½	17½* 24	11½
							Outlet 4½ in. and under							1½	16 17	
10	16	1 3/16	12	14¼	1	⅞	11	7 6½	20 16½	11	10	22	20½	5	18½† 25½	12
							Outlet 5 in. and under							1	18 18	
							Outlet 6 in. and under				9½	18				
12	19	1¼	12	17	1	⅞	12	7½ 7	22 19	12	11½	24	24½	5½	20† 30	14
							Outlet 6 in. and under							1	20½ 20	
							Outlet 8 in. and under				11	20				
14	21	1⅜	14	18¾	1⅛	1	14	7½ 7	24 21½	14	12½	28	27	6	21½† 33	16
							Outlet 7 in. and under							1	23 22	
							Outlet 9 in. and under				13	22				
15	22¼	1⅜	16	20	1⅛	1	14½	8 7½	26 22¼	14½	13	29	28½	6	22¼† 34½	17
							Outlet 7 in. and under							1	24 23	
							Outlet 9 in. and under				13½	23				
16	23½	1 7/16	16	21¼	1⅛	1	15	8	28 24	15	13½	30	30	6½	22½† 36½	18
							Outlet 8 in. and under							1	25½ 24	
							Outlet 10 in. and under				14	24				
18	25	1 9/16	16	22¾	1¼	1⅛	16½	8½	30 26½	16½	14½	33	32	7	24½† 39	19
							Outlet 9 in. and under							1	27½ 26	
							Outlet 12 in. and under				15½	26				
20	27½	1 11/16	20	25	1¼	1⅛	18	9½	32 29	18	15½	36	35	8	25½† 43	20
							Outlet 10 in. and under							1	29½ 28	
							Outlet 14 in. and under				17	28				
22	29½	1 13/16	20	27¼	1¼	1⅛	20	10	34 31½	20	16½	40	37½	8½	27½† 46	22
							Outlet 10 in. and under							½	31½ 29	
							Outlet 15 in. and under				18	28				
24	32	1⅞	20	29½	1¼	1⅛	22	11	36 34	22	17½	44	40½	9	28½† 49½	24
							Outlet 12 in. and under							½	34½ 32	
							Outlet 16 in. and under				19	30				
26	34¼	2	24	31¾	1⅜	1¼	23	13	39 36½	23	19½	46	43½	9½	30¾† 53	26
							Outlet 18 in. and under				20	32				
28	36½	2 1/16	28	34	1⅜	1½	24	14	42 39	24	20½	48	46½	10½	33† 57	28
							Outlet 18 in. and under				21	32				
30	38¾	2⅛	28	36	1½	1⅜	25	15	45 42	25	21½	50	49½	11	35½† 60½	30
							Outlet 20 in. and under				23	36				

Extra Heavy Flanged Fittings

1	4½	11/16	4	3½	5/8	½	4½ 4	2½ 2	5½ 5	4½ 4		8½ 8	7½	2½	—	9½		
1¼	5	¾	4	3⅝	5/8	½	4½	2½	6 5½	4½		8½	7½	2½	—	9½		
1½	6	13/16	4	4½	¾	⅝	4½	2½	6½ 6	4½		9	8½	2½	—	11		
2	6½	⅞	4	5	¾	⅝	5	3	7 6½	5		10	9½ 9	2½	—	12	11½	
2½	7½	1	4	5⅞	⅞	¾	5½	3¼ 3½	7½ 7	5½		11	10½	2½	—	13		
3	8¼	1⅛	8	6⅝	⅞	¾	6	3½	8 7½	6		12	11½ 11	3	—	14½	14	6
3½	9	1 3/16	8	7¼	⅞	¾	6½	4	9 8½	6½		13	12½	3	—	15½		6½
4	10	1¼	8	7⅞	⅞	¾	7	4½ 4½	10 9	7		14	13½	3	—	16½		7
							Outlet 2½ in. and under							2	13	15		
4½	10½	1 5/16	8	8½	⅞	¾	7½	4½	11 9½	7½		15	14½	3½	13½*	18		7½
							Outlet 2½ in. and under							2½	13½	15		
5	11	1⅜	8	9¼	⅞	¾	8	5	12 10¼	8		16	16 15	3½	14*	19½	18½	8
							Outlet 3 in. and under							2½	14½	16		
6	12½	1 7/16	12	10⅜	⅞	⅝	9 8½	5½	13 11½	9 8½		18 17	17½	4	14½*	21½		9
							Outlet 3 in. and under							2½	15½	17		
7	14	1½	12	11⅞	1	⅞	9½ 9	5½ 6	14½ 12¾	9½ 9		19 18	19 18½	4½	15½*	23½ 23		10
							Outlet 3½ in. and under							2½	16½	18		
8	15	1⅝	12	13	1	⅞	10½ 10	6	16 14	10½ 10		21 20	20½ 20	5	*	25½ 25		11
							Outlet 4 in. and under							2½	18½	20		
9	16¾	16¼	1⅝	12	14	1⅛	1	11 10½	6½	18 15¼	11 10½	10½	22 21	22¼ 21	5	20½†	27½ 26	11½
							Outlet 4½ in. and under							2½	19½	21		
10	18¼	17½	1⅞	16	15¾	15¼	1⅛	1	12 11½	7	20 16½	12 11½	11	24 23	25¾ 23	5½	21½†	30½ 28½ 12
							Outlet 5 in. and under							2½	21½	23		
							Outlet 6 in. and under				11	18						
12	20¾	20	2	16	17¾		1¼ 1⅛ 1⅛	1	13½ 13	8	22 19	13½ 13	12	27 26	27¼ 26½	6	23†	35½ 32½ 14
							Outlet 6 in. and under							2½	24½	26		
							Outlet 8 in. and under				12½	21						
14	23½	22½	2⅛	20	20¼	20	1⅜ 1⅛ 1¼	1	15 14½	9½ 8	24 21½	15 14½	13	30 29	31¼ 30	6½	24½†	37½ 36½ 16
							Outlet 7 in. and under							2½	27½	29		
							Outlet 9 in. and under				13½	22						
15	25	23½	2 3/16	20	21½	21	1⅜ 1¼ 1¼	1⅛	15½ 15	10 8½	26 22¼	15½ 15	13½	31 30	33¼ 31½	6½	25½†	39½ 38 17
							Outlet 7 in. and under							2½	28½	30		
							Outlet 9 in. and under				13½	23						
16	26	25	2¼	20	22½		1½ 1¼ 1⅜	1⅛	16½ 16	10½ 9	28 24	16½ 16	14	33 32	34½ 33	7½	25½†	42 40½ 18
							Outlet 8 in. and under							3	30½	32		
							Outlet 10 in. and under				15	24						
18	28½	27	2⅜	24	24¾	24½	1½ 1¼ 1⅜	1⅛	17½ 17	11 9½	30 26½	17¾ 17	15	35½ 34	37½ 36	8	27½†	45½ 44 19
							Outlet 9 in. and under							3	32½	34		
							Outlet 12 in. and under				16½	27						
20	31	29½	2½	24	27	26¾	1⅝ 1⅜ 1½	1⅛	19¼ 18½	11½ 10	32 29	19¼ 18½	16	38½ 37	40½ 38	8½	28½†	49½ 46½ 20
							Outlet 10 in. and under							3	36	37		
							Outlet 14 in. and under				17½	30						
22	33	31½	2⅝	28	29¼	28¾	1⅝ 1⅜ 1½	1¼	20½ 20	12 10½	34 31½	20½ 20	17	41 40	43½ 41½	9½	30†	53½ 51 22
							Outlet 10 in. and under							3	39	40		
							Outlet 15 in. and under				18½	30						
24	36	34¼	2¾	28	32	31½	1⅞ 1⅜ 1⅝	1⅜	22½ 22	13 11½	36 34	22½ 22	18	45 44	47½ 44½	10	31½†	57½ 54½ 24
							Outlet 12 in. and under							3	43	44		
							Outlet 15 in. and under				19½	30						

12. Up to and including the 4-in. size, center to face and face to face dimensions of reducing fittings will be the same as that of a straight fitting of the larger opening.

12. *Tees and crosses 9 in. and down, reducing on the outlet or run and outlet, use the same dimensions as straight sizes of the larger port.*

Sizes 10 in. and up, reducing on the outlet, are made in two lengths, depending on the size of the outlet as given in the table of dimensions.

If the outlet is larger than that given in table, use dimensions of straight sizes.

Laterals 3½ in. and down, reducing on the branch, use the same dimensions as straight sizes of the larger port.

Sizes 4 in. and up, reducing on the branch, are made in two lengths, depending on the size of the branch as given in the table of dimensions.

If the outlet is larger than that given in the table, use dimensions of the straight sizes.

Y's are special and are made to suit conditions.

Double sweep tees are not made reducing on the run.

Steel flanges, fittings and valves are recommended for superheated steam.

13. Pipe sizes 14 in. and over refer to outside diameter.

Some New Condensing Apparatus

Some of the more notable products in the line of condensing machinery built by the C. H. Wheeler Manufacturing Co., as seen on a recent visit to the works at North Philadelphia, Pa.

The writer recently paid a very interesting visit to the works of the C. H. Wheeler Manufacturing Co., of Philadelphia, Penn., and saw many features in the design of condensing machinery which will be of interest to the readers of POWER.

The plant is located at North Philadelphia and occupies the site of the factory established by William Barr in the early nineties, and where many record-breaking pumping engines were built by him.

Control of this plant was taken over by C. H. Wheeler (whose name is so intimately associated with condensing machinery) about eight years ago and the works remodeled to engage in the manufacture of condensing apparatus, particularly the high efficiency designs necessary for turbine practice.

Fig. 1 shows an arrangement or tube layout for a turbine surface condenser, designed for the exhaust steam to enter on one side and to operate on the wet

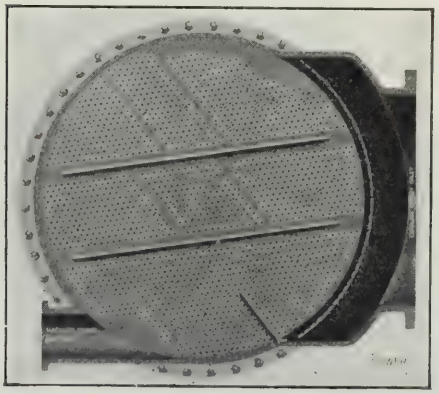


FIG. 1. DRAIN-CIRCULATION PLATES AND TUBE ARRANGEMENT IN A TURBINE CONDENSER

system, one pump removing the condensed steam, air and vapor. The drain circulation plates, shown together with the entering steam lanes provided by the omission of tubes at certain points, it is claimed, keep the entire tube surface active and enable the condensed steam to be removed at a temperature closely approximating that of the vacuum. This removes a serious objection many engineers have to a combined system of this type, viz., the chilling of the feed water, which was generally considered unavoidable in the operation of a strictly wet system.

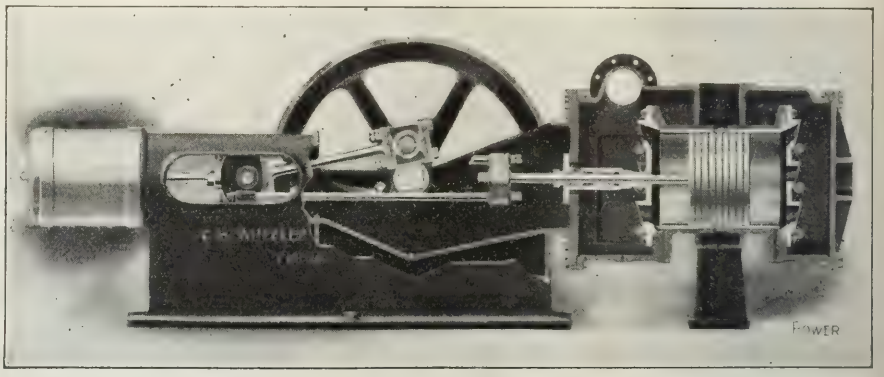


FIG. 2. BROKEN VIEW OF THE MULLAN SUCTION VALVELESS VACUUM PUMP

Inquiring as to the design of vacuum pump used in connection with such a condenser, the writer was shown a number of the Mullan pumps. Fig. 2 reveals the construction of the vacuum cylinder of these machines. The design is attractive in its extreme simplicity; the pump is of the double-acting horizontal type and all the working parts are bronze composition to withstand the action of the distilled water handled and avoid corrosion and deterioration, if shut down for any length of time. The pump is suction valveless, the place of these valves being taken by a series of ports cut in the cylinder liner, and the piston in its travel to and fro, alternately connects the two ends of the cylinder with the condenser, then discharges this vapor and water on the return stroke through the valves on the vertical valve decks at each end of the cylinder. This pump has previously been described in POWER, but the recent improvements noted were the increased port areas and the new type of coil flexible metallic valves, enabling the pump to operate at a higher speed than the earlier designs and increasing its volumetric and mechanical efficiencies.

For circulating service a regular centrifugal pump is used and a number of these machines were seen under construction, some engine, some motor and some steam-turbine driven. Fig. 3 is a view, partly in section, of such a pump. It is of the double-suction volute design

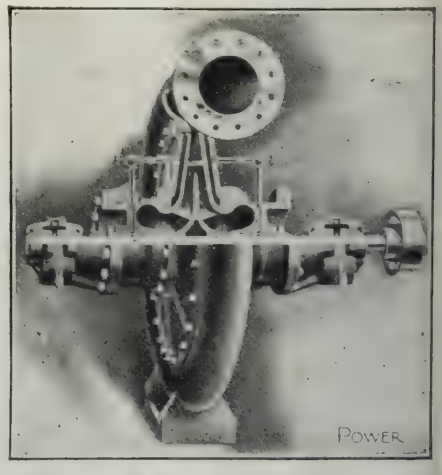


FIG. 3. CENTRIFUGAL CIRCULATING PUMP PARTLY IN SECTION

with outboard independent ring-oiling bearings.

One of the most interesting develop-

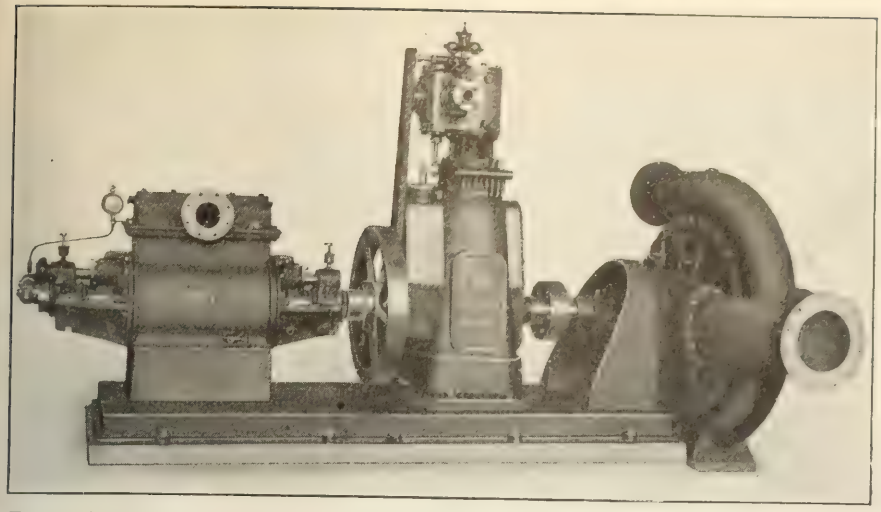


FIG. 4. A COMBINATION UNIT CONSISTING OF A VACUUM AND A CIRCULATING PUMP DRIVEN BY THE SAME ENGINE

ments this company has produced however, is what is termed the Rotrex combination system, this consisting of the Rotrex vacuum pump (which has previously been illustrated and described in POWER), combined with a centrifugal circulating pump, both directly connected to one vertical or horizontal engine. Such a combination with a vertical engine is shown in Fig. 4. For units of medium size this makes a simple and desirable outfit. On the larger units, say above 2000 kw., the writer was told that it is more usual to operate the pumps separately.

Fig. 5 shows a complete installation of steam turbo-generator with surface condenser and Rotrex combination air and circulating pumps, engine driven. In this case the condenser is located directly under the turbine. The exhaust connection includes a copper expansion joint and a special type of gate valve made by the company, which has a side outlet for connecting the atmospheric relief valve.

This gate valve, shown in Fig. 6, consists of a regular gate valve with a wedge device for seating. The unusual feature is the provision of an outlet nozzle cast on the bottom of the valve body. In normal operation, with the gate open, steam passes directly from the turbine to the condenser, and, should the vacuum be lost, the automatic relief valve opens and the steam passes to atmosphere. If the condenser requires attention, necessitating shutting it down while the turbine must be operated (as is sometimes the case where there is an insufficiency of standby equipment), the gate valve can be closed, cutting out the condenser, but still permitting the steam to pass to the atmosphere through the automatic relief valve. This arrangement dispenses with the usual tee in the exhaust line which, with the large size of turbine exhaust connections, takes up considerable space, is very expensive and introduces another large bolted joint with its potential possibilities of air leakage. Provision is made for the water sealing of the stuffing boxes of the gate valve and the water sealing of the atmospheric relief valve, so as to avoid possibility of leakage at these points.

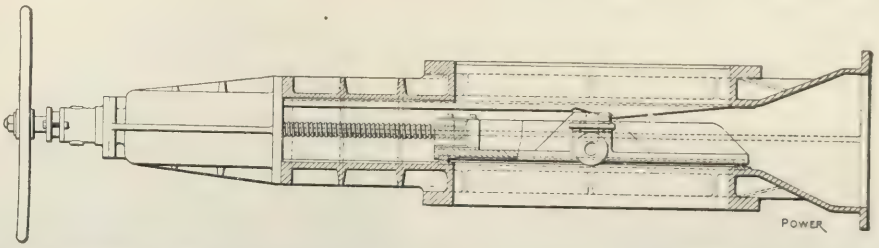


FIG. 6. THE C. H. WHEELER GATE VALVE WITH WEDGE SEATING DEVICE

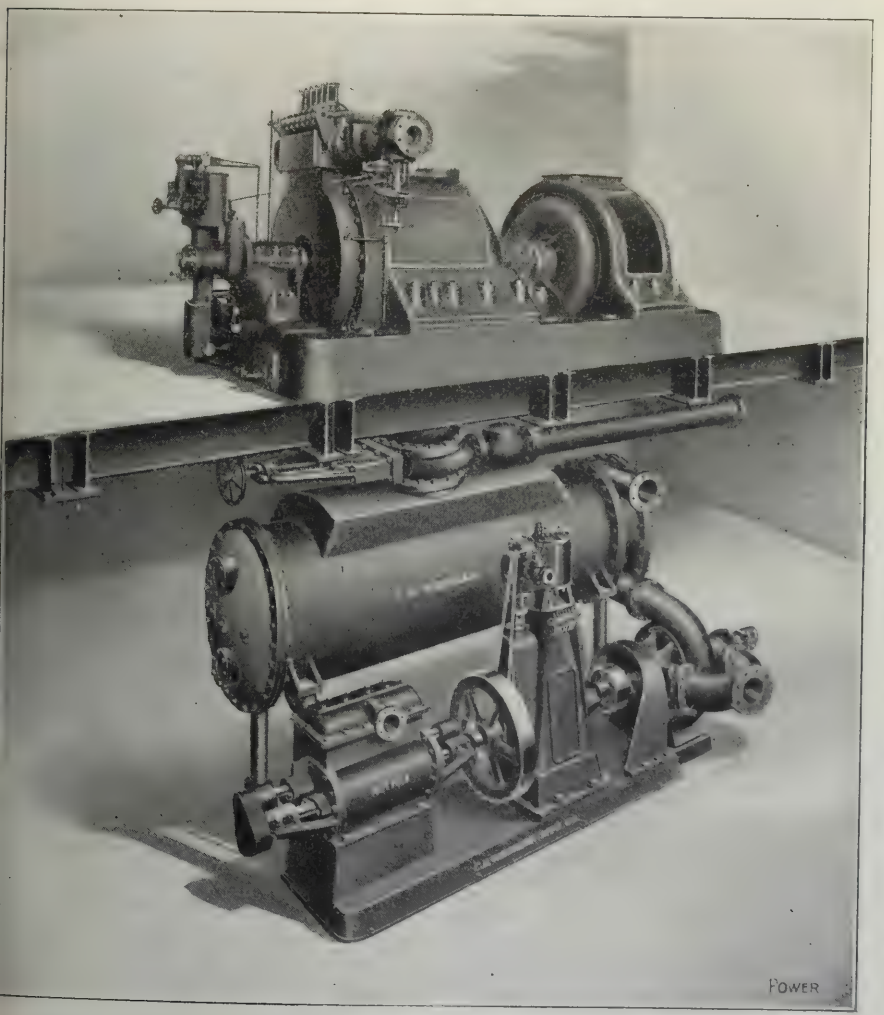


FIG. 5. SURFACE CONDENSER WITH ROTREX VACUUM PUMP AND CENTRIFUGAL CIRCULATING PUMP OPERATING WITH STEAM TURBO-GENERATOR

SCHOOL NOTES

Two new scholarships in agricultural engineering, the first of their kind in any institution, have just been established at the Iowa State College. One was founded by the M. Rumley Co., La Porte, Ind., and is known as the Meinrad Rumley scholarship in honor of that pioneer manufacturer of agricultural implements. The

other was established by the International Harvester Co., of Chicago. Both are worth $250 a year to the students and will be awarded annually to young men qualified to do special research work along the lines of agricultural engineering. Prof. J. B. Davidson is head of the new department.

The Massachusetts Institute of Technology opened its summer surveying camp at Gardner Lake, East Machias, Maine, Aug. 7.

The camp forms part of the summer school maintained by the institute, and students in certain courses are required to attend the camp. Nearly 350 is the total registration this year, an advance over average figures even without taking into account the 70 who will be in the camp. An anonymous graduate bought the camp site and was planning for the buildings when another alumnus, C. W. Eaton, of Haverhill, offered $10,000 for buildings and equipment. The total plant has been established at a cost of about $50,000. There are three permanent buildings, living room, dining room and drafting room, about which are arranged 40 tents. There is an equipment of boats, motor boat and barge, and provision for bathing. Seventy students and twelve of the teachers, including practically the whole instructing force of the department with Prof. C. M. Spofford, will remain till the middle of September. The location of the camp gives unusual advantages for all the varieties of surveying.

OBITUARY

James J. Waters

James J. Waters, of Brooklyn, N. Y., a well known marine engineer, died on Aug. 1, from complications and shock resulting from an automobile accident.

Mr. Waters was born in Jersey City, N. J., July 8, 1870. After serving his apprenticeship as a machinist he went to sea as engineer for the Morgan and Clyde Lines.

He was on watch in the engine room of the Old Dominion S.S. "Wyanoke" on the night of Apr. 29, 1896, when she collided with the U. S. S. "Columbia" in Hampton Roads, and was the only man in the engineers' department on watch at the time who escaped with his life. He was, however, so severely scalded that he was incapacitated for fourteen months.

After his recovery he served on harbor boats of the Old Dominion Line until 1903, when he was elected secretary of the Marine Engineers' Benevolent Association No. 33. He afterward became business manager, corresponding and financial secretary, serving successfully until his death. He was elected representative to the thirty-first annual convention in 1906 and has been reëlected annually ever since. In 1907 he was elected secretary and treasurer of the Board of Trustees of the *American Marine Engineer*. In January, 1911, he was elected chairman of the board and in January, 1912, was reëlected for a term of five years.

In the various association positions in which he has served he rendered exceptional service. He took a deep personal as well as official interest in all association matters and was well known and much loved in engineering and other circles.

Mr. Waters was the oldest son of John Waters, chief boiler maker for the Clyde Lines. He is survived by his widow, two children, father, mother, four sisters and a brother.

James J. Waters

He was an enthusiastic member of Jersey City Lodge No. 211, B. P. O. E., and of Fairview Council No. 498, Royal Arcanum.

William Boissy

William Boissy, engineer of the power plant of the Woonsocket Rubber Co., Woonsocket, R. I., died recently at his home in that city. He had been seriously ill for some time.

Harold H. Fletcher

Harold H. Fletcher, for many years a mechanical engineer for the Boston & Albany R.R., died at his home, 5 Winchester St., Brookline, Mass., on July 25. Mr. Fletcher had been employed by several large railroad interests previous to his connection with the Boston & Albany. He was a graduate of the Massachusetts Institute of Technology.

Ralph P. Gillette

The death is announced of Ralph P. Gillette, manager of mechanical sales for the Minneapolis Steel & Machinery Co., Minneapolis, Minn., on July 12. Mr. Gillette had sought shelter under a tree during a severe storm and was instantly killed by lightning.

Leland B. Smith

Leland B. Smith, once chief engineer of the Yazoo Light & Power Co., Yazoo City, Miss., and later with the Georgia Power Co., Atlanta, Ga., recently met his death by drowning.

Mr. Smith was about 25 years of age, and his rapid advancement in the engineering field gave promise of his becoming one of the ablest of the young engineers in the South. He was instrumental in organizing Col. Goethals Branch No. 1, I. O. E., and was its first secretary.

PERSONAL

Bartholomew Scannel, of Lowell, Mass., who has represented the boiler-manufacturing interests of the Massachusetts State Board of Boiler Rules for several years, has declined a reappointment to that office. Gov. Eugene N. Foss has named Henry H. Lynch, of Brookline, in his stead.

Lieut. Walter G. Diman, son of George H. Diman, of the American Woolen Co., at Lawrence, Mass., has been appointed superintendent of motive power for the Amoskeag Mills, Manchester, N. H. Capt. Charles H. Manning, who has so long been identified with this plant, will retain his connection with the corporation.

Alfred Still was recently appointed chief electrical engineer of the Algoma Steel Corporation, with headquarters at Magpie Mine, Ont. He is a member of the British Institution of Electrical Engineers, the Institution of Civil Engineers and the American Institute of Electrical Engineers, and is the author of standard electrical books and a contributor to technical journals.

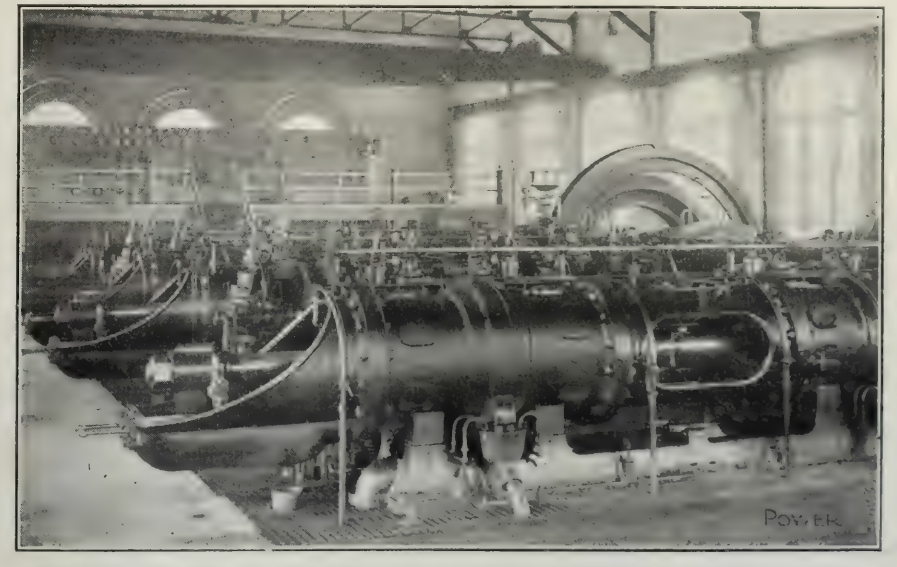
President C. A. Griscom, Jr., and Vice-President Arvine Wales are the guiding officials of the Griscom-Russell Co., which has taken over the business of the Griscom-Spencer Co. and the Russell Engine Co., of Massillon, Ohio. The new company will assume all the obligations of both companies, and carry on all the business heretofore conducted by them. So far as is necessary, pending business will be completed by the former companies. The management and staff remain unchanged. The general offices of the Griscom-Russell Co. will be at 90 West St., New York City.

POWER

Vol. 36 NEW YORK, AUGUST 20, 1912 No. 8

WHEN gas engines first entered the power field it was assumed that good men could be obtained to operate them for about $1.25 per day, that is, start and stop them and keep them lubricated during the working period.

If anything went wrong a machinist was called in to make repairs, no matter whether he understood gas engines or not. He received 60 cents an hour for his work, and often worked two or three days trying to make an engine run, when the only trouble was that the exhaust ports were almost closed with acarbon deposit.



Steam engineers are not gas engineers. They are in separate fields and both must be trained in their respective callings. When a steam engine goes wrong the cause is usually apparent to a steam engineer; when a gas engine refuses to operate, it requires a man well versed in its operation to overcome the difficulty without undue loss of time.

Engineers who can successfully handle gas engines and may be looking for a larger opportunity than seems to be theirs, so far, may well turn their attention to the oil region of Kansas, Oklahoma, Texas and other oil producing states where the gas engine is largely used for field work. Competent men to take charge of repairs are not plentiful and a capable man would find renumerative employment superintending such work.

The old idea that anybody can run a gas engine plant still obtains in the minds of some owners and managers. The engines are run with cheap help until they demand attention when a competent engineer is engaged to put the plant in good running order. Even then the old method of operation is sometimes resumed, an excuse being given for letting the high-grade engineer go.

But this class of owners is disappearing and more and more are operators of real ability being sought. Not cheap help, but economical operation is the motto. Why not be a gas engineer and get in line to accept some of these positions when they appear?

Several engineers have become specialists in superintending the repairing of gas engines, but they studied hard and obtained practical experience before they were competent to undertake the work. There is room for others in this same field and the necessary knowledge can be gained in the same way.

Do not confuse the modern gas engine with the wood-sawing outfit. There are gas power plants in operation that can put thousands of steam plants in the shade for size, and they require good men to operate them. Capable men who understand their business should have no trouble in making good in the gas engine field.

Running a Steam Turbine

By F. R. Brosius

This, in the opinion of the judges, is the second best of the 46 articles submitted in the contest for the 50-Dollar prize. A symposium of the subject made up of the salient points of the other contributions will appear in another issue.

The same general principles apply to operating turbines as to engines, the turbine, however, is the more easily handled. There are numerous bothersome things to contend with in the turbine, but its operation is very much simpler than that of the engine. Doubtless, a large proportion of the earlier troubles were such as are common to all radical departures from existing practice, but these were gradually overcome as undesirable features were eliminated and experience in operation was gained. How far this was true depended upon the engineer himself. It was his task to master the intricacies and finer points of operation.

Starting a turbine, preparatory to cutting it in on the line, is purely mechanical and routine work. Consequently, the following deals more with those things which would tend to interrupt a turbine's service, and to find the remedies to be applied. The writer's experience has been altogether with high-pressure turbines, and since most turbines in use are of this type, and the repair and operation of the low-pressure type is so like that of the high pressure, only the operation of the latter will be described.

Placing the Turbine in Service

1. Turn steam, or water, as the case may be, into the packing seals. This will prevent leakage of air which would tend to lower the vacuum.

2. Start the air pump—get it up to speed.

3. Start the step-bearing pumps and get the pressure up to normal.

4. After the air pump has run long enough to free the circulating pump and its suction pipe of air, and the condenser is primed, start the circulating pump and get it up to speed.

5. Start the oil pumps and have oil circulating freely to all bearings. Oil all parts that have to be oiled by hand. See that the oil pressure is up to that required for your particular installation, as different sized turbines require different pressures on the step, while, with the horizontal turbine, the oil pressure on the relays for operating the governor is different from that required for the bearings.

6. Start the turbine turning around slowly until thoroughly warmed up. When up to speed, note particularly if the governor controls the turbine; if so, open the throttle.

7. The turbine is now ready to be cut in on the line.

Note: In case the dry-vacuum system is used, start the hot-well pump before the air pump, to remove any condensation which may be in the condenser.

Taking the Turbine Out of Service

1. Gradually remove the load. See if the governor controls the turbine; then shut down by using the emergency trip, to keep the trip in perfect working order.

2. Shut off the steam (or water) from the packing seals.

3. Shut down the condenser pumps.

4. Shut down the oil pumps after turbine has stopped.

Note: In the case of a vertical turbine, do not close down the step pumps until the turbine has stopped.

Only within the last ten years has the steam turbine come into prominence as a prime mover. It has proved its great economy in floor space, steam consumption, first cost, repair costs, and attendance, in the great modern power plants. Now it is demonstrating its value as a prime mover in the very small plant. Therefore, it behooves the engineer to know something about the turbine.

It is only a question of time until the engineer in the present reciprocating-engine plant will be called upon to advise his employers as to the saving or advantage to be effected by installing turbines. He must be prepared to furnish the figures, and still more important, he must have the requisite knowledge to manage and operate the turbine, if he expects to hold his job. Not only for this reason must he keep up with the times, but also as a matter of pride in his profession.

Books without number have been published setting forth the principles of the turbine, describing its mechanical construction in every detail, the why and wherefore of the adoption of various devices tending to simplify and make safer its operation, going into great detail concerning differences in type, giving efficiencies and preferences, etc. In fact, the engineer ought never to be at a loss for proof of any theory about, or principle of, the steam turbine. All this information is easily procured and is a matter simply of reading, while the actual operating troubles are a matter of experience. Remedies must be applied, and those sometimes at a moment's notice. Provision must be made against their repetition. Each emergency becomes a problem to be mastered.

The turbine is yet far from perfect, and will never be so nearly perfect that it will not need attention and adjustment to operate at the highest efficiency. Information regarding turbine troubles has been hard to get. The manufacturers are familiar with turbine failings, but do not like to admit that their machines are not capable of perfect operation at all times. In view of this, we believe that the engineer who makes the greatest success in operating the turbine will be the one who knows the troubles likely to be met, and who knows how to overcome them.

Causes and Seats of Turbine Troubles

Vertical Type

1. Step bearing—injury to buckets.
2. Top-middle-bottom bearings — injury to buckets.
3. Valves—mechanically and hydraulically operated.
4. Valves—electrically operated.
5. Carbon packing rings.

Horizontal Type

6. Water seals.
7. Loss of blading.

Common to Both Types

8. Lubrication.
9. Rapid starting.
10. Cooling the generator.
11. Cleaning the generator.
12. Water in the steam.
13. Magnetic balance.
14. Assembling after repairs.
15. Poor mechanical adjustment.

Prevention and Cure for Turbine Troubles

1. *Step Bearing*—The pump should be kept in very best repair, as a loss of pressure means injury to buckets on both the rotor and intermediates.

The gage indicating the pressure carried must be kept correct.

If water is used on the step, it must be kept free from sediment.

Many turbine installations include a weighted accumulator to keep the step up in case the water or oil pressure is lost. A cheaper way, and one requiring less room but needing more attention, would be to install an emergency steam pump with a regulator in the line to the steps, set at about 5 lb. lower than the pressure regularly carried. If the pressure is dropped by the other pumps, the emergency pump at once picks it up. The steam end of the pump must be kept drained and the pump should be tried at intervals to see that it is working properly, by dropping the pressure on the regular pumps.

A baffle screw in the line to the steps regulates the variations in pressure due

to the pump pulsations. This must be examined frequently and kept clean. Great care should be taken in packing the step-bearing pumps. The rods and glands should be kept in good condition to prevent packing from getting into the water, which will stop up the baffle screw and prevent pressure from reaching the step. The writer has found it well to use a ring of hydraulic packing in the back end of the stuffing-box and to use a good grade of lubricated flax packing. He has tried metallic packings and found them unsatisfactory.

2. *Bearings*—The top, middle and bottom bearings must be kept in perfect condition and alignment. If not kept perfectly aligned, even if they do not heat, they will wear much faster, and when they become loose, the machine is more likely to vibrate.

If water is used on the step, care should be taken to clean the bottom bearing which is lubricated by the water flowing up around it from the step. If the grooves in this bearing become stopped up, or the flow of water is retarded, the pressure will become excessive, raise the rotor and damage the tops of the buckets on the rotor and the bottoms of the intermediates.

The top and middle bearings must not be flooded with oil as are the bearings in the reciprocating engine. There has been considerable trouble with these bearings throwing oil into the generator, but this can be overcome by fitting felt collars tightly around the shaft above the middle and below the top bearings. These should be watched carefully and when the felt becomes worn it should be renewed. If oil is permitted to soak into the windings, it will in time soften and break down the insulation, which usually means rewinding the entire armature.

3. *Valves—Mechanically and Hydraulically Operated*—These valves must be kept properly adjusted, and must not be allowed to get loose to make possible the breaking of any parts. They should also be kept thoroughly lubricated.

All dogs and moving parts in any way controlled by the governor should be close fitting but yet perfectly free, if the machine is to be well governed. Attention given these parts is similar to that required by the valve-gear on a Corliss engine.

4. *Valves — Electrically Operated* — The needle valves must be watched constantly and kept tight, or the leaking will cause the controlling valve to refuse to either open or close.

The baffle-plates may be broken and become wedged around the controlling valve, causing it to stick.

The solenoid spools may become charred and short-circuited, whereupon they could not control the needle valves, and then the controlling valves would not operate.

5. *Carbon Packing Rings*—The carbon-packing rings must be constantly watched and fitted to the shaft to prevent air leaks at this point, which would lower the vacuum. Much more care must be taken in fitting the rings than is necessary with the average bearing surface or with boxes. They must be scraped carefully to the shaft.

6. *Water Seals*—The present type of horizontal turbine is sealed for vacuum by a thin film of water thrown out against the casing by a small centrifugal pump mounted on the turbine shaft. Turbine manufacturers advise using soft water in these seals, and their experience has proved the wisdom of this. With hard water, the parts will in time become covered with a precipitate which will finally either entirely clog up the openings, or eat away the pump blades, according to the composition of the water used. The writer has known this seal to be entirely filled with lime and magnesia in 30 days, and the operator was at a loss to know what affected the vacuum until the man from the shop arrived. This entails expense for repairs, or loss of economy through low vacuum by air leaks at this point and the loss of the use of the machine while making repairs.

The water seals should be regularly inspected for trouble, which may be more easily prevented than cured.

7. *Loss of Blading*—In the earlier types of turbines—the experimental stage—considerable trouble was caused by the blades pitting from the action of the steam upon them. When pitted they became weakened and broken, and in being thrown out struck and injured others. Poor workmanship in placing the blading was also frequently responsible for this trouble. Then, again, blades may be lost by carelessly leaving a nut, small bolt, washer, etc., in the turbine, after inspection or repairs.

In the horizontal turbine, where the clearance is at the periphery of the wheel, it is very important that the shrouding at the outer end of the blades should be perfectly secured. The writer knows of one instance where the end of a section of shrouding on the last row of the last stage gradually worked loose and began striking the casing. This made it become looser until it bent and struck the blades, tearing out or badly damaging all the blades in that row and the corresponding row of the stator. If this had occurred farther in on the stator, it would have torn all the blading out to the low-pressure end of the machine.

8. *Lubrication*—Next to keeping the blading in perfect condition, proper lubrication is most important. The oil must never be allowed to stop for even a few seconds, because it takes but a moment to burn up the bearings. The bearings are so hard to get at, that rebabbitting, or even scraping them, after burning, involves considerable expense.

Turbine makers generally recommend a certain oil which has been found suitable, according to their experience. It would be well to try this oil at first, but it should be closely watched. It may not meet the local conditions. Sometimes an oil is not uniform from month to month; such an oil will leave a deposit in the tanks and piping and probably cause a loss of oil pressure when least expected.

An alarm signal should be installed in connection with the turbine-oiling system so that instant warning may be given of loss of pressure below a certain point. Oil should be cared for in regard to filtering and cleaning about the same as for an engine.

9. *Rapid Starting*—The turbine can be more quickly made ready and cut into service than the reciprocating engine because of the smaller number of parts, but it is far more important that the turbine be thoroughly warmed up before being brought up to speed than is the case with the engine. The clearance between parts is so very small in the turbine, that unless completely warmed up the blades will be injured, due to the warping from uneven heating. The larger the turbine, the more caution should be taken.

10. *Cooling the Generator*—In the early types of turbines, no provision was made for cooling the generator during heavy load; that is, as far as providing openings and passageways in the generator casing itself is concerned. It was found that the capacity of the earlier turbine-driven units was much greater than the rating indicated, if means could be found to prevent the generator from heating. In very many cases, air drawn from outside the turbine room was forced into the turbine by fans, through a system of piping, thus increasing the rated capacity from 25 to 50 per cent. This is provided for in the later units. Besides the possibility of getting foreign particles or dust into the generator there is one source of danger in this operation. During the cold season, if too much air is drawn from outside, frost may be formed in the air passages; and when melted the water may be drawn into the generator. Means must be provided for properly mixing warm and cold air to prevent formation of frost. The air leaving the generator should always be warmer than that of the room.

11. *Cleaning the Generator*—The generator must be kept thoroughly clean. It must be blown out with air at regular intervals and frequently examined to see if it is free from oil or condensed steam. It is well to have a lamp, wired in with a switch, on a vertical turbine, placed below the generator and above the turbine, so that the generator may be examined at any time for oil or moisture, more especially while operating.

12. *Water in the Steam*—The steam

line should contain a separator near the turbine to catch all water which the steam may contain. The separator must have a trap to eject the water, leaving no chance for it to get into the turbine. Water getting into the turbine will slow it down and lower the frequency.

If from any cause water gets to the blading on the condenser side, it will slow down the machine and likely strip the blades, due to their high velocity. These blades have been torn from the rotor where a jet condenser was used, and water from some cause backed up into the machine.

13. *Magnetic Balance*—On a horizontal turbine the generator should have the proper magnetic balance to prevent wear on the thrust collar. The coupling is designed with room for some wear without damage, but if the thrust collar shows wear, it is advisable to watch it to see that the clearance on the coupling is not all taken up. When this occurs, the magnetic pull will be exerted against the dummies. This will be corrected by the maker when it occurs.

14. *Assembling after Repairs*—When the rotor has been taken out of a horizontal turbine for repairs every row of blades should be thoroughly examined *before assembling* for loose blades or foreign objects lodged in them. The stator should be examined just as closely. After the rotor is replaced and before the top of the casing is put on, the rotor should be turned over by hand. The operator should be familiar with the force required to turn it, or should be perfectly sure that it is free, with no undue friction from any cause. After the rotor is in and the top half of the casing is in place and bolted down, the rotor should be *turned over again by hand* to make certain that the blading has no interference. The same precaution should be observed with a vertical turbine in regard to turning around by hand. It may save serious damage to the buckets.

If the bearings are injured and have to be scraped, one should be very sure that the rotor in a horizontal turbine is centrally located, and that the bottom and top clearance, when the cover is put on, are the same. In the vertical type one must look out for alignment.

15. *Poor Mechanical Adjustment*—More care must be used in assembling and adjusting the working parts of a turbine than is usually the custom with an engine. The reader interested in this article should commit this to memory. The governor and controlling valves must be properly lubricated and cleaned, and must not be allowed to become gummed up. In a horizontal turbine, operated by a single controlling valve, sticking of the valve would mean sudden shutdown. No lost motion should be tolerated. It is assumed that the engineer is enough of a machinist, and enough of a stickler for exactness, to see to it that all turbine repairs are properly made; hence no details regarding this have been covered, for it can be left to his common sense to realize the importance of work well done.

Inspecting the Turbine

Owing to the necessarily fine adjustment of the moving parts of a turbine, it seems imperative that inspections should be made frequently. Troubles may be largely overcome by so doing. The engineer cannot see the inside of his turbines too often. He becomes so thoroughly familiar with all its parts that much time and money may be saved in time of trouble. Minute preparations should always be made for examination, so that it may be done with as little labor as possible. Troubles may be thus avoided which under other circumstances would mean a large expenditure for repairs.

It becomes necessary at times to completely overhaul a turbine. Where this is done for the first time, it is advisable to have an erecting man come from the factory who understands the machine. Later the engineer in charge will become familiar enough with the work to do it properly. Enough experience may be gained by working with the erectors at the time of installing for the engineer, unfamiliar with the turbine, to make the proper inspections and adjustments necessary from time to time.

A most valuable aid to the turbine engineer, is a detailed written record of his findings in these cases. Often facts taken from his record will strengthen the stand he may take with his superiors in regard to alterations or improvements he thinks advisable. This record will also indicate ways to avoid repetition of a certain accident, and will always serve him better than his memory alone.

Fig. 1 shows a repair order found very valuable for such a record, which is issued for every repair job to be done. This gives a detail of the repairs made. The important part of this information is tabulated on the sheet shown in Fig. 2. These repair orders are numbered and kept on file, and if giving more detail than it is wished to put on sheet 2, the repair-order number is put at the end of the notes, and can be referred to for full particulars.

POWER STATION REPAIRS – WORK ORDER No. 419

To F. A. Robinson Date Dec. 21 1907

Work to be done

Change oil piping #8 Turbine as decided upon

Immediately upon completion of above work, make full detailed report of same in space below, and return to office.

Date Completed Dec. 22 1907

Work Done

$\frac{1}{2}$" oil piping was changed to $\frac{3}{4}$" – this was thought necessary because $\frac{1}{2}$" was not large enough on account of the number of blades.

Put $\frac{1}{2}$" drain from tank to draw off the water carried over from the carbon packing

Use other side for writing if necessary Signed F. A. Robinson

FIG. 1.

F. R. Brosius

Economical Operation

Since the engineer's success depends upon his ability to operate his machinery at the least expense, as well as to keep things running smoothly, his constant aim should be to keep everything in the best repair consistent with the time and labor at his command. By so doing, accidents and shutdowns will be less probable. The vacuum will be kept at the highest point attainable, and no steam-saving factor in turbine operation is more important than carrying a high vacuum. Innumerable tests have proved that the economy of operation increases as the vacuum increases.

Station No 1 — TURBINE No 8 — 1907

DATE	STEAM ADMISSION	BEARINGS	CONDENSER	AIR PUMP	CIRCULATING PUMP	MISCELLANEOUS
Sept 19	Readjusted 5th valve.	Ground in west valves on west steps bearing pumps.				
20		Broke crosshead east steps pumps - replaced it with new one			Keyed crank	
22	Took off throttle valve - went over it	Ground in east valves - west steps pumps			Keyed crank + crosshead	
Oct. 11			Condenser cleaned			End pulley sprung onto taper bearing leaks - removed same - ground seat - went over pump. Made new pointer for controlling valve of stage valve
20	Put new gasket of "Superheat" packing on steam chest - south bank of valves.					
30					Keyed crank + crosshead	
Nov. 8	Turbine ran off with 1400 KW. Found 4th valve had become loosened + lengthened out, causing the bottom dog to miss. It also held the cam so as not to allow the other valves to close.					
9	Found cam on north bank of valves stuck, which caused it to work with rocker arm + hold valves up					
10	Examined throttle valve. Found gear shaft binding in bottom bush. Lined up boxes.	Ground in all valves on east steps pumps		Looked at valves - OK. - Tightened glands on air rods		Packed all valve stems.
17		Packed west steps pumps	Tested for air leaks - found one small one. Condenser cleaned.	Looked at valves + tightened up stuffing box glands.		Took out carbon packing - burnt to ring - circled up + bottom ring cut badly. Filled top rings open. Turned bottom rings. Put new packing rings in - had to scrape rings to fit. The rings were for larger shaft. Allowed .012" on bottom ring.
Dec. 8	Had throttle valve out - plugged holes in valve. Put new gasket on bend on north bank of valves. Had half of gear case off - worm filled up with dirt from the oil. Worm + gear look good.	Packed steam end of both steps pumps. Put oil shield on half of middle bearing.		Took out heads off - nuts bore. Made new nuts. Faced off 6 valves.	Took out heads off - nuts bore. Made new nuts. Put new Hayden packing on piston rod. Tightened nuts all over pumps. Repacked shaft.	
22			Tested for air leaks - found none. Condenser cleaned			Changed ½" oil feed pipe to ¾" - Put ½" drain on bottom of tank. See repair work order No 419

FIG. 2. F. R. BROSIUS

Time has also proved it advisable to use superheated steam; the temperature depends upon local conditions, there being a place where the saving effected in steam consumption will be over-balanced by the interest on extra capital invested and increased cost of maintenance. Erosion is more likely to occur where saturated steam is used.

The largest power plants of the country have a force of men constantly employed in testing turbines and generating equipment. The results of their tests are used in determining the repairs to be made. The engineer in a small plant would be greatly benefited by tests made upon the apparatus under his control. Knowing what the machinery will do is one of the greatest essentials of turbine operation. For instance, tests will show that at a certain per cent. of a turbine's rated load it will produce a kilowatt with the least steam. The wide-awake engineer will then endeavor to operate his machine at that load as far as plant conditions will allow. Again, with the vertical turbine, the amount of clearance between the nozzles and buckets greatly influences the economy of the turbine. As a result of a series of tests which the writer personally supervised, a saving of 6 per cent. was made by reducing the clearance on a vertical turbine from 0.120 to 0.090 in., while a subsequent test, after reducing the clearance from 0.090 to 0.050 in., showed a further reduction in steam consumption of only 0.2 per cent. In a horizontal turbine the clearance on the dummies should be so adjusted that the balancing pipes will be at a low temperature.

On another occasion, placing a new set of intermediates in the turbine saved 8 per cent. in steam consumption, greatly over-balancing the cost of the new parts.

Nearly every engineer in charge of a power plant has enough time and labor at his bidding, if he only thinks so, to make tests which will give him valuable information as to the condition of his generating equipment. He may have to use a fireman or an oiler to weigh water and possibly work them overtime on such occasions, but he will always find the old adage applies to special work around the station, "Where there is a will there is a way." Maybe he cannot go into the finer points regarding efficiencies, etc., but he can at least obtain the correct steam consumption. As soon as possible after the turbine is erected, he should make his first test. This is generally made to see that the maker meets guarantees, and all subsequent tests should be compared with this one. Results of turbine tests made by reputable engineers can be easily obtained and are frequently reported in the technical magazines. The engineer should compare these results with his own, using the figures on turbines similar to his, and if his are not up to the mark, he should find out why, and set to work to improve them. This not only means a saving to his employers, but reflects to his credit, inspiring confidence in his ability, watchfulness and reliability.

To summarize—the engineer should

1. Make frequent inspections.
2. Keep the condensing apparatus in condition to maintain the highest possible vacuum.
3. Make tests when thought advisable.
4. Keep the blading in the best repair possible and adjusted to produce a kilowatt at the lowest possible cost as determined by test.
5. Give particular attention to the governor and valve-gear adjustment.
6. Use great care in assembling.

Distortion of Indicator Drum Motion

By Julian C. Smallwood*

Results of an experimental investigation of the inaccuracies of indicator drum motion caused by the variable stress, and consequent stretch, of the indicator cord.

*Associate professor of experimental engineering, Syracuse University.

Stress in an indicator cord is the resultant of three distinct forces; namely, the drum-spring tension, the force required to overcome the inertia of the drum, and the force of friction between the drum and its spindle. The force of the spring increases with the forward stroke of the drum and decreases upon the return; the increase is usually uniform with the motion.

In general, at the beginning of the forward stroke, inertia increases the cord stress since, as the speed is increasing, inertia effects a tendency of the drum to lag. Beyond midstroke, however, the speed is decreasing, and, as the drum tends to exceed the velocity induced by the cord, a slackening results. Upon the return stroke the force of acceleration varies in the same way as upon the advance.

The force of friction is always opposite to the drum motion and therefore changes its direction at each stroke. During the forward stroke it tends to increase the cord stress, and upon the return to decrease it. The three forces and the resultant cord stress are represented graphically by Fig. 1.

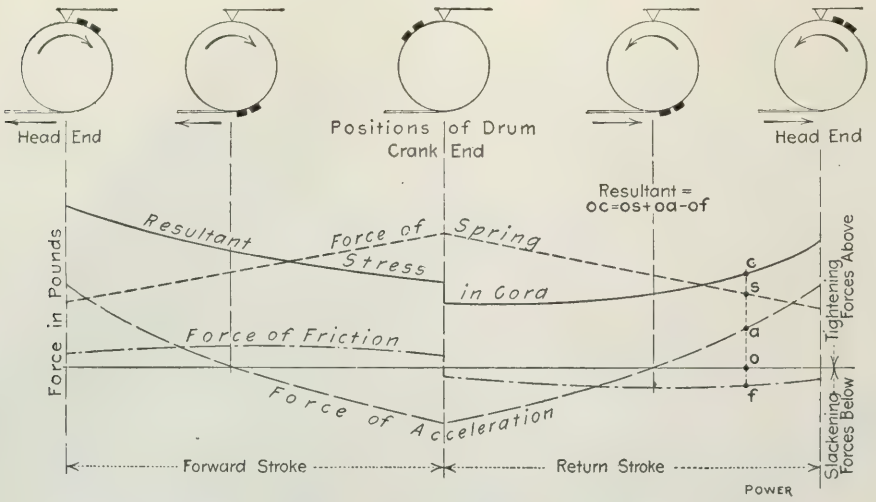


FIG. 1. DIAGRAM OF FORCES PRODUCING STRESS IN INDICATOR CORD AT DIFFERENT PARTS OF STROKE

CORD STRESS DIAGRAMS

Autographic cord-stress diagrams obtained by the use of a Brown drum-spring tester,† in which stress is represented vertically and stroke horizontally, are shown in Fig. 2. The diagrams bear the records (by the dotted lines) of cord stress at dead slow speed. At the higher tensions, these records are double lines, for at such tensions the friction is materially increased. At less than 100 r.p.m. there is practically no effect from inertia, the curve at such speeds being closely coincident with the "no-speed" curve. As the speed is increased, the resultant cord-stress curve approaches the horizontal because the increased force of acceleration tends to equalize the cord stress. When the speed is made still greater, the force of acceleration predominates and the slope of the cord-stress curve is reversed.

It was found that an increase in the spring tension at a given speed made the range of stress higher with a greater difference between the stresses at corresponding points on the forward and return strokes.

RELATION BETWEEN STRETCH AND STRESS

Experiments to determine the relation between the stretch of the cord and the stress causing it yielded unexpected results which, because of the general application of the principle discovered, were presented in a paper on "The Exception to Hooke's Law," appearing in the October, 1911, issue of the *Physical Review*. The purpose of the tests was to find if the stretch of the cord varied in the same way as under static conditions when the stress is rapidly applied, as is the case in the operation of the indicator. The following facts were established: First, the relation between stress and stretch is independent of the speed of variation of the stress. Second, for increasing stresses over about 1 lb., the stretch increases uniformly with the stress, according to Hooke's law. Third, for decreasing stresses, the stretch decreases according to a different law. This is because the work done upon the cord, when an external force stretches it, must be greater than the work returned by the cord when its fibers draw up, due to the friction of the fibers upon one another. The effect is a "lag" in resuming the original length which increases the drum-motion distortion.

Turning to the effect of the established variations of stress and stretch upon the drum motion, it may be observed that the ideal operating condition demands constant cord stress. This, however, can never be attained, but disregarding friction, there is one speed at which the resultant of the spring and acceleration forces is approximately constant through the stroke for any particular drum and spring.

If an indicator drum be put on its extremes of travel and vertical lines ruled on the card to indicate these extremes, then horizontal lines made at increasing speeds will overtravel the vertical lines. From the diagrams of Fig. 2, it is

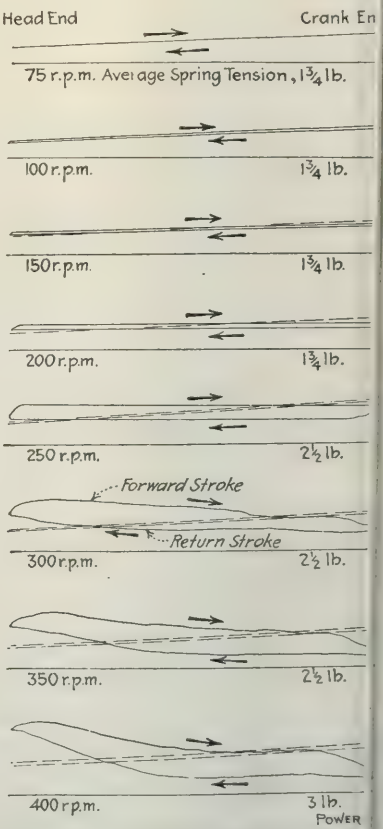


FIG. 2. CHORD STRESS DIAGRAMS

†The drum-spring tester diagrams presented in this article were taken by Edward H. Zeiber.

hat the effect of speed is to increase the tress at the head end and decrease it at he crank end. At the head end, therefore, the cord is longer and the drum ravels further in this direction. Similarly, at the crank end, the cord is horter and the drum is pulled further ward that end.

Overtravel in itself does not cause rror in the indicator diagram, although it enerally accompanies it. If the whole ing stress is to make the cord still longer, and to increase the error. During the return stroke, by similar reasoning, the cord is too short, a point on the diagram obtained is to the right and again behind its true position, since the motion is reversed. There is no effect from cord lag on the return stroke since the stress is increasing. The net effect of the stretch of the cord, then, is to make the mean effective pressure appear

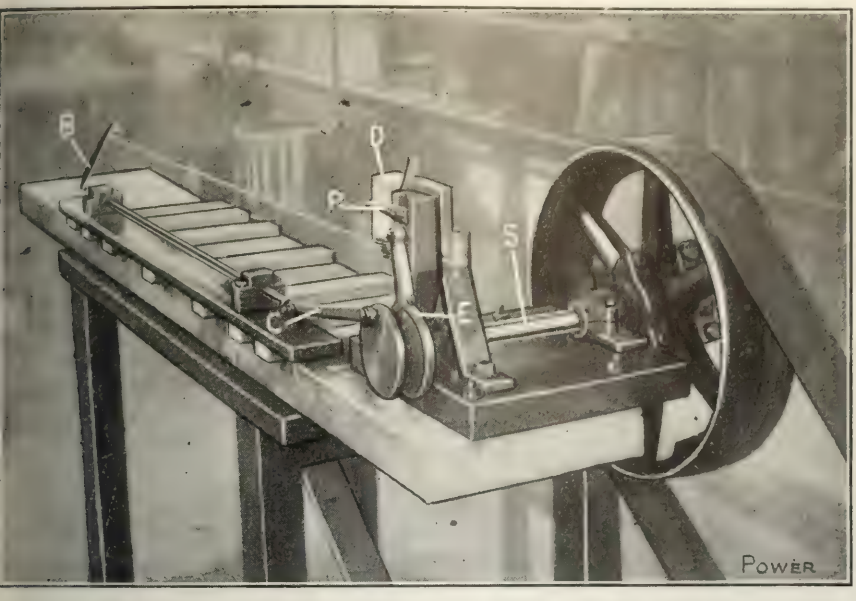


FIG. 3. VIEW OF APPARATUS FOR RECORDING INDICATOR DRUM MOTION

iagram were lengthened uniformly, its roportions would remain correct. A uniformly increasing cord stress would have his effect because then the cord stretch ould be uniform and the diagram would e merely lengthened uniformly in all s parts, but a uniformly decreasing cord tress would not have this effect because f the property of the cord to lag. At ny point of the indicator diagram, its

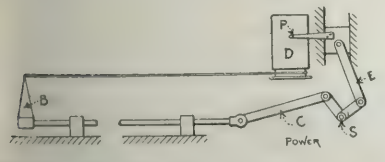


FIG. 4. SHOWING PRINCIPLE OF APPARATUS FOR RECORDING DRUM MOTION

rror should not be judged by the absolute stress or stretch at that point, but nstead by the difference between this tress or stretch and that necessary to roduce a uniform lengthening.

The following reasoning has reference o conditions such as shown by Fig. 2 t speeds above 200 r.p.m. During the orward stroke, the cord stress is dereasing in value, but not uniformly; ence the cord is too long and a point n the resulting indicator diagram is to he left and behind its correct position. The effect of cord "lag" with decreasing stress is to make the cord still longer, and to increase the error. During the return stroke, by similar reasoning, the cord is too short, a point on the diagram obtained is to the right and again behind its true position, since the motion is reversed. There is no effect from cord lag on the return stroke since the stress is increasing. The net effect of the stretch of the cord, then, is to make the mean effective pressure appear smaller, and the cutoff, compression and release earlier than their true values.

EXPERIMENTAL RESULTS

The distortion of the diagram could be determined inferentially from the analysis already presented. For experimental verification, however, the author devised the apparatus shown in Fig. 3 and illustrated diagrammatically by Fig. 4. A shaft *S*, whose rotary speed could be controlled by a motor having a water rheostat in its armature circuit, actuated two similarly proportioned crank trains, *E* and *C*, set exactly 90 deg. apart. The motions of the two crossheads were thus exactly proportional at all parts of their strokes. The indicator drum *D* was rotated by fastening its cord to a bracket *B* carried on an extension of the horizontally moving crosshead. The crosshead with vertical travel carried a pencil point *P* which traced a diagram upon the drum. A diagram thus obtained is one of the crosshead motion shown vertically and drum motion horizontally. If a rigid connector between the drum and the bracket had been used instead of an indicator cord, the motion of the drum would have been exactly proportional to that of the pencil, and the diagram would have been an inclined straight line. The effect of an elastic connector, as cord, variously stressed, is to give a curved line. If a straight line is drawn between the highest and lowest points of this curve, then the horizontal departure of any point on the curve from the straight line shows the error in the drum motion at that point.

Fig. 5 shows a typical "error diagram" taken with this apparatus, and Fig. 6 the method of correcting the indicator diagram by its use.

Tests were made on two indicators, *A* and *C*, of the same make, the former with a 2-in. drum and the latter with a 1¼-in. drum; hence the inertia effects of *C* were somewhat less than those of *A*. Two kinds of cord were used, but the error diagrams presented are all for one cord, a 4-ft. length of which was used. The stroke of the horizontal crosshead was 3.72 in., corresponding to a diagram 3.72 long at no speed. Tests were made for various spring tensions and for each the engine speed was varied from very slow to about 350 r.p.m. The effect of the forces previously analyzed

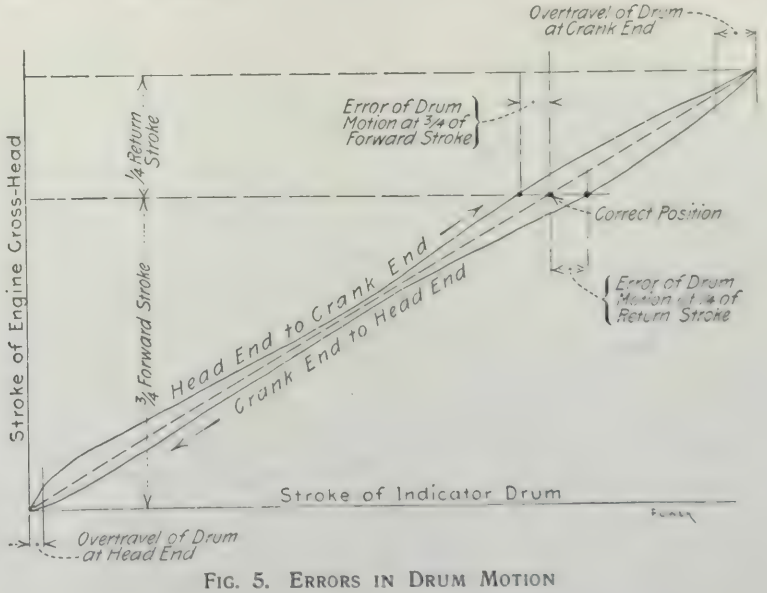

Overtravel of Drum at Crank End
Error of Drum Motion at ¾ of Forward Stroke
¼ Return Stroke
Correct Position
Error of Drum Motion at ¼ of Return Stroke
Stroke of Engine Cross-Head
¾ Forward Stroke
Head End to Crank End
Crank End to Head End
Stroke of Indicator Drum
Overtravel of Drum at Head End

FIG. 5. ERRORS IN DRUM MOTION

was obtained by leading the cord straight from the indicator drum. A series of similar tests was made with the cord running over one and two guide pulleys, such as are almost invariably used when indicating an engine, and these, perhaps, yielded the most interesting results of the whole investigation.

Referring to Fig. 7, it is seen that the

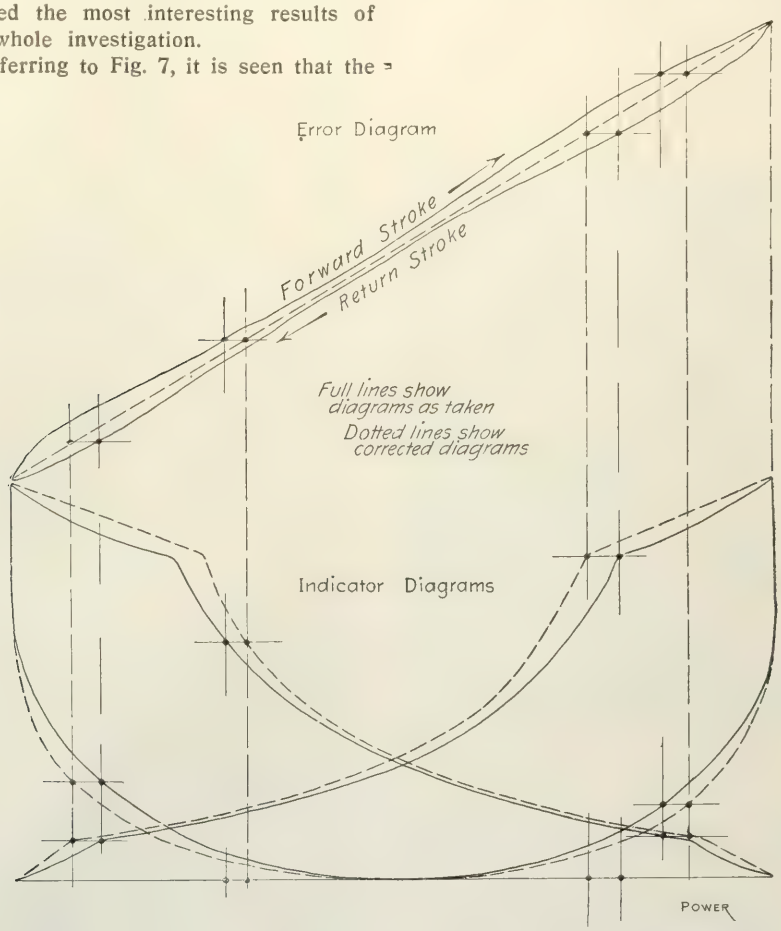


FIG. 6. METHOD OF CORRECTING INDICATOR DIAGRAMS FROM ERROR DIAGRAMS

best spring tension for indicator *A* when running at 300 r.p.m. is between 3 and 4 lb. A higher value increases the distortion by increasing the friction caused by the reaction on the drum spindle. With a 4-lb. spring tension, the maximum error is 0.1 in., occurring at about one-eighth of both forward and return strokes. The 1½-lb. spring tension gives excessive distortion, at the maximum, 0.37 in. The crank-end overtravel shows by its amount that the cord stress was reduced to zero at this end, yet there was no visible slackening of the cord. This phenomenon has been frequently observed, that the drum will actually overtravel the cord at the crank end, but not until the overtravel has become excessive will the slackening and consequent vibration of the cord be apparent to the eye.

Fig. 8 shows that at 150 r.p.m. the indicator diagram was practically correct. At 250 r.p.m. the errors grade from zero to about 0.1 in. at several points in the stroke.

The effect of horizontal distortion upon measurements from the indicator diagram will vary greatly with the type of diagram. A diagram having late cutoff and compression will be but little affected as distortion of horizontal lines appears only at their ends. Also, in general, the less the area of the diagram, the more will be the percentage of error in the mean effective pressure. But the effect of horizontal distortion upon the apparent positions of the events of the stroke may be large and may involve considerable inaccuracy in such calculations as for cylinder condensation, clearance, the equation of the expansion curve, valve setting, etc.

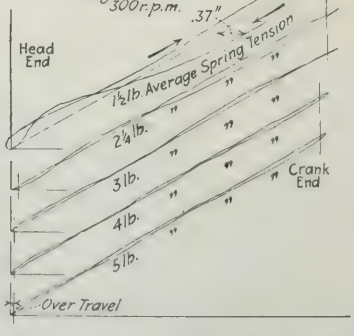


FIG. 7. EFFECT OF TIGHTENING DRUM SPRING

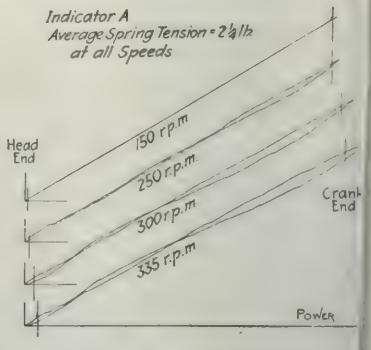


FIG. 8. EFFECT OF SPEED

GUIDE PULLEYS

The effect of guide pulleys used
the indicator is similar to that of dr
spindle friction and imposes an a
tional tightening force on the forv
stroke and slackening one on the ret
This is clearly shown by Fig. 9,
which it appears that the errors are
only very materially increased at
higher speeds but, at the lower
distortion is created where it did
exist before. The further the pu
from the indicator, the less is its e
to stretch the cord abnormally.

It is impossible to offset the stre
due to guide pulleys by tightening
drum spring, the effect being m
to increase the friction still more
to aggravate the trouble. Further
it appears from the results that
impossible to get a correct diagra
any speed, even if the best quali
cord is used, when the cord is
than 2 ft. long and run over a
pulley.

Fig. 10 shows the effect upon indi
diagrams of the errors presented by
9. It may have been observed in
of the error diagrams that the
advanced its correct position inste
lagging behind it on the return s
This is caused by the departure c
acceleration force from the variatio
sumed in Fig. 1 because of the s
of the cord and the consequent c
in the driven velocities.

GENERAL DEDUCTIONS

The obvious method of reducing
zontal error is to use cord which str
so little as to make the errors
cases inappreciable. There is, ho
no such cord on the market to the w
knowledge, the best obtainable gi
stretch of 0.008 in. per pound pe
of length. Using 4 ft. of this cord
cator *C* could be run at 300 r.p.m

a maximum error of 0.06 in., b
eighth of a turn about a guide
caused this error to be mor
doubled.

The price of indicator cord is no criterion of its quality; hence it is advisable to test the cord to be sure that it does not stretch excessively. This may be simply done as follows: Tie one end of about 4 ft. of the sample to a fixed point on a bench or table and the other end to a spring balance. Mark this end a few inches from the balance with a fine ink line, and under this line place a piece of paper or a foot-rule. Stretch the cord by pulling the balance horizontally until about 5 lb. are indicated. Now reduce the force to about 1 lb., and repeat this procedure a few times. The elongation may then be noted for any applied range of stress. As the cord in the operation of the indicator is generally not stressed more than 5 lb. or less than 1 lb., this range was used in the results previously quoted. A good cord should not stretch more than 0.03 in. when stressed to 2.5 lb. It will be found, however, that the elongation and contraction of the cord are very much greater at stresses less than 1 lb. This explains the phenomenon that there may be marked overtravel at the crank end of the drum motion and yet the cord will not show it by visibly slackening or vibrating.

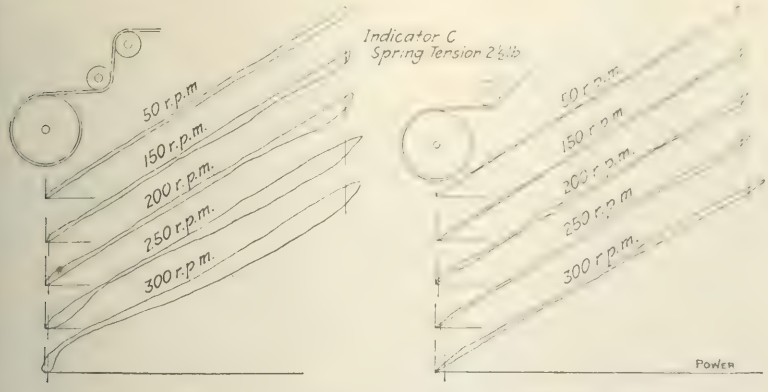


FIG. 9. EFFECT OF GUIDE PULLEYS AT DIFFERENT SPEEDS

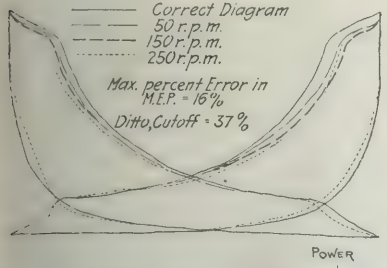


FIG. 10. EFFECT OF TWO GUIDE PULLEYS AT DIFFERENT SPEEDS

The writer has experimented with a number of high-grade stranded bronze wires, but has found none conveniently adaptable to indicator work. Such wire may be obtained sufficiently light and stiff, but it wears out very rapidly when subjected to flexure in operating an indicator drum.

For accurate work and where guide pulleys are unavoidable, wire may be used to advantage as part of the connector. A short piece of indicator cord, about 4 in. long, may be wrapped about the drum and to it the wire attached. Wherever the connector passes over a guide pulley, a short length of cord pieced to the wire may be used. Such an arrangement, though effective, is inconvenient to set up, and, unless the wire is very flexible, it will kink and act as a spring, causing worse error than would a cord.

The effects of inertia may be reduced by adjusting the length of the diagram. The force of acceleration varies directly with the length of the stroke of the drum and with the square of the engine speed. Assuming that a particular indicator gives a reasonably correct diagram S inches long at a speed of N revolutions per minute, then at another speed, n, and diagram length, s, if

$$S \times N^2 = s \times n^2$$

there will be the same inertia forces. Applying this to indicator A, it is found from the error diagrams and from the cord stress diagrams that this indicator will give practically no error at 200 r.p.m. and a 3.6-in. diagram. Then, if it is wished to run at a speed of 250 r.p.m., the length of the diagram should be

$$s = \frac{S N^2}{n^2} = \frac{3.6 \times 200^2}{250^2} = 2.3 \text{ in.}$$

However, not much use can be made of this method as, at 300 r.p.m., the diagram becomes so short as to be impracticable.

The inertia forces may be balanced at different speeds by using different stiffnesses of spring, but this, too, is generally impracticable.

In conclusion it may be said that, for accurate work, it is necessary in most cases either to use an indicator especially adapted to the operating speed, or to use wire instead of cord. Guide pulleys should under all conditions be avoided. When a small percentage of error in the results from engine indications is not objectionable, the ordinary indicator may be employed, but the best quality of cord only should be used except at the speed for which the instrument is adapted.

Replacing Boilers and Engines

BY F. C. SNEIDER

The work of installing new engines and boilers to replace those in service is dreaded by both manufacturer and engineer. While the time lost in completing the work is usually a large item, in the following case much time and money were saved by exercising judgment and foresight.

Two horizontal return-tubular boilers, each 18½ ft. long and 72 in. in diameter, were removed and the new boilers of the same type and size installed, with a loss of but five working hours. An old 20x48-in. Corliss engine was also removed and a new 22x48-in. Corliss engine erected on the original foundation with a loss of only seventeen working hours. This shows how little actual working time need be lost if the work is done when power is not required, as on holidays and Sundays.

As it was decided to install the new boilers on Labor Day, at 6 p.m. on the preceding Friday the fires were drawn and the work of removing the old boilers begun. At 6 p.m. the following day both boilers and settings had been removed and the settings for the new ones started. The bricks used in the new settings were dipped in water and firmly rubbed to the surface of the adjacent bricks. This accounts for the satisfactory manner in which the walls withstood the intense heat to which they were so soon subjected.

The new boilers were resting on their settings at four o'clock Sunday afternoon, and as soon as the necessary connections were made the boilers were filled and the fires started. The masons and steam fitters in the meantime completed the brickwork and boiler connections. The steam pressure was gradually increased on Monday night so that at 7 a.m. Tuesday the engine was started on the day's run. The total time required to renew the boilers, settings and connections was 75 hours, and the actual loss to tenants only five hours.

Thanksgiving Day was chosen to remove the engine, and at 4 p.m., Nov. 29, the work was begun. At 4 p.m. Saturday the old engine had been removed and the foundation made ready by removing the top bricks to a depth of 24 in. and substituting a bed of concrete of the same thickness. At 4 a.m. Monday the erecting was completed and the engine started. The speed was gradually increased so that at 7 o'clock the full load was thrown on the engine. The total time required for this installation was 111 hours, but the actual working time lost to the tenants only 17 hours.

While the engine was being installed the belts, idlers, jackshafts and pulleys were renewed, and as the pumps, piping, etc., were new or thoroughly overhauled during the preceding year the generating

equipment of the plant was practically renewed with a loss of only 22 working hours. When it is considered that 10 shoe manufacturers employing nearly 5000 hands are dependent on this plant for heat, light and power, the saving of time over what is usually required for such installations represents a considerable sum of money.

Single and Twin Filter and Grease Extractors

The Elliott Co., 6907 Susquehanna St., Pittsburgh, Penn., is manufacturing two types of filters and grease extractors. The single filter consists of a filtering chamber containing the filtering media and a divided base directing the flow of liquid through the filtering chamber or bypass when cleaning. Valves are provided at the end of the base to put the filter in or out of service. Terry cloth of double thickness and supported by spacers and spreaders about a central column is used for the filtering media.

The twin filter consists of two filtering chambers having a common base, but is otherwise constructed similarly to the single filter except that the bypass is eliminated.

Referring to Fig. 1, the liquid to be filtered enters at the left from the feed pump and passes up and out through the central slotted column, coming in contact with the filtering media, which removes the impurities. The filtered liquid then passes into the surrounding chamber, down and out at the right into the line to the closed heater or wherever desired.

To clean the filter, the flow is directed from the filtering chamber containing the fouled media by shifting the valve at either end. By admitting water from the feed line through the connection in the filtering chamber, the direction of flow is reversed, and the water will partially remove the impurities from the interior surface of the filtering material and discharge them through the drain in the base. Fig. 2 shows a filtering cylinder removed from the filter body.

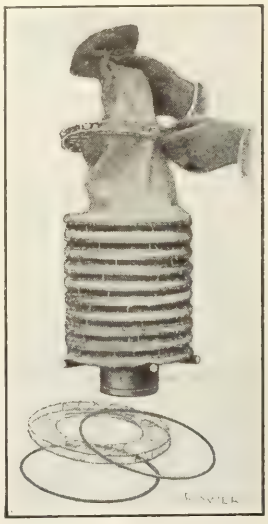


FIG. 2. CYLINDER REMOVED FROM FILTER

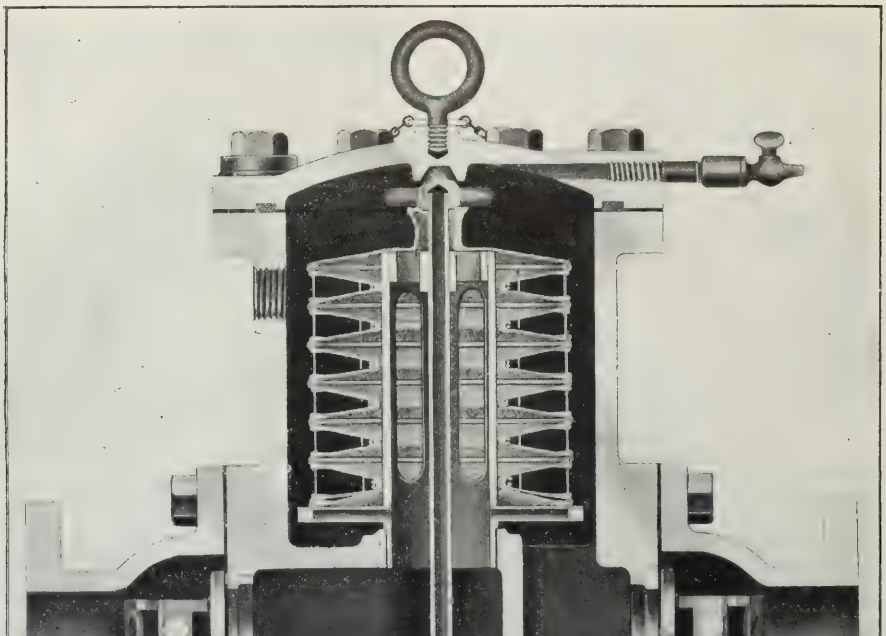


FIG. 1. SECTIONAL VIEW OF GREASE EXTRACTOR

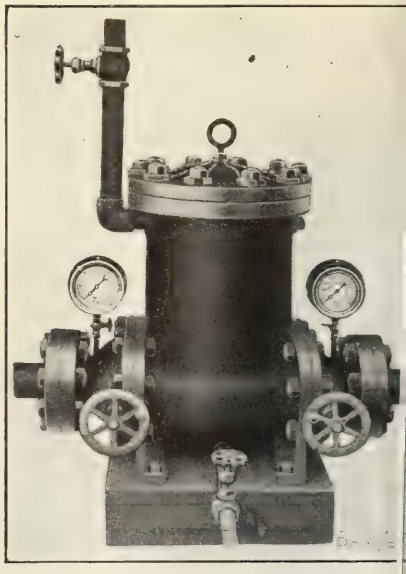
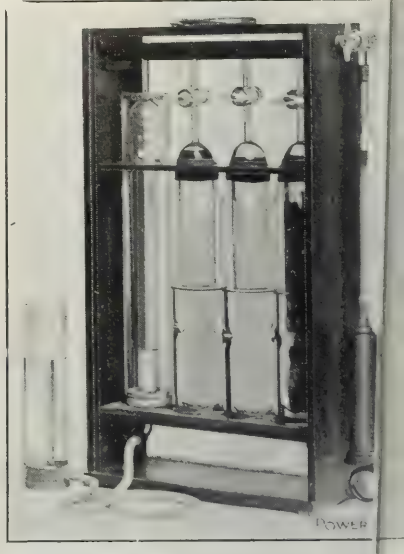


FIG. 3. EXTERIOR VIEW

The filter is made in capacities of f
20,000 to 250,000 lb. of water per h

Williams Improved Flue G Apparatus

The instrument illustrated is com and portable and is designed to facil the work of the operator. The mea ing burette is placed at the left side lowing freedom of the right han manipulating the stop cock, etc.



WILLIAMS IMPROVED FLUE-GAS APPA

The explosion burette is fitted wi minals permanently connected to b posts on the outside of the case. bulb is low upon the tube, permitt c.c. of clear gradation on the small of the burette above the swell. are two forms of absorption pipet simple and the bubbling, either for ping interchangeably into the pipe or solution reservoirs, and the at the top is effected by a rubber

The bubbling pipette accomplishes absorption very rapidly because the gas is caused to bubble through a long column of liquid, and the bubbles are finely broken up by the thin glass tubes with which the pipette is filled to give a large absorption area in the wet surfaces of these tubes.

The detachable three-way T-stop-cock with hard-rubber pump and valves makes it possible to take a dry sample of gas directly from the source of supply, eliminating the inaccuarcy and bother of collecting over water, transference, etc. After connection with the supply, the pump is operated, discarding the gas in the connecting line until the fresh sample is brought up to the apparatus, when, by turning the stop-cock and lowering the water-level bottle, the measured sample is taken directly into the burette.

The introduction into the line of an aspirator tank also enables an average sample to be taken covering any desired period of time. This method of control has advantages in that its accuracy and adjustment are unquestionable. It is also possible to determine carbon monoxide.

The apparatus is manipulated by lowering the water level and, after closing the three-way stop-cock, drawing the solutions in the pipette up to a definite reference mark on the capillary tubes, about midway between the stop-cock and the body of the pipette. The stop-cock is then opened and by raising the water-level bottle the burette is completely filled with water. The apparatus is then ready for a sample.

The device is connected either with the gas supply directly, or with the sample-collection bottle, and with a few strokes of the pump a fresh sample of the gas is drawn up to the apparatus, discarding the air in the connecting line out through the slit or valve. The stop-cock is then turned until the head is on top and, by lowering the water-level bottle, a little more than 100 c.c. of the sample is drawn into the burette. This is allowed to stand until the gas has come to the temperature of the apparatus when a final adjustment is made to 100 c.c. When readings are taken, the bottle is held so that the water level is the same as that in the burette.

This apparatus is manufactured by F. M. Williams, 43-44 National Bank & Loan Bldg., Watertown, N. Y.

Air Compressors in the Transvaal

In the Transvaal mines, South Africa, according to the government mining engineer, in 1910 there were 257 compressors, with 100,291 hp., while only 400 hp. of compressed air was purchased. No separate returns of the value of air compressors were rendered to the department.

The gold mines in the Johannesburg consular district employ on an average three 50-drill compressors on each mine for the driving drills. The value of the total imports for both gold and coal mines for the period mentioned was $543,402. Up to about two years ago the air compressors were almost entirely driven by steam, but since then electric power has been furnished to nearly all of the mines, and steam power for air compressors has been superseded by electricity.

Purchased Current for Summer Service

In a recent contribution to a discussion before the City Club of Chicago, W. A. Blonck, consulting engineer of that city, brought out some interesting points tending to show that it would be to the mutual advantage of both the public-service corporation and the office-building owner if the isolated plant in the average office building were closed down during the season when steam-heating service is not required and the current necessary for light and power were purchased from the central station.

In driving home his arguments he pointed out that but 12 per cent. of the coal burned in the central station is converted into salable energy, the rest is simply used to heat that particular body of water which supplies the condensers. On the other hand, 70 per cent. of every ton of coal burned in a heating plant is converted into usable heat and about 200 kw.-hr. can be obtained as a byproduct, the latter representing a value of from $6 to $20, depending on the rate at which it is sold. From this it will be seen that the building owner who operates a moderately large heating plant during the heating season without availing himself of the electrical energy obtained with but a very slight reduction in the heating effect of the coal, may be compared with the old-fashioned butcher who sold only the meat and threw the rest of the animal away, while it is well known that the modern packer has grown rich on the profits from that part of the animal which but a few years ago was considered valueless.

Entirely different, however, is the aspect in the summer months when the coal is used solely for generating current. Then, according to Mr. Blonck, the owner must purchase, haul by wagon and burn in the congested district, say 4 tons of coal to develop a given amount of electrical energy which can be produced at the central station from about 1 ton. Thus, considering the matter from a manufacturing viewpoint, if the larger manufacturer, or central station, pays $1 for its raw material the small manufacturer, or building owner, must pay $5 for the raw material necessary to produce the same amount of finished product. This is allowing $1 for buying and handling in team-load lots instead of car-load lots.

It seems evident, then, that the building owner should be able to buy electrical service from outside source in the summer cheaper than he can produce it, and allow a good margin of profit for the public-service corporation. The central station would secure a larger summer load. This would mean a higher yearly load factor, decreased fixed charges per unit of energy delivered and greater revenue. The property owner would benefit financially because he would get current during the summer for less than it would cost him to make it. He could afford to use chaper engines because their efficiency is of relatively small importance when all the exhaust steam is being used for heating; hence his total investment would be less. Finally, during the summer he would be rid of the radiant heat of the boilers, engines, piping, etc., as well as the dust and dirt caused by coal and ash handling.

The city as a whole would be bettered because there would be less smoke and 50 per cent. of the coal now burned during the summer in the business district would be saved.

Faulty Separator Trap Connection

By E. B. Lloyd

High-speed, automatic cutoff engines are frequently damaged by water entering the cylinders. To prevent this the steam pipe should contain a reliable trap-drained separator placed as close as possible to the engine, and the cylinder should have drains and dependable release valves.

The separator-trap discharge and cylinder and steam-chest drains are frequently found emptying into the exhaust pipe close to the cylinder. The separator trap should discharge into a return tank or into the sewer and never into the exhaust pipe, or into the same line with the other drains from the engine.

When an automatic engine is first brought up to speed with no load the cutoff is so early that an indicator diagram would show practically no steam line and frequently the expansion line would be carried below the atmospheric line, forming a negative loop.

Under such a condition, the pressure in the exhaust pipe exceeding that in the cylinder when the valve is opened, the previously exhausted steam along with the water discharged from the separator trap rushes into the cylinder and the cylinder drain cock at that end of the cylinder if it is open.

The result is at least a disagreeable valve slap, if not a blown-out cylinder head, a bent crankshaft or a ruptured flywheel.

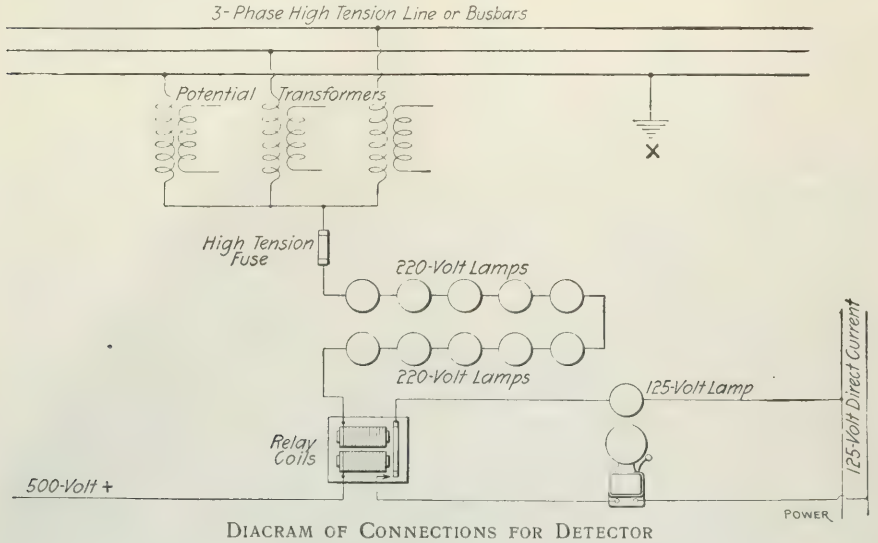
Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Ground Detector for High Tension Circuits

By J. W. Himmelsbach

While there are a number of instruments on the market for indicating the presence of a ground on a high-tension circuit, they are practically all of the static type. Although operating satisfactorily, they as a rule only indicate a ground; consequently, unless the operator happens to be where he can see the indicator, he may be unaware that a ground exists. To provide a detector which would not only indicate, but also notify the operator, called for the design of the apparatus herein described. This will not indicate the magnitude of the ground, but only its presence, and if used in conjunction with static detectors it provides a reliable and sensitive means of attracting the attention of the operator; he may then look at the static detectors to ascertain the intensity of the ground and also the individual phase or phases affected.



Diacram of Connections for Detector

A source of direct current is necessary, preferably a three-wire, 500-volt system, with a grounded neutral. For application to a three-phase, three-wire system the following material and apparatus is needed: Three potential transformers, with the high-tension winding designed for the voltage of the system, ten 16-cp. 220-volt incandescent lamps, one 16-cp., 125-volt lamp, one telegraph relay, wound for about 1000 ohms, and one ordinary electric alarm bell.

The primaries of the potential transformers are connected in star, across the three wires of the system, the secondaries being left open and the ends well taped to prevent accidental short-circuits. From the neutral point of the primaries, a wire is run to one terminal of a high-tension fuse and from the other terminal of the fuse a wire is run to one end of a bank of ten 16-cp., 220-volt lamps, connected in series. From the other end of the bank of lamps a wire is run to that binding post of the relay to which the magnet coil is connected. The other end of the relay coil is connected directly to the positive side of the 500-volt direct-current system. The 16-cp., 125-volt lamp is connected in series with the bell across a 125-volt direct-current circuit, the contacts on the armature of the relay closing this circuit. A storage or dry battery may be used in place of the 125-volt circuit, by omitting the lamp, but the former method is the better, for, in case the bell refuses to operate the lamp will light. A full diagram of connections is given in the illustration. The operation of the apparatus is as follows:

Suppose a ground to develop as at *X;* then, keeping in mind that there is a potential difference of 250 volts between one side of the 500-volt circuit and the ground, a current will flow from the positive wire, through the relay magnet coil, to the lamp bank, through that to the fuse and on to the neutral point of the transformers. At that point the current will select the transformer connected to the grounded phase and so flow to the ground. The current passing through the relay coil attracts the armature which in turn closes the contact in the auxiliary circuit; with this closed the lamp will burn and the bell ring, thus calling the attention of the operator.

This apparatus has been found to work admirably in spite of two disadvantages, which, although undesirable, are nevertheless unavoidable. First, connecting the apparatus to a three-wire system, operating with an ungrounded neutral paves the way for a difference of potential between any phase wire and the ground equal to 250 volts plus or minus the system voltage divided by $\sqrt{3}$; in a system operating at 6000 volts between phase wires, the potential difference between any phase wire and the ground is equal to 250 volts plus or minus 6000 divided by 1.73, which is equivalent to a varying potential of from 3210 volts to 3710 volts. The other disadvantage is, that on any occasion when lightning happens to strike one of the wires the invariable result will be the burning out of the magnet coil. However, the first may be in part cancelled by exercising greater care in coming in contact with the line wires, and the second will not occur so frequently as to make the item of rewinding the coils very expensive.

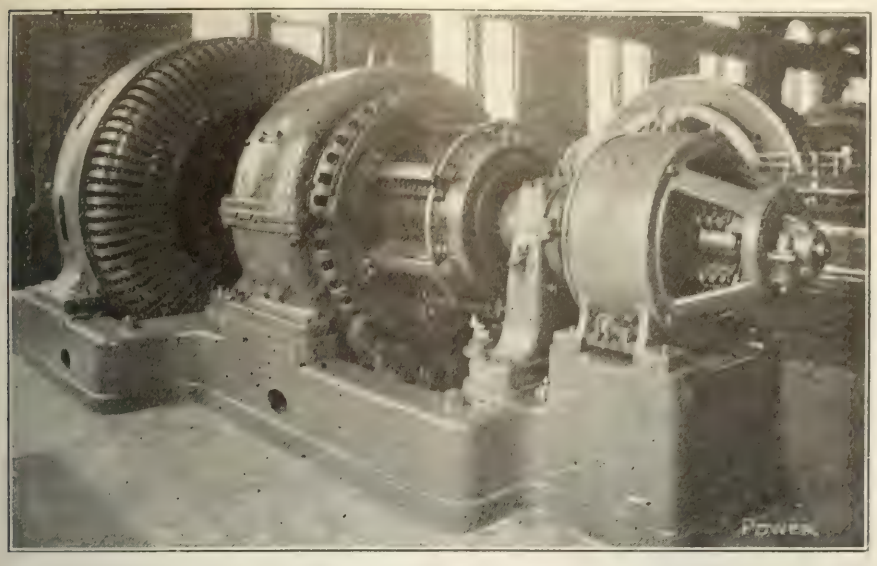
Synchronous Motor-Generator

An interesting application of a synchronous motor-generator for power-factor correction is that at the plant of the Spang Chalfont Co., at Etna, Penn. There are two mills, one near the power house and the other about 4500 ft. distant. Under the conditions that obtained in the plant prior to the installation of the synchronous motor set, the first mill was operating at a power factor of about 48 per cent. Another synchronous set previously installed takes care of the second mill, originally at about as low a power factor. Therefore, before the present set was installed, the generators at the power house were operating at about 75 per cent. power factor. To carry the load at this low power factor it was necessary to operate four generating units leaving no spare.

The low power factor was due to the presence of a large number of induction motors which are underloaded for a considerable part of the time.

The synchronous motor-generator set was temporarily installed in its present location in the power house to raise the power factor sufficiently to permit keeping one or two of the four generating

units shut down. This renders available one or two spare units. Moreover, the direct-current generator of the synchronous motor set now carries the load of a former direct-current engine-driven gen-



SYNCHRONOUS MOTOR-GENERATOR SET FROM EXCITER END

erator, still further increasing the economy.

In operation, the synchronous motor of the set is overexcited, thereby raising the power factor of the generators to 97 or 100 per cent. Under existing conditions, the motor is usually operated at 20 per cent. leading power factor. At present the actual kilowatt load on the generator of the set is small, which, with full input to the motor, accounts for the low leading power factor.

The set is driven by a 220-volt, three-phase, 60-cycle synchronous motor rated at 1000 kilovolt-amperes, and running at 600 r.p.m. It is of the three-bearing design and has, for the synchronous motor, a direct-connected exciter, overhung on the shaft, beyond the generator outboard bearing. The illustration shows the set from the exciter end. The generator is a direct-current, 400-kw., 220-volt, compound-wound machine.

CORRESPONDENCE

Wattmeter Problem

The diagram shows a proposed wattmeter installation on which I would like to have comments.

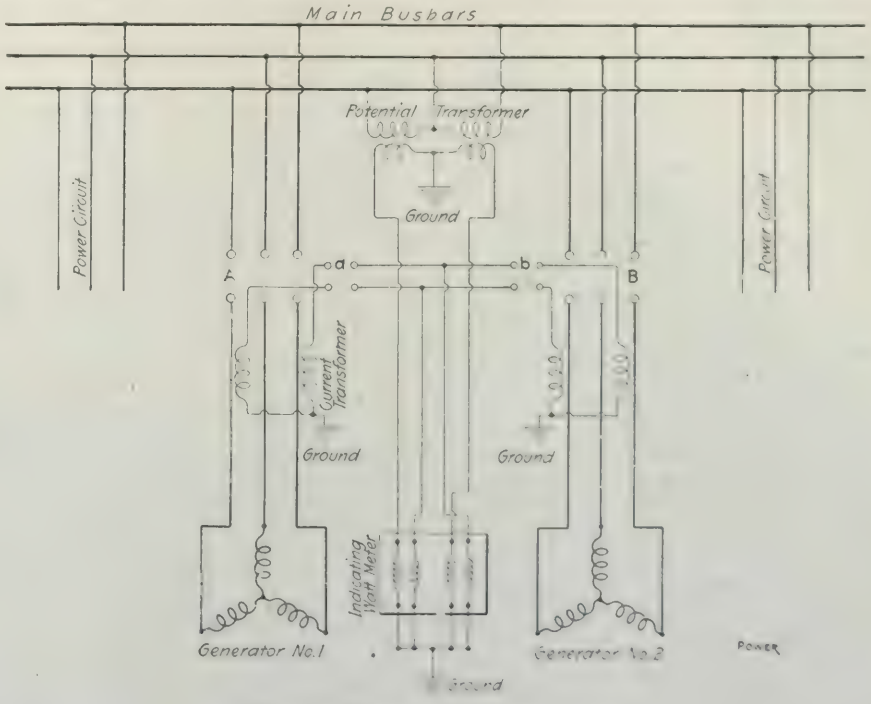
Consider two alternating-current generators connected in parallel through oil switches *A* and *B* to the main busbars. The generators are connected near the middle of the busbars while power circuits are run from each side. It is desired to install a recording wattmeter to make a total output chart of the station load. Owing to the nature of the installation, current transformers in the main busbars will not carry the total power current.

If the wattmeter should be installed as shown, would it give a true indication of the load on both generators? The sketch shows the secondaries of the current transformers on corresponding legs of each generator connected in parallel and supplying the current coils of the instrument. The small switches *a* and *b* in the secondaries of the current transformers are opened and closed automatically with the operation of the oil switches *A* and *B*.

These small switches are for opening the secondary circuits of the current transformers on the generator when not in use, thereby eliminating the shunt path which would exist through the current transformers on the idle machine.

The middle junction point of the transformers and one terminal of the instrument coils are shown grounded. Under what conditions would the instrument give a true indication and under what condition would its accuracy be affected?

League City, Texas. C. L. GREER.



PROPOSED WATTMETER CONNECTIONS

What Made Transformers Heat

Mr. Fisher's comment under the above caption in the July 30 issue is incorrect. Although the secondary voltage is high as stated, there can be no current flowing when the secondary circuit is opened; hence the heating cannot be due to excessive secondary current. Just as soon as this circuit is closed, so that normal current can flow, the secondary voltage falls to its normal value.

Mr. Brindley's transformers became hot upon removal of the meter, because he prevented the flow of current in the transformer secondary; this current being normally in such a direction as to oppose the primary current in setting up flux in the transformer core. The primary current alone caused an excessive flux in the core with large losses in the iron, due to eddy currents and hysteresis, and these heated the transformer.

Another bad effect of the large flux was a comparatively high voltage across the terminals of the secondary winding, which in some cases has given a dangerous shock, or broken down the insulation between turns and ruined the transformer. Workmen accustomed to handling current transformers always short-circuit the secondary circuit before breaking any connections, and thus avoid danger to themselves or the transformers.

York, Penn. H. R. DIETTERICH.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Exhaust Piping

By James H. Beattie

The exhaust pipe of a gas engine is a part of the installation that should be laid out and installed with as great care as is given any other part. Many think it sufficient to run the exhaust to the most convenient point outside the building, irrespective of the size of the pipe employed and the number of turns necessary to get to the desired point. Consequently many an engine has been condemned when the real fault was lack of proper facilities for getting rid of the burned gases from the cylinders.

The burned gases from the average engine are released at a pressure considerably above atmosphere and escape must be provided so that the pressure will drop to that of the atmosphere almost immediately, or the engine will not develop its rated power without consuming excessive fuel. Permitting the exhaust to escape directly to the atmosphere is objectionable on account of the noise, so some sort of a muffler is usually used. Mufflers, if not properly designed, are likely to increase the back pressure on the engine and cause loss of power. Desire to make their machines quiet has led many manufacturers to put on mufflers, which, while very effective in reducing noise, sometimes rob the engine of considerable power.

A medium-size stationary engine had been in use for nine years, operating a deep-well pump. It had received very little attention and had been losing power until it could operate at only partial capacity. The batteries had been changed from time to time and the engine was to all appearances in fair condition. The exhaust pipe ran up through the roof, and a muffler was attached to the end of the pipe. This muffler is shown in section in Fig. 1. It consisted of a spherical shell perforated with two transverse slots. These allowed the gases to escape into the space between this inner shell and the two halves of the muffler walls. These two halves were bolted together, the flanges being serrated so that the gases could escape. These openings had become clogged with carbon until there was very little means of escape for the exhaust. A thorough cleaning of the muffler and grinding the valves put the engine in good condition again.

Another engine was installed by the purchaser and failed to develop the rated power. As a last resort a man was sent from the factory and found that the engine had been connected up with nine elbows in the exhaust line and that the pipe was the same size its entire length, although nearly 100 ft. long. The pipe was rearranged so that four 45-deg. elbows replaced seven of the 90-deg. elbows. It was also shortened nearly one-

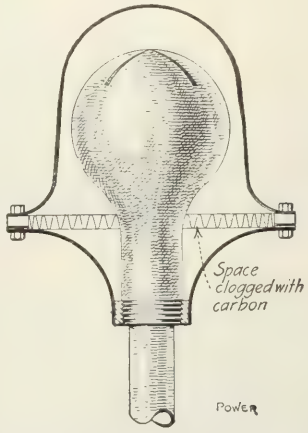


Fig. 1. Section through Muffler

half and where 2-in. pipe had been used in the first place 2½-in. pipe was substituted. The engine developed its full power after the changes were made and used no more fuel than before.

Gas engines, and particularly those of medium size, are sent out equipped with an exhaust vessel, usually of cast iron and with a capacity of three to five times the displacement of the engine cylinder. These are of great service in allowing the exhaust to drop to atmospheric pressure near the exhaust valve, but usually are not very effective as a silencer. One of the most satisfactory mufflers for medium-sized engines is a concrete box buried underground somewhere near the engine room and outside the building. These boxes have a capacity several times that of the engine cylinder, and are as a rule filled with broken rock. This breaks up the gases, yet gives them a chance to escape rapidly and without causing back pressure on the engine. The box is fitted with a manhole on which is placed a light cover, serving as a safety valve in case of an explosion of unburned gases in the muffler. A pipe is run from the box to the atmosphere, arranged so that it will drain out any water.

An installation similar to the foregoing had been installed in connection with a stationary engine and had been satisfactory for the several months it had been in service. The engine, after being started one morning and coming up to speed, suddenly stopped when the muffler cutout was closed, and it was found that the cylinder was filled with water. At first it was supposed that the gasket between the cylinder and the head was leaking, but it proved not to be. By referring to Fig. 2, it will be seen that the muffler is near the building and directly under the overhang of the roof where water could fall directly on the cover. During the night a heavy rain had fallen which had washed obstructions over the end of the outlet pipe near the muffler. Water had accumulated in the muffler and as the engine had a hit-and-miss governor that governed by holding the exhaust valve open, the water had been sucked back into the cylinder the minute the engine had been thrown on the muffler. The water short-circuited the igniter and stopped the engine. In another case the same thing happened, due to water running down the exhaust pipe and filling the exhaust vessel while the engine was idle. An exhaust pipe

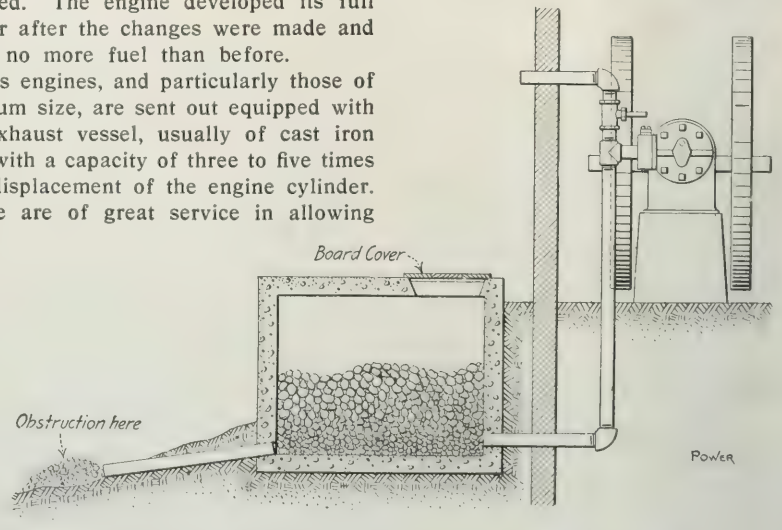


Fig. 2. Section through Muffler, Showing Obstruction

should be protected so that water cannot enter it. Trouble of the kind mentioned is not likely to happen to engines that do not govern by holding the exhaust valve open, but are constant compression engines or are throttle governed.

Many get into trouble through not allowing for expansion in the exhaust pipe. An engine was installed in the basement of a building and the 4-in. exhaust pipe run over to the wall and then up. No exhaust vessel was used and the 10-ft. length of pipe was run so near the wall that the elbow almost touched it when cold. Trouble was experienced in keeping the packing between the exhaust-valve casing and the cylinder tight; the expansion and contraction of the exhaust pipe being the cause of the trouble, which disappeared when the pipe was changed. A swing joint was placed in the pipe, which is perhaps the easiest way to provide for expansion and contraction in an exhaust pipe. Expansion joints cannot well be used on account of the difficulty in keeping the packing in good condition. The pipe is sometimes mounted on rollers where the arrangement is such that one end is free to come and go.

American Practice in Rating Internal Combustion Engines—V*

By T. C. Ulbricht and C. E. Torrence, Jr.

Efficiency Table for American Practice

One of the objects of this investigation was to derive an efficiency table for American practice similar to that derived some years ago by Güldner for German practice. Consequently, Table 4 was developed from the data given in the preceding articles of this series. This table contains the lower heat values, the actual and theoretical air supply, the British thermal units per cubic foot of mixture with different excess coefficients, the fuel consumption per rated brake horsepower-hour, and the thermal efficiency based on the brake horsepower for engines from 5 hp. up, for all the common gases and oil fuels used for power purposes.

The lower heat values per cubic foot and per pound are from the data given in Table 1 of Article III. However, in actual design it is unnecessary to use such exact values; for instance, instead of 589 B.t.u. per cu.ft. for carbureted water gas, 590 or 600 B.t.u. would be a sufficiently close approximation.

The columns on British thermal units per cubic foot of mixture marked with asterisks (*) show the excess coefficient of air generally used for best results. Under air required in cubic feet, actually,

TABLE 4

Fuel	Lower heat value B.t.u.-per Cu.ft	Lower heat value B.t.u.-per Lb.	Air required in cu.ft. Theoretically per Cu.ft. of fuel	Air required in cu.ft. Theoretically per Lb. of fuel	Air required in cu.ft. Actually per Cu.ft. of fuel	Air required in cu.ft. Actually per Lb. of fuel	B.t.u. per cu.ft. of Mixture with Excess air Theoretical air	1.2	1.3	1.4	1.5	Fuel consumption (e), (e′) and (e″) per rated b.hp.-hr. and thermal efficiency (e) 5 b.hp. e	5 b.hp. e′ Cu. Ft.	5 b.hp. e″ Lb.	10 b.hp. e	10 b.hp. e′ Cu. Ft.	10 b.hp. e″ Lb.	25 b.hp. e	25 b.hp. e′ Cu. Ft.	25 b.hp. e″ Lb.	50 b.hp. e	50 b.hp. e′ Cu. Ft.	50 b.hp. e″ Lb.	100 b.hp. e	100 b.hp. e′ Cu. Ft.	100 b.hp. e″ Lb.	Over-100b hp. e	Over-100b hp. e′ Cu. Ft.	Over-100b hp. e″ Lb.	e
Illuminating gas																														
(a) Water gas	304	....	2 63	....	3 0 to 4 0	....	83	8 73	2 68	8 64	8 61	5	14,300	47	....	17 8	13,100	43	....	19 4	11,550	38	....	22 0	10,680	35 2	....	23 8	10,160	33 4	25 0
(b) Carbureted water gas	589	....	6 84	....	7 50 to 12 00	....	75	1 63	9 59	5 55	7 52	3	14,300	24 3	....	17 8	13,100	22 2	....	19 4	11,550	19 6	....	22 0	10,680	18 1	....	23 8	10,160	17 2	25 0
(c) Coal gas	608	....	6 66	....	....	....	86	8 71	2 69	2 64	7 60	8	14,300	23 5	....	17 8	13,100	21 5	....	19 4	11,550	19 0	....	22 0	10,680	17 6	....	23 8	10,160	16 7	25 0
Producer gas					*																									
(a) Anthracite	125 7	....	1 053	....	1 0 to 1 5	....	61	3 55	4 53	.50	8 48	7	15,700	125	....	16 2	14,360	114	....	17 7	11,920	95	....	21 3	10,380	82 6	....	24 3	9,630	76 6	26 5
(b) Bituminous	125	....	1 120	....	0 0 to 1 5	....	59	0 53	4 50	8 48	7 46	6	15,700	126	....	16 2	14,360	115	....	17 7	11,920	95	....	21 3	10,380	83	....	24 3	9,630	77	26 5
(c) Coke	136 3	....	0 985	....	0 0 to 1 5	....	63	7 57	5 55	4 53	1 51	0	15,700	124	....	16 2	14,360	113	....	17 7	11,920	94	....	21 3	10,380	82	....	24 3	9,630	76	26 5
(d) Lignite	134	....	1 145	....	0 0 to 1 5	....	62	7 56	4 53	8 51	5 49	3	15,700	117	....	16 2	14,360	107	....	17 7	11,920	89	....	21 3	10,380	77	....	24 3	9,630	71 6	26 5
(e) Oil	151	....	0 990	....	0 0 to 1 5	....	75	9 69	0 66	3 63	2 60	7	15,700	104	....	16 2	14,360	95	....	17 7	11,920	79	....	21 3	10,380	68 7	....	24 3	9,630	63 6	26 5
(f) Peat	141	....	1 170	....	0 0 to 1 5	....	65	0 58	8 56	3 53	3 51	2	15,700	111	....	16 2	14,360	102	....	17 7	11,920	84	6	21 3	10,380	73 6	....	24 3	9,630	68 5	26 5
(g) Wood	128 7	....	1 076	....	0 0 to 1 5	....	62	2 56	3 53	8 51	5 49	4	15,700	122	....	16 2	14,360	112	....	17 7	11,920	92	7	21 3	10,380	80 6	....	24 3	9,630	74 6	26 5
Blast-furnace gas	95 2	....	0 735	....	1 0 to 1 3	....	54	8 50	5 48	6 46	8 45	2	11,350	119	....	17 8	13,100	26 9	....	22 1	11,150	117	....	22 8	10,480	111	....	23 1	10,500	110	25 8
Coke-oven gas	487	....	4 76	....	7 15	....	84	5 72	1 67	5 63	5 59	8	14,300	29 4	....	17 8	13,100	26 9	....	19 4	11,550	23 8	....	22 0	10,990	22 0	....	23 8	10,160	20 9	25 0
Kerosene	....	18,755	....	189	....	200 to 380	104	8 79	7 74	6 69	....	....	16,300	....	0 87	15 6	15,500	....	0 826	16 4	14,370	....	0 766	17 7	13,900	....	0 714	18 3	13,800	....	....	....
Crude oil (Diesel)	....	18,636	....	187	....	280 to 375	102	8 85	3 78	8 73	1 68	2	10,910	....	0 587	23 3	10,720	....	0 576	23 7	10,330	....	0 555	24 6	9,880	....	0 528	25 8	9,880	....	27 3	....
Gasoline	....	18,500	....	185 5	....	250 to 350	102	8 85	2 78	8 73	2 68	2	15,900	....	0 86	16 0	14,120	....	0 763	18 0	12,520	....	0 677	20 3	....	....	....	....	....	....	....	....
Natural gas (arch)	853	....	8 95	....	6 6 18*	....	85	7 72	7 67	5 63	.59	1	12,900	15 1	....	19 7	12,400	14 5	....	20 5	11,400	13 4	....	22 3	10,380	12 2	....	24 3	9,420	11 0	27 8
(a) High-hydro gen	851	....	8 66	....	6 6 18*	....	86	3 73	1 68	5 63	5 59	5	12,900	15 5	....	19 7	12,400	14 9	....	20 5	11,400	13 7	....	22 3	10,380	12 4	....	24 3	9,420	11 3	27 8
(b) Low-hydro gen	870	....	9 20	....	6 6 18*	....	85	2 72	4 67	1 62	7 58	8	12,900	14 8	....	19 7	12,400	14 1	....	20 5	11,400	13 1	....	22 3	10,380	11 9	....	24 3	9,420	10 8	27 3
Oil gas	542	....	4 94	....	7 15	....	91	2 78	4 73	1 68	4 64	4	14,300	26 4	....	17 8	13,100	24 2	....	19 4	11,550	21 3	....	22 0	10,690	19 7	....	23 8	10,160	18 8	25 0
Fuel oil	....	18,183	....	178	....	250 to 350	102	8 85	2 78	8 73	6 68	2	16,300	....	0 897	15 6	15,300	....	0 832	16 1	14,370	....	0 79	17 7	13,900	....	0 765	18 3	13,800	....	0 758	18 4

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the values representing average practice are also marked with asterisks.

The heat consumption in British thermal units per brake horsepower is found as follows for each case:

$$\frac{2545}{\text{Thermal efficiency}} = \text{B.t.u. per b.hp.-hr.}$$

The consumption of fuel in cubic feet per brake horsepower is found by dividing the British thermal units per brake horsepower by the British thermal units per cubic foot of fuel (lower heat value) as, for instance, for carbureted water gas,

$$\frac{14,300}{589} = 24.3 \text{ cu.ft.}$$

Fuel Consumption Guarantees

The following tabulations show the range in fuel consumption as guaranteed by American manufacturers for their engines with different fuels. In most cases the values are on the safe side, as actual tests show much lower results.

Producer Gas

Full load—
9370 to 13,500 B.t.u. per b.hp.-hr.
15 to 18 cu.ft. of gas per b.hp.-hr.
1.12 to 1.25 lb. of coal as fired.
Three-quarter load—
11,000 to 13,000 B.t.u. per b.hp.-hr.
One-half load—
12,250 to 16,000 B.t.u. per b.hp.-hr.
One-quarter load—
17,000 B.t.u. and upward per b.hp.-hr.

Natural Gas

Full load—
8000 to 15,300 B.t.u. per b.hp.-hr.
10 to 18 cu.ft. of gas per b.hp.-hr.
Three-quarter load—
10,700 to 12,000 B.t.u. per b.hp.-hr.
One-half load—
12,250 to 16,000 B.t.u. per b.hp.-hr.
One-quarter load—
17,000 B.t.u. per b.hp.-hr. and upward.

Illuminating Gas

Full load—
10,000 to 13,800 B.t.u. per b.hp.-hr.
15 to 20 cu.ft. of gas per b.hp.-hr.
Three-quarter load—
11,000 to 12,000 B.t.u. per b.hp.-hr.
One-half load—
13,000 to 16,000 B.t.u. per b.hp.-hr.

Blast Furnace Gas

Full load—
10,500 B.t.u. per b.hp.-hr.
Three-quarter load—
11,500 B.t.u. per b.hp.-hr.
One-half load—
13,600 B.t.u. per b.hp.-hr.

Kerosene

Full load—
13,240 to 16,150 B.t.u. per b.hp.-hr.
0.1 gal., or 0.75 to 0.84 lb. per b.hp.-hr. for small engines.
0.56 to 0.65 lb., or 0.725 to 0.901 pints per b.hp.-hr. for large engines.

Gasoline

Full load—
10,820 to 15,500 B.t.u. per b.hp.-hr.
⅛ to ⅒ gal. per b.hp.-hr.
0.586 to 0.968 lb. per b.hp.-hr.
0.8 to 1.1 pints per b.hp.-hr.

Fuel Oils

Full load—
8720 to 13,320 B.t.u. per b.hp.-hr.
0.1 to 0.128 gal. per b.hp.-hr.
0.393 to 0.74 lb. per b.hp.-hr.
1 pint (average) per b.hp.-hr.

Fuel Oils for American Diesel Engines

Full load—
9029 to 11,200 B.t.u. per b.hp.-hr.
0.0608 to 0.0784 gal. per b.hp.-hr.
0.447 to 0.588 lb. per b.hp.-hr.

A Jacket Water Trouble and Its Cure

By A. D. Williams

A newly installed 300-hp. gas engine gave considerable trouble. The cylinders were vertical and the jacket water entered the bottom of the jackets, then passed up to the cylinder head, thence to the exhaust valves and the jacket surrounding the connection to the exhaust manifold. Some of the water passed to the exhaust gas spray and the rest returned to a sump, from which it was pumped to a cooling tower. This method of circulation brought the coolest water around the lower part of the cylinder, and though the cylinder had been bored with a slight taper to compensate for contraction at this point, it was found that the piston had some tendency to seize at the bottom of its stroke, and required excessive lubrication.

The engine had a fairly constant load and was operated 24 hours a day, seven days a week. The continuous operation gave ample time for the engine to get heated up all over and no shutdown periods allowed opportunity to cool off. It was found, however, that the machine could not be held to the continuous schedule, but had to be shut down frequently to provide cooling periods.

One of the troubles was a tendency of the cylinder-head jackets to become vapor bound; hence it was found necessary to vent them. The exhaust outlets and the jacket water got exceedingly hot. The upper part of the cylinder and the top head became so hot that to touch them burnt the hand. Some slight exhaust-valve troubles, due to overheating, were also experienced.

The engineer in charge suggested that reversing the cooling-water circulation would reduce the trouble. The builders of the engine were dubious, but agreed to make the change without charge provided the plant would pay the cost of restoring the old system if the one suggested proved inoperative. The change consisted in removing the waste-water manifold from below the exhaust outlets and connecting the cooling-water supply at this point; while the old cooling-water connections to the bottom of the cylinder jackets were replaced by a waste-water manifold. The waste line from this manifold was run with an inverted "U" in it and connected to a pipe leading to the cooling-water sump. This inverted "U" was made high enough to maintain about 1½ in. of water on top of the upper cylinder head and a vacuum breaker was placed at its highest point to prevent this water siphoning out when the supply was cut off.

With the new system of circulation the cool water becomes warmed in the exhaust-outlet jackets and causes a large drop in the temperature of the exhaust gases with a reduction in the amount of water required to be injected into the exhaust manifold. The water on the top of the cylinder head and in the cylinder jackets has a gradually increasing temperature, and there are no marked differences in temperature, such as occurred when the cool water entered the cylinder jacket. The consumption of cylinder oil is considerably reduced and the piston no longer tends to seize. Runs of several weeks' duration are now frequently made with this engine.

CORRESPONDENCE

Piston Rings Caused Trouble

Our 15x14-in., three-cylinder, vertical, producer-gas engine had been giving trouble by the governor hunting and jumping at all loads. The engine had been in use about two years, running on an average of 6 hr. a night and during this time the pistons had not been removed; consequently the rings were in bad condition, causing a loss of compression as the indicator diagrams showed. At first we did not connect this with the actions of the governor, but after going all over the governor bearings, taking up all lost motion, setting the valves and timing the ignition, without any relief, we decided to remove the pistons, taking one out each night after shutting down. A slight improvement was noticed after the first piston had been cleaned and replaced, which encouraged the work of removing the others, and, after they had all been overhauled, the engine settled down to its usual satisfactory performance.

This happened about 18 months ago, and recently the engine began to act in the same manner, namely, loss of compression coincident with the governor hunting. Therefore we again removed the pistons, cleaned the rings, putting in new rings for those that appeared worn, and now the engine is again running as it should and carries its full load.

J. W. Fries.

Middleboro, Mass.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Combined Central Heating and Electric Plants*

By Edwin D. Dreyfus

Operating a heating plant singly is only attractive under favorable circumstances. Electric stations, operating independently, are successful in most cases; but by effectively combining the two, very fair returns should accrue. This paper aims to present some new developments in prime movers, more particularly in the steam-turbine field, whereby improvement in efficiency of the consolidated electric steam-heating plant may be effected.

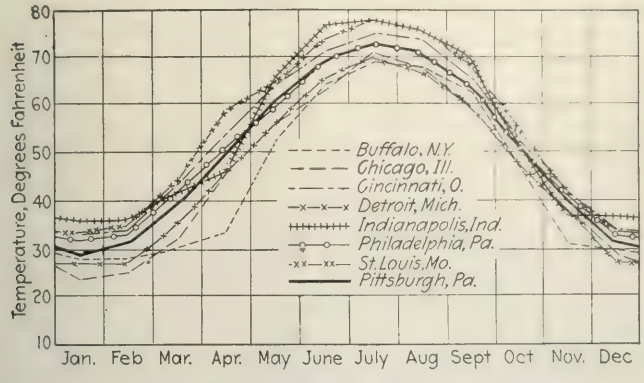


Fig. 1. Monthly Temperature Curves

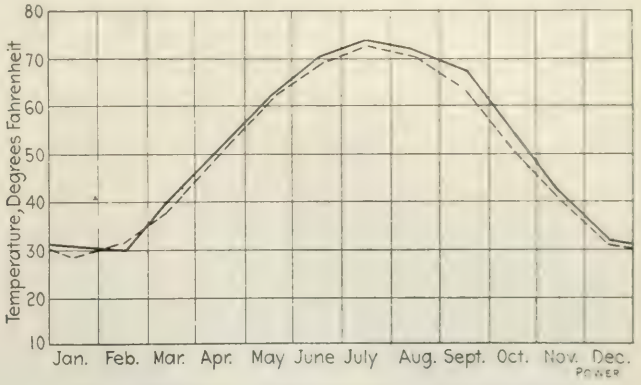


Fig. 2. Monthly Temperatures at Pittsburgh

The paper is concerned principally with heating requirements, a study of the seasonal, daily and hourly variations in temperature in some typical latitude where a heating plant may be installed is important. The monthly temperature for the year from the U. S. Signal Service and Blodget's Climatology for eight first-class cities of this country are reproduced in Fig. 1. Pittsburgh's climatic conditions, excepting possibly a small difference of humidity, proves to be a fair average between latitudes 38 and 43 deg., which embrace the territory investigated. This is shown more clearly in Fig. 2, where the average of all these curves is shown by the dotted line, while a 20-year average for Pittsburgh is represented by the heavy line. The analysis of these conditions is not complete. Effects of not only daily changes, but the hourly variations as well, must be properly examined. Hence the records of the U. S. Weather Bureau at Pittsburgh have been consulted, and 20-year averages have been plotted in Fig. 3.

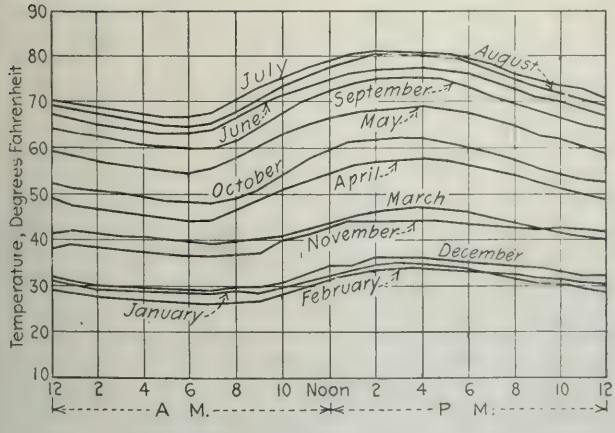


Fig. 3. Hourly Atmospheric Temperatures at Pittsburgh

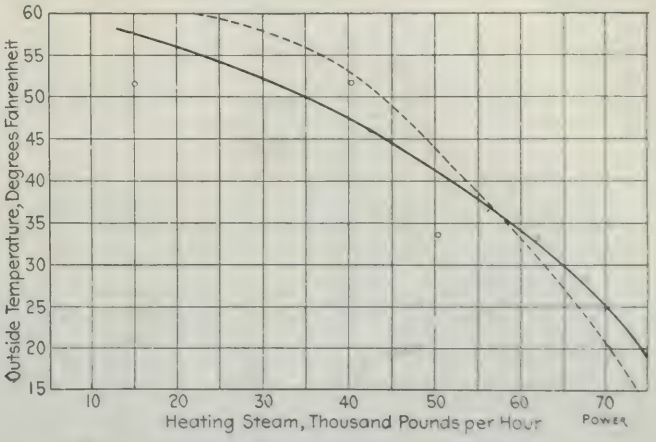


Fig. 4. Outside Temperatures and Steam Requirements

Temperature Factors

As the paper is concerned principally with heating requirements, a study of the seasonal, daily and hourly variations in temperature in some typical latitude where a heating plant may be installed is important. The monthly temperature for the year from the U. S. Signal Service and Blodget's Climatology for eight first-class cities of this country are reproduced in Fig. 1. Pittsburgh's climatic conditions, excepting possibly a small difference of humidity, proves to be a fair average between latitudes 38 and 43 deg., which embrace the territory investigated. This is shown more clearly in Fig. 2, where the average of all these curves is shown by the dotted line, while a 20-year average for Pittsburgh is represented by the heavy line. The analysis of these conditions is not complete. Effects of not only daily changes, but the hourly variations as well, must be properly examined. Hence the records of the U. S. Weather Bureau at Pittsburgh have been consulted, and 20-year averages have been plotted in Fig. 3.

Steam Consumption

As the exhaust-steam demand through the year will depend upon the extent and character of the heating system, it is unsafe to generalize. A specific example should be followed, and therefore an assumed station has been laid out to contain three 1000-kw. automatic bleeder turbines and two 1000-kw. standard condensing turbines. This station is intended to serve a maximum average heating demand of 65,000 lb. of steam per hour, and an electric peak of 4600 kw. These demands occur during different months of the year, the high electrical peak in December, and the greatest heating consumption in February. The ratio of the maximum kilowatts to heating steam demand of the plant is 1 to 14.2. The conditions apply approximately to a city of about 80,000 to 100,000 population.

The hourly steam consumption for different outside temperatures (standard inside room temperature of 70 deg. F. understood) is arrived at empirically from available data, as given in Fig. 4. The dotted curve is derived from a curve given by F. C. Chambers in the proceed-

*Abstract of a paper read before the National District Heating Association at Detroit.

ings of the National District Heating Association for 1911. The circles represent points given by a 6000-kw. central station in Illinois. The full line represents an average curve assumed for the accompanying calculations.

As the outside temperature decreases and the steam consumption in the heating system changes correspondingly, the pressure at delivery from the power house must be raised to overcome the friction losses and effect of condensation which increases as the demand becomes greater. A schedule of pressure is usually es-

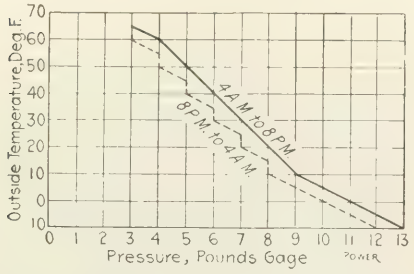


FIG. 5. TEMPERATURE-PRESSURE CHART FOR PARTICULAR SYSTEM

tablished, and the curves in Fig. 5 will indicate suitably the proportionate pressures that should exist at the outgoing feeders from the power house for different outside temperatures. These figures are somewhat arbitrary, as obviously the absolute pressures in any case will

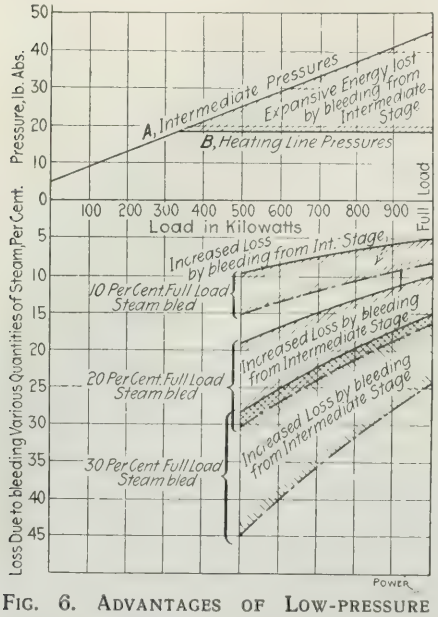


FIG. 6. ADVANTAGES OF LOW-PRESSURE OVER INTERMEDIATE-PRESSURE BLEEDING

depend upon the size of mains used and their length or lineal equivalent.

TYPES OF PRIME MOVER

Where the heating demand was a large factor, reciprocating engines have been used principally. They have been simple or compound noncondensing, compound condensing with provision for bleeding from the receiver, or else the plant has been a combination of these types to produce the best results for existing conditions. Compound reciprocating engines with moderate cylinder ratios, have been often selected so that the engine may run alternately condensing or noncondensing as the demands may dictate, but Fig. 7 makes plain that a large loss of economy will occur during the nonheating seasons, as well as during times of light heat demand. Noncondensing steam turbines have been installed in a few cases, as well as condensing steam turbines where steam has been bled from the intermediate stage.

Within the last two years a special turbine has been devised in which the steam not bled is automatically diverted to the low-pressure section. The earlier types of engines and turbines are already well understood. The construction of the new Parsons' machine closely resembles a standard condensing turbine, the main departure being locating a diaphragm between the intermediate- and low-pressure stages; the flow of steam is regulated by a gravity controlled valve.

In the summer months when no exhaust steam is demanded the bleeder valve may be lifted entirely out of the zone of action. In some installations bleeding all the steam from the turbine for extended periods may be desirable. If the turbine is fairly well loaded, as it would probably be under these circumstances, having the steam pass through the entire turbine would prove more economical if none escapes to the atmosphere. This would be accomplished by piping arranged so that the condenser could be cut out and the entire exhaust from the turbine directed into the heating system.

At times of brief heavy peaks in the electric load, a special device attached to the turbine can be used to automatically disconnect it from the heating system, allowing it to operate strictly condensing and thereby develop its maximum power.

BLEEDING AT LOW AND INTERMEDIATE PRESSURES

The distribution of work throughout a turbine is divided roughly as two-tenths in the high-pressure, three-tenths in the intermediate-, and five-tenths in the low-pressure section. This shows the significance of expanding the steam through the intermediate section before utilizing it in a heating system. The advantage derived is shown graphically in Fig. 6, the upper portion of which indicates the pressure variation at the entrance of the intermediate section. With the bleeder connection at this point, considerable throttling losses occur, as will be appreciated from lines *A* and *B* representing the pressure at the latter section and the heating system respectively.

For every pound of steam bled, expansive energy will be lost correspond-

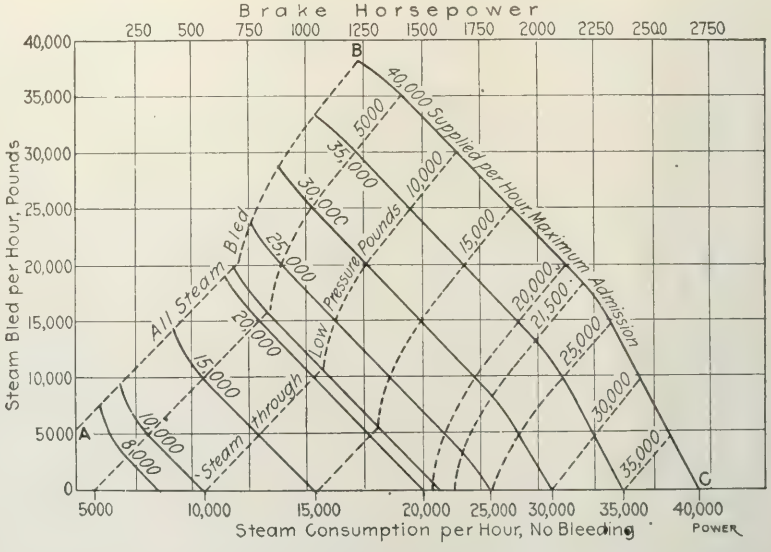


FIG. 7. RELATION OF STEAM DISTRIBUTION AND POWER LOAD WITH VARIABLE BLEEDING DEMAND

ing to the pressure drop from the intermediate section to the exhaust. Expanding the steam throughout this section before bleeding will correspondingly reduce the energy loss, the amount being equivalent to the additional work performed. The large drop in pressure occurring in the high-pressure stage does not indicate that this section is doing a proportionate share of the work as the heat drop through high-pressure ranges is far less than through the same range at lower pressure.

The lower set of curves, Fig. 6, compares the loss in capacity which would be caused by the two bleeding systems for varying percentages of steam bled, assuming an equal amount of steam passing through each turbine. The quantity of heat per pound theoretically lost by cutting short the expansion at the intermediate pressures determined by the

pressure-load line *A*, has been computed and expressed on a percentage basis for different amounts of steam bled. Values thus obtained are shown by dotted lines, the solid lines representing losses that occur with the improved method of bleeding from the low-pressure section. Evidently, therefore, the vertical distance between curves corresponding to equal bleeding demand, closely approximates the saving affected by a bleeding arrangement utilizing steam at low, rather than at intermediate pressure. Besides being more economical, low-pressure bleeding permits operating the heating system with the overload valve of the turbine open.

Characteristics of Automatic Bleeder Turbine

As the performance of the automatic bleeder must be known to appreciate the work in the continuation of the study or examination of the specific case undertaken, the characteristics of such a turbine are illustrated and discussed at some length. The triangular chart, Fig. 7, enables a ready comprehension of the relation between the steam bled, steam

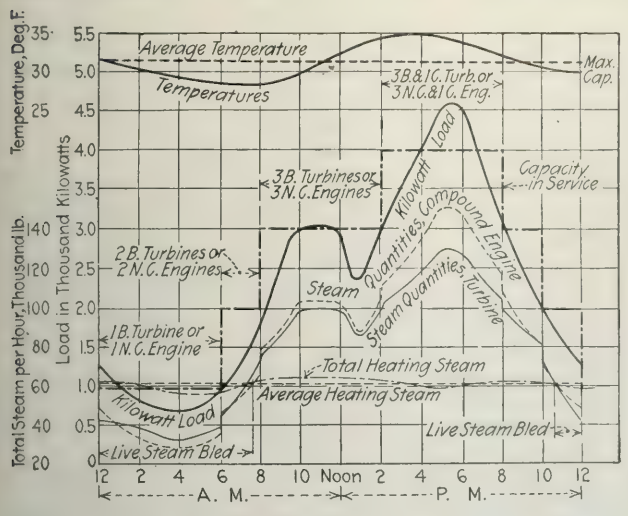


Fig. 8. Mean January Demand—Load Factor 40.4 per Cent.; Total Kw.-hr., 1,380,000

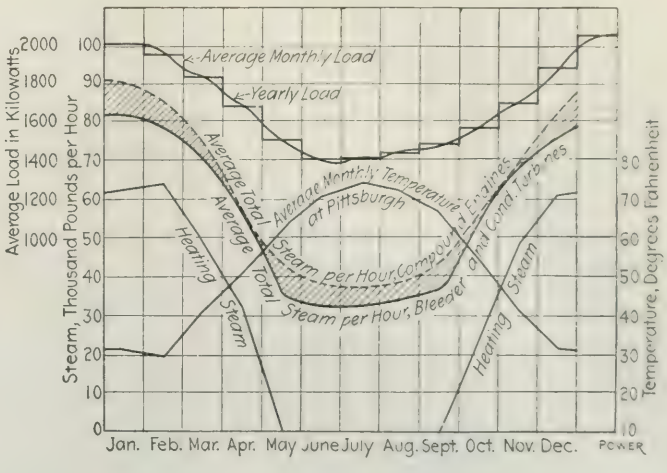

Average Load in Kilowatts
Steam, Thousand Pounds per Hour
Temperature, Degrees Fahrenheit
Average Monthly Load
Yearly Load
Average Total Steam per Hour, Compound Engines
Average Monthly Temperature at Pittsburgh
Average Total Steam per Hour, Bleeder and Cond. Turbines
Heating Steam
Heating Steam
Jan. Feb. Mar. Apr. May June July Aug. Sept. Oct. Nov. Dec.
POWER

Fig. 9. Summary Curves

supplied to the turbine (that is, draft on the boiler) and the quantity passed to the condenser for a given power load sustained by the unit. These curves have been plotted against brake horsepower on the horizontal, and weight of steam on the vertical scale. The solid lines extending up and to the left, are lines of constant steam supply and show the increase of available power with decreased bleeding. As will be observed, this is a straight line relation within certain ranges, the power output increasing at a constant rate as the amount passing through the low-pressure stages increases, as shown by the dotted lines which extend up and to the right. As the latter reach a certain value, i.e., about 21,500 lb., the slope of both the solid and dotted lines becomes steeper and correspondingly the rate of increase of power output is diminished. This is due to the secondary overload valve opening, as the turbine is designed for a normal bleeding demand and the low-pressure stages are accordingly proportioned to pass a certain quantity of steam and, therefore, when this is exceeded, the turbine output while still increasing, does not increase at so fast a rate.

To complete the exposition, a supplementary scale has been added besides that of the brake horsepower which gives at a glance the total steam consumption of the unit operated strictly condensing (a limiting condition). The other two interesting limits are lines *AB* and *BC*. The first gives total steam with no work developed in the low-pressure cylinder. Line *BC* establishes a boundary for maximum power which may be developed when a predetermined amount of steam is bled. When all steam is bled, as shown by the limiting line *AB*, the total amount supplied is slightly greater than that accounted for by the quantity bled. On first thought these should be equal, but owing to the construction employed, a little steam leaks past the packing into the low-pressure section, which explains the apparent discrepancy.

The curves shown in Fig. 7 are for one set of steam conditions only, 150 lb. initial pressure, dry and saturated, 5-lb. gage bleeding pressure and 28-in. (30 in. bar.) vacuum at the turbine exhaust. These particular curves apply to the turbine intended where the steam would always be bled in large quantities, and, therefore, contains only a relatively small low-pressure wheel. Where conditions require it, the automatic bleeder turbine may be made equally as efficient in operation as the standard designs of complete expansion turbines.

Working Results

With the character of the steam and heating loads laid down, and the performance of the different units ascertained, the results for each progressive month may be developed. In deriving the final curves, Fig. 9, an average curve sheet for each month was prepared, such as the one shown in Fig. 8. To keep the related curves together, the average hourly temperatures were transferred from Fig. 3 and the steam required for heating during the day, from Fig. 4. The average hourly steam required is obtained from Fig. 4, and the variations occurring from hour to hour are interpolated from the previous data. The curves of the average day of the month have been made, in the twelve conditions introduced, to correspond with the typical monthly difference throughout the year.

The total capacity of the units in service is shown by the stepped-lines. The units in operation, whether engine or turbine equipment, are scheduled on these average daily sheets; this was planned so that no hasty cutting in or out of different types would be necessary if the load swung widely from the normal. Furthermore, ordinarily the load surges appreciably so that the curve of output is quite irregular instead of smooth as represented. The use of the smooth curve is permissible, however, to facilitate working out the final quantities, if proper regard of the deviations is taken when connecting or disconnecting the units from the busbars.

Taking the loads and the steam demands and referring back to the original data, the various points on the final steam-consumption curve were determined.

The monthly curve sheets show that for a few hours in the early morning the power load is so light that live steam must be bled to the heating main. This live steam is taken as part of the plant consumption, and is included in the total steam curves in Fig. 9.

According to the curve in Fig. 4 no heating steam would be demanded when the outside temperature reached about

60 deg., and since the average monthly temperatures of May and September are more than 60 deg., no heating requirement is shown for these months. Evidently, however, during part of these months some steam would be required, and hence the value of the total steam for these two months has been increased slightly when transferred to Fig. 9. This final curve sheet, shows graphically that the automatic bleeder turbine plant follows the heating-steam requirement curve more closely than the compound-engine plant, indicating about 10.6 per cent. less steam used per year by the bleeder turbine equipment than by the piston-engine plant. It is not claimed that this comparison is by any means absolute, but practical operation will undoubtedly bear it out as a general proposition.

CORRESPONDENCE

Mr. Jahnke's Heating Coil

It is difficult to understand how the 150-ft. wall coil, shown by H. A. Jahnke in the June 11 issue (Fig. 1), even remained on the wall plates when steam was turned on unless the pipe header shown at the right was so arranged that it would allow for expansion. A pipe coil of the length stated will expand a considerable distance, and, as in most long coils the upper pipes will heat up before the lower ones, ample provision should be made to allow each pipe to expand independently of the others; otherwise the cast-iron headers will likely be broken by the unequal expansion.

The pipes may also buckle so much that the coil will lift in parts from the hook plates, or if inclosed hangers are used, will probably be torn from the wall supports.

The plan shown in Fig. 2 is good only for a coil about 200 ft. long. The turn shown at one end should be at least 3 ft. long at the short side of the turned pipes. If there is not room overhead for a long turn, it can be made around a corner of the room.

Some steam fitters claim that the turn around a corner has the advantage over the upper turn, for, in a long coil, as in Fig. 2, when steam is turned on, the expansion may increase the length of the pipes several inches. If little steam is admitted the upper pipes will heat first and, as they lengthen, the elbow of one pipe will reach the next one to it and, as the last pipe mentioned may be cold and rigid, the pressure of the hot pipe against it may be great enough to crack the elbow.

In a coil 200 ft. long it is good practice to have a turn at each end. At the feed end there should be an upper turn, as shown in Fig. 2, and at the drip end, a corner, or down turn. This would allow the pipes to move both ways and the strain upon them and the headers would be greatly decreased as compared with that shown in Fig. 2 of Mr. Jahnke's letter.

JAMES E. NOBLE.

London Junction, Ont.

Mr. Jahnke's Fig. 1 in the June 11 number is surely a bad installation. Any heating coil installed should be mitered either on the supply or return end and better still on both ends when the coil is 100 ft. long or over. I do not think two steam manifolds necessary for the system in question, but would suggest that at least one end be mitered and that steam be introduced at each end of the supply manifold.

A short time ago I installed a heating system in which one coil had nine 1¼-in. pipes 225 ft. long. The supply manifold received steam at each end and the return end was mitered. The results from the system are entirely satisfactory.

R. G. Cox.

Harrisburg, Penn.

From my experience I think the proposed coil shown in Fig. 2 is good practice, considering its location and use. There should, however, be swinging, or swivel connections and valves at both inlet and outlet to the coil. This swinging connection alleviates any strain that might be brought upon the coil by the expansion and contraction of the supply and return pipes.

M. C. ST. JOHN.

Houston, Tex.

Mechanics of Heating and Ventilating

In the review of my book on "Mechanics of Heating and Ventilating," which appeared in the June 11 issue, it is claimed that nothing is said about arranging the piping to balance distance as well as the friction heads. The distance is taken into account in all problems. While not always specially mentioned, the ways and means of allowing for length are really self-evident. The example for forced hot water shows a case with distances of about 1500 ft. each way.

Regarding the relation of the static head with reference to the height of the expansion tank, I think that when the tank connections are properly placed and sized, and the suction head is kept well below the static head, there will be no trouble with an open tank.

The apparent errors quoted for the differential heads 70.8 and 83.8 arise from the fact that the last decimals of the densities from which they were figured had been omitted in the test. In any event, the discrepancies are within the variations of different data on the properties of water. They would make less than 1 per cent. difference on the differential head and be of no consequence at all in equalizing a system.

My statement on omitting reducing valves was given more as a theoretical possibility, and refers only to the transmission of live steam at high-pressure drop, not to exhaust heating.

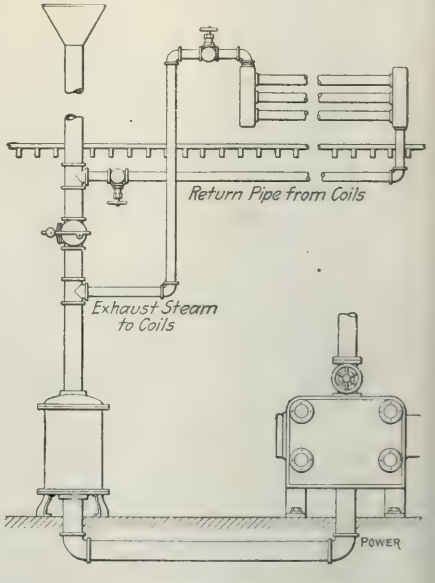
The pressure drop of 6 to 12 lb. per sq.ft. will give slightly larger mains than are used by most engineers, but this in my opinion is desirable, all things considered. If equilization is carried fairly close, there is no reason why a larger drop cannot be assumed. It is all a matter of judgment, depending on the situation.

K. MEIER.

New York City.

Expert Piping

The accompanying sketch shows how the condensation from a back-pressure heating system is to be returned to the exhaust pipe, above the back-pressure valve, unless the engineer, in charge of the plant, succeeds in presenting a more convincing argument than a plain statement of facts and principles, to the "expert" installing the system, and the owner blindly allowing the expert full authority.



NOVEL CONNECTION OF HEATING COIL

So positive is the owner that the exhaust will merely be bypassed through several hundred feet of heating coils and return to the exhaust pipe without any change in its condition, that, should he discover the engineer seeking to enlist outside interference, he would possibly discharge him. Before risking his reputation in operating such a system, the engineer would like the opinion of readers on the proposed piping, and what course they would suggest to convince the owner of the fallacy of the idea.

R. ESSINGER.

Nahant, Mass.

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POWER

Purchased Current For Summer Service

Elsewhere in this issue we print the arguments of W. A. Blonck, of Chicago, who endeavors to convince the office-building owner that it would be cheaper to shut down his plant during the summer months and purchase electrical energy from the central station. He at the same time concedes that during the heating season the operation of the isolated plant is the more economical.

If such a course were feasible we should gladly commend it; for not only are we solicitous as to the welfare of the engineer, but we also want to see the owner run his plant as cheaply as possible consistent with efficiency and good service. Mr. Blonck's arguments, however, although attractive at first glance, prove very superficial upon analysis.

In the first place, if the plant were run on central-station service during the summer months, it would necessitate electric elevators and all motor-driven pumps and auxiliaries. Doubtless very few office buildings are thus equipped and to change over would incur considerable expense. Also with all electric-driven auxiliaries a profitable source of exhaust steam for feed-water heating during the winter months would be removed, and this exhaust steam would have to be taken from the main engines at the expense of the heating system. Moreover, many buildings are fitted with absorption refrigerating systems which require exhaust steam during the summer.

As for the labor, it would be poor policy for any management to break up the personnel of its operating force during the summer season. This would tend to less efficient service and consequently higher cost of operation during the heating season. On the other hand, if left intact it would mean a large labor item which must be charged against the summer service. Furthermore, not only must the labor be thus charged but the interest, depreciation, taxes, insurance, etc., on the idle equipment must be likewise charged against the purchased current. It is doubtful if current could be purchased so cheaply, under a short-term contract, with these additional charges as to make the total cost of summer service less than when operating the isolated plant.

Turning now to the central-station viewpoint, it is questionable whether any central station would care to enter extensively into this class of service without charging a rate commensurate with the conditions, unless on the chance that having once obtained the customer for a five months' contract, and thus secured a foothold, it would be easy to induce the owner to use central-station service altogether.

Two Kinds of Engineers

Assurance, self-confidence and a general belief in his ability and knowledge to do things are strong assets to an engineer, when they are justified. Such men, if not overburdened with egotism, make valuable acquaintances, especially if disposed to impart their knowledge to others.

Not uncommonly, a man well informed on some particular subject more or less interesting to other engineers, refuses to impart any of his information, taking the stand that, as he had to dig for what he knows, others can do the same. This is a questionable policy. A selfish man seldom accomplishes much or attains to high regard.

At the other extreme is the engineer who thinks he knows much more than he really does and is prodigal with his advice. When an engineer reaches the stage that he cannot admit anyone is right who differs with him, disputes everything that does not coincide with his experience, however limited that may be, and insists on his opinions against all argument or reasoning, he becomes an unmitigated bore.

Fortunately, few engineers belong to either of these classes. Most of them are glad to share the benefit of any information they possess and have respect for

the views held by others. Both attitudes are evidences of broad-mindedness, and that is one of the characteristics of the truly great.

Massachusetts Boiler Law

That the provision made for considering changes in the Massachusetts Boiler Rules twice annually, was wise cannot be gainsaid. Advantage has been taken of this provision a number of times, and it has relieved many situations that were embarrassing to both the boiler manufacturers and those interested in the just enforcement of the rules. It appears from recent developments that some changes in the law itself are advisable.

The recent resignation of the chief inspector of the department developed the fact that no provision had been made for such a contingency and a successor could not be appointed; but this was a matter that affected the department alone. A more serious condition has been recently developed which affects the right of an inspector to earn his living. A company writing boiler insurance in the State of Massachusetts, recently dropped that line and had its business taken over by another company, which necessitated disbanding its boiler-inspection force. The company going out of business had in its employ an inspector who only a very short time before had passed an examination and secured a certificate of competency. This inspector was now out of a job, but having received the stamp of approval of the Massachusetts authorities, his services were naturally in demand, and another company engaged in the same line of business employed him.

The Massachusetts law provides that an inspector's certificate is automatically revoked when he ceases to be employed by the company that employed him at the time of making application for the certificate; so this inspector had no certificate of competency, but there appeared to be no reason why a new one could not be obtained. However, when application was made, the information was received that ninety days must elapse between two examinations to comply with the law. This is certainly a great injustice to a man who earns his living by inspection work, and we believe it will be found unconstitutional to deprive a man of his rights in such manner. Why should not an inspector holding a certificate of competency have it transferred without further examination if he should desire to become employed by a different insurance company?

What chance has an inspector in Massachusetts to better his condition? If he seeks a more remunerative position with another company he must first forfeit the certificate of competency which he holds and stand another examination. If he should fail he would be without means of obtaining work as an inspector at all, and could not even return to his former position. Such conditions as these need prompt attention if the Massachusetts Boiler Laws are to continue their successful start.

Excess Air Indicated by Superheat

With a correctly designed furnace and proper firing, complete combustion depends almost entirely upon the control of the air supply. This is now recognized as fundamental. When using oil fuel this regulation, for obvious reasons, is much easier than when burning coal. In either case the temperature and composition of the flue gases are usually depended upon to indicate whether too much or too little air is admitted.

We understand, however, that during some tests on one of the large oil-burning plants on the Pacific Coast an entirely different method was employed to ascertain the proper amount of air. It was discovered during the preliminary tests that too much excess air was invariably accompanied by an increase in the superheat, whereas an inadequate supply was indicated by a decrease in the degrees of superheat. While this puzzled the engineers at first it was finally explained as follows:

The boilers were of the B. & W. type and with an excess of air the furnace temperature was lowered and consequently less heat was absorbed by the first pass. This being true the temperature of the gases passing the superheater were higher and more heat was absorbed by the superheater.

This means of indicating the proper air supply was found to work so satisfactorily that it was depended upon to a large extent throughout the tests. While local conditions may have favored this way of indicating the air supply, still it is reasonable to assume that it could be employed in other cases. It is simple and its general reliability is worth investigating.

Precautions against Damage from Tube Failure

As has been stated many times in these columns, the only legitimate excuse for boiler laws and the licensing of engineers is the duty which devolves upon the government to adopt and enforce such safeguards as may reasonably be expected to protect the public or those concerned with the operation of steam plants.

One of the simpler devices that affords considerable protection to the attendant, where a water-tube boiler is used, but which is rarely adopted, is the inward opening fire-door. Tube failures in water-tube boilers are of daily occurrence, and while it behooves the insurance interests as well as municipal and state authorities to investigate and see if the number of such failures cannot be reduced, there is little likelihood of the problem being thoroughly solved in the near future; in the meantime life, limb and property are being sacrificed in such accidents.

Inward opening fire-doors will not prevent serious or fatal results in all cases of tube failures but are likely to in the majority of such accidents.

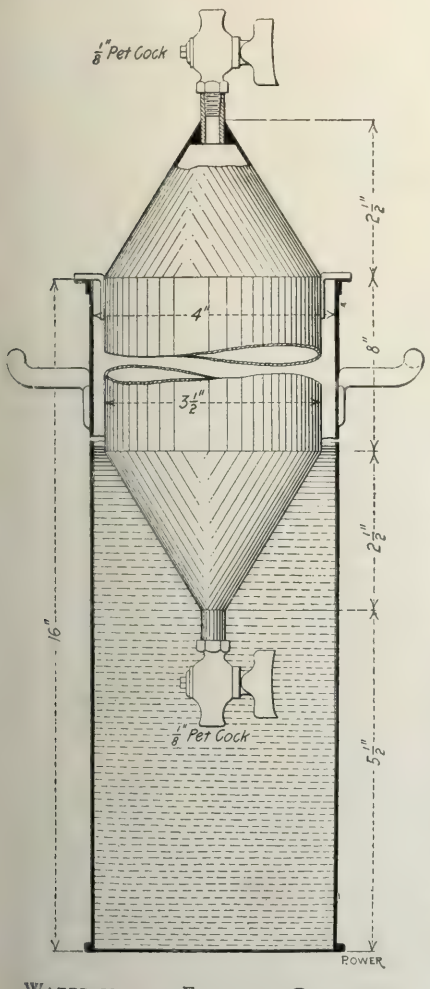
Not often is the brickwork disturbed by these accidents; but the fire-doors are usually blown open, and if the fireman happens to be in his accustomed place, directly in front of the boiler, he is either burned by flying coals or escaping steam, or inhales flame, and this nearly always is fatal. POWER is constantly striving to be of benefit to the power-plant operative, and we would advise every engineer employed where water-tube boilers are used to endeavor to persuade the owner to equip them with doors which will not fly open when a tube bursts. Every fireman, because he is the one most vitally interested, should insist that his engineer urge this safeguard. POWER advises adopting such rules as are necessary to procure these devices for use in water-tube boilers, in states or municipalities having enacted or contemplating the enactment of laws governing the operation of boilers.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Water-Sealed Flue-Gas Collector

The flue-gas sampler illustrated is made of light galvanized iron riveted together, soldered at the joints, and may be of any convenient size. The handles on the side of the open cylinder allow hanging it up if it cannot be set down.



WATER-SEALED FLUE-GAS COLLECTOR

If the closed cylinder is placed in the open one, as shown, and the water run out, it will cover the lower cock, acting as a water seal preventing air from entering the collector if it is not removed instantly. If a sample over several hours is desired (three hours being the practical limit for the size shown), the cocks should be adjusted at intervals, for when the water flows out, the pressure is reduced, decreasing the amount of flow, thereby sucking in more gas during the first periods than the last.

If the collector is connected to the flue by a short piece of rubber tubing, it can be easily removed and carried to wherever the Orsat apparatus is situated.

C. L. BARNUM.

West New Brighton, N. Y.

State License Laws

I have read with interest the articles in POWER from time to time relative to state engineers' license laws. I hold an engineer's license and believe that every engineer should be required to pass an examination based on years of practical experience as well as technical education before he is intrusted with the responsibilities of operating a plant, the number of years experience varying according to the class of license he procures. However, I do not believe an engineers' license law is needed, particularly in this state, as badly as a state boiler-inspection law. This has been brought more forcibly to my mind since I took charge of the plant I am now operating.

To save the expense of a boiler plant and also of rehandling coal, a local plant erected its engine room near a coal mine and receives steam from the boilers that operate the mine. The coal company is paid for the steam used. In the mine boiler house are four 72-in. and two 60-in. boilers, 16 ft. and 18 ft. long, respectively, set up in large sandstone rocks and lined inside with ordinary red brick; all are in bad shape. They are all half-flush front boilers with the iron fronts cracked in several places, and most of the brick lining is gone.

The management refuses to use glass water gages and for gage-cocks ordinary pet-cocks and ½-in. globe valves are used which either fail to work at all, or blow all the time. Each boiler is fitted with a steam gage, but no two register alike and none are correct. The flues are so scaled inside and out, and so corroded that every few days some need rolling. This has been done so often that the flues in some of the boilers are almost as thin as paper.

Most engineers agree that a boiler-inspection law would help them in their work. As I have to be on duty most of the day, I am very thankful that the engine room is some distance from the boiler room. It seems to me, while the states are passing other laws to protect life and property, it would be well to condemn the old lever safety valve, and require a good reliable spring-loaded pop valve having means to lock it after being set.

From my experience, as well as from articles published in POWER, it appears that ignorant firemen continue to hang heavy weights, etc., on the lever which prevent the valve from working. In a little plant a few years ago where the boilers were fitted with lever safety valves the working pressure was 100 lb., but the steam gage in the engine room registered 130 lb. and so did those on the boilers. After questioning the fireman I climbed on top of the boiler and found a piece of 2x4-in. joist placed between the lever and the roof truss.

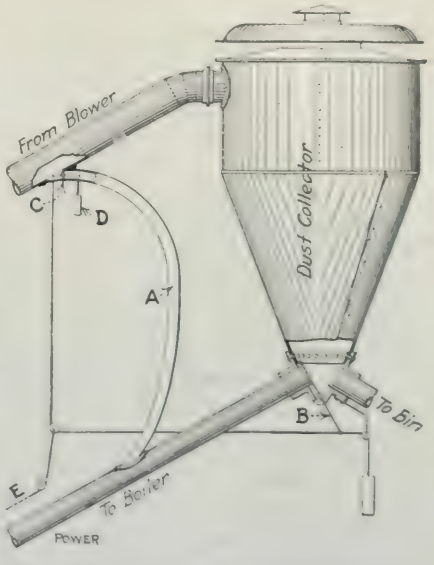
Of course, the fireman denied placing it there. If a reliable pop valve was used this could not happen.

D. E. ADEN.

Wilburton, Okla.

Bypassing the Draft in a Blower System

The illustration shows how I produced draft in a pipe carrying sawdust from a dust collector to a boiler, to move the sawdust when the incline of the pipe was insufficient for it to flow by gravity.



BYPASS AND DAMPERS ON BLOWER PIPE

By tapping the blow pipe, which is 12 in. in diameter, and connecting a 3-in. pipe *A* the outlet of which blows down the 8-in. pipe leading to the boiler, enough draft was produced to blow the sawdust

to the boiler. The gate *B* turns sawdust to the bin or boiler; gate *C* turns on the draft. When the rope is pulled in the boiler room, gate *C* having the light weight *D* attached, opens first, starting the draft. By continuing to pull the rope, gate *B* turns the sawdust under the boiler. By loosening the rope in the boiler room, gate *B* closes first, turning the sawdust into the bin, then the gate *C* closes, cutting off the draft.

CHARLES G. BUDER.

St. Louis, Mo.

Home Made Devices for the Plant

The following are three of the most useful devices about the plant that I know of; they are all home-made, too, The pipe-bending plate, Fig. 1, can be made of a piece of 4x¾-in. angle iron *A* and two sheaves *B*, and is held in a vise while making the bends. The sheaves should have the radius at which it is desired to bend the pipes, but almost any bend or combination of bends can be secured by shifting the position of the sheaves, and they will be smooth and free from kinks and will be uniform. The bender will be found especially useful on brass pipes and conduit.

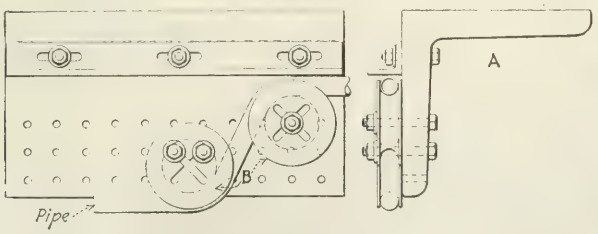


FIG. 1

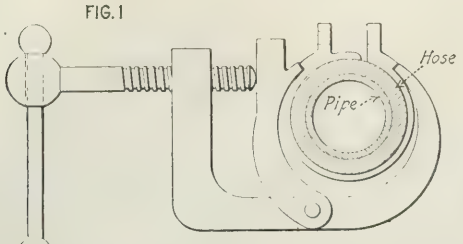


FIG. 2

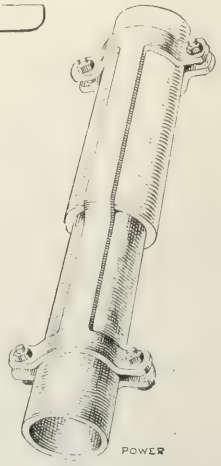


FIG. 3

FIG. 1. PIPE-BENDING DEVICE HOSE-CLAMP VISE A SERVICEABLE HOSE CLAMP

The hose-clamp vise, Fig. 2, makes the putting on of clamps easy. The clamp is slipped on the hose, the vise clamped on it and screwed tight, and the bolt is put in. The vise will be found especially useful on heavy hose used about the boiler room.

The steam hose clamp, Fig. 3, is made of ⅛-in. plate steel cut and bent to suit the hose and pipe for which it is intended. It is the only clamp I have ever found which will positively hold a steam hose working under high pressure. It eliminates all possibility of the firemen getting burned while blowing soot from the tubes.

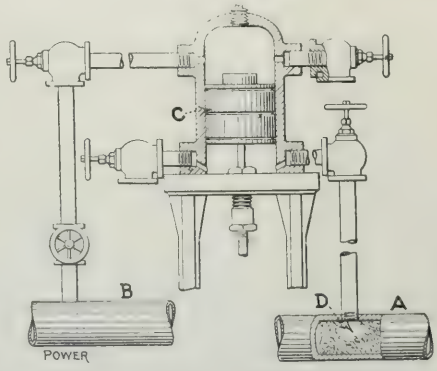
S. H. FARNSWORTH.

Chicago, Ill.

Steam Pipe Peculiarly Obstructed

We had considerable trouble with the 2-in. steam line to our feed pumps. One point of this line kept clogging, at times to such extent as to almost cut off the steam supply to the pumps. At first we were puzzled as to the cause of the trouble, but when it was repeated again at exactly the same point (the pipe being perfectly clean at all others) I concluded that the sediment could not have been carried over with the steam.

These pumps are equipped with piston governors which are connected to the steam and discharge pipes *A* and *B*, as shown. As the pressure in the discharge pipe is higher than that in the steam pipe, and the governor piston *C* leaks more or less, water moves continuously toward the latter pipe, and whatever sediment is in the water moves with it. This sediment remains in solution up to the point where the governor pipe enters the steam line, where it comes in contact with the steam, settles on the walls of the pipe, bakes, and remains as at *D*, until it is chipped out.



OBSTRUCTED STEAM PIPE TO PUMP GOVERNOR

Until provisions were made to remedy this, the pipe had to be removed and cleaned at least once a month.

K. W. KARLSON.

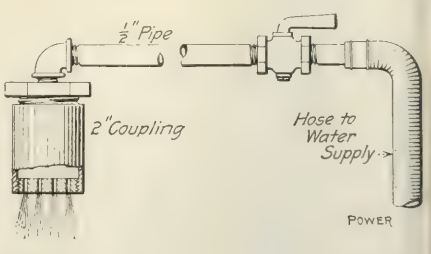
Racine, Wis.

A Handy Home Made Water Sprinkler

In many manufacturing plants, the engine and boiler rooms are the pride of the establishment and every effort is made to keep them thoroughly clean.

One of the most useful articles in our boiler house for this work is a water sprayer for dampening ashes at fire-cleaning time. Those using stationary grates and pulling ashes out through the fire-doors know how the clouds of fine ashes arise and settle on the boiler fronts, and all surrounding walls and objects.

The sprayer is made of pipe fittings, as illustrated. When cleaning a fire, after pushing it back to the bridge-wall, the clinkers are allowed to burn off if possible, then with the damper open, the ashes are sprayed, taking care to keep the water off the hot brickwork and dead plate. The grates are not harmed as they are fairly cool and the water warmed before reaching them.



SPRINKLER MADE OF PIPE FITTINGS

A heavy and serviceable sprayer head may be made by screwing a plug into a 2-in. coupling and cutting it off flush with the end of coupling. Then the plug is drilled full of 3/32-in. holes. In the other end of the coupling bushings are used to reduce to the size of the water supply pipe. A length of ½-in. pipe about 2 ft. longer than the firebox is connected to the water supply with a short length of flexible hose.

This arrangement is also handy for sprinkling floors or anything the movi of which creates dust.

I. S. CHAMBERLAIN.

Paterson, N. J.

Repairs to a Gridiron Valve Gear

Fig. 1 shows the cylinder and valves of an automatic engine I was employed to adjust so that it would require less steam and give better speed regulation. The steam valves are long and narrow and admit steam very rapidly, as they are of the gridiron or multi-port type. When raised they admit steam, and falling they cut it off.

The exhaust valves are of the same general type, but are larger and move horizontally at right angles to the axis of the cylinder. They are operated by the valve-gear, shown in Fig. 2, in which

A is a gage plate that moves parallel to the axis of the cylinder. It is driven by an eccentric on the crankshaft. The secondary links are pivoted at *B* and *C* so that, when the gage plate moves, as indicated by the arrow, the upper valve stem *D* is pushed inward to close the crank-end valve, and the lower stem *E* is drawn outward to open the head-end valve.

In practice the throttle valve could not be opened wide as it would cause the engine to race until there was danger of rupturing the flywheel, and causing other damage. The reason for this action was not plain as all four valves worked freely by hand, and opened and closed promptly when steam was admitted to the cylinder.

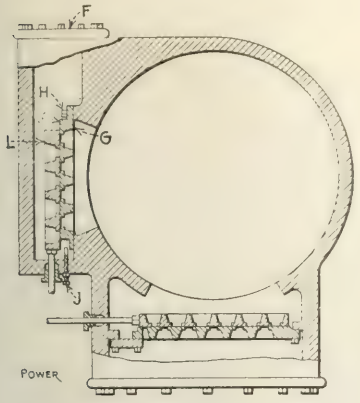


FIG. 1
SECTIONAL VIEW OF GRIDIRON VALVE

Indicator diagrams showed that the cutoff took place early in the stroke, but steam was admitted after the cutoff valve closed, and this was sufficient to give dangerous speed.

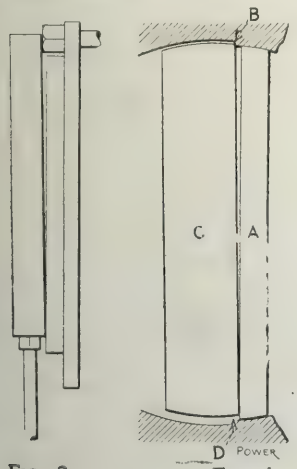


FIG. 3 FIG. 4
IMPERFECTLY SEATED VALVE

To understand the cause for this condition it will be necessary to study the design of the steam valves and their seats. When the round steam-chest cover *F*, Fig. 1, was removed, it was found that the valve seat *G* was held in place by the capscrew *H*, and the lower end was fastened by another screw *J*, Fig. 1. The shoulder *K* in Fig. 1 lets the capscrew heads down low enough to clear the valve *L* when moved to its highest position, or at least this was the case before repairs were completed. As the screw *H* was damaged when removing it to take out the valve seat, a new one was substituted, but the head was longer than the old one. With the valve in its lower position this was not noticed, but when lifted by the valve-gear, it took the position shown in Fig. 3; hence it leaked badly.

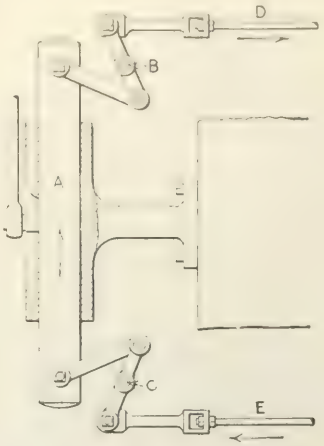


FIG. 2
GRIDIRON VALVE-GEAR

The capscrew was put in a lathe chuck and shortened about ⅛ in., after which there was no trouble from this source, but still the engine raced dangerously. The valve seat *A*, Fig. 4, was taken out and planed off; consequently it was not so thick as before, leaving a shoulder *B* on which the valve *C* would ride as shown; therefore steam leaked through the tapering opening. The opposite edge showed the same defect, at *D*; hence the valve was raised by one or the other shoulders at almost every stroke, admitting enough steam to cause racing unless the throttle valve was nearly closed. To remedy this well concealed, but dangerous defect, I removed the seat *A* and the valve *C* and filed off the shoulders with a half-round file.

This valve was on the head end; the crank-end valve seat had to be given the same treatment. After this was finished the engine maintained a uniform speed with the throttle valve wide open.

W. H. WAKEMAN.

New Haven, Conn.

Criticize These Diagram

The accompanying diagrams were taken from an 8½x6-in. vertical piston-valve (inside-admission type), shaft-governor engine, making 400 r.p.m. under 80 lb. steam pressure. With this type of valve the eccentric is set behind the crank instead of ahead of it. The eccentric is loose on the shaft and is screwed tight to a movable lug in the shaft, the lug being held down by a tension spring against the action of the governor weights.

This engine takes steam from a 2-in. pipe which branches from a main supplying steam to other larger engines. The 2-in. pipe has a reducing valve which reduces the pressure from 120 to 80 lb.

Will readers comment on these diagrams?

VICTOR BONN.

Brooklyn, N. Y.

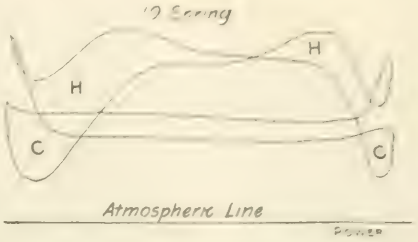


FIG. 1

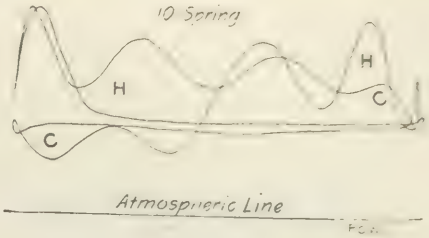


FIG. 2

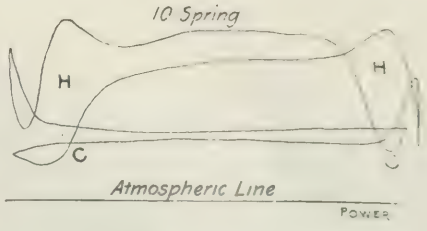


FIG. 3

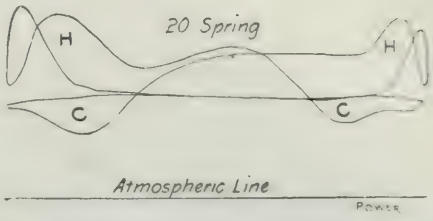


FIG. 4. HIGH-SPEED ENGINE DIAGRAMS

Commenting on the "sootability" of Pittsburgh, *Graphite* says that 40 per cent. of the dust in that city is soot, and that its citizens wear plain black soots, with stripes when it rains. While Pittsburgh is apparently one of those few places where everybody gets sooted, nevertheless a pound of prevention is well worth two pounds of soot.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Correctives for Fluctuating Water Level

In the June 25 issue, B. H. Chester asks the cause of the fluctuating water level in a Babcock & Wilcox boiler.

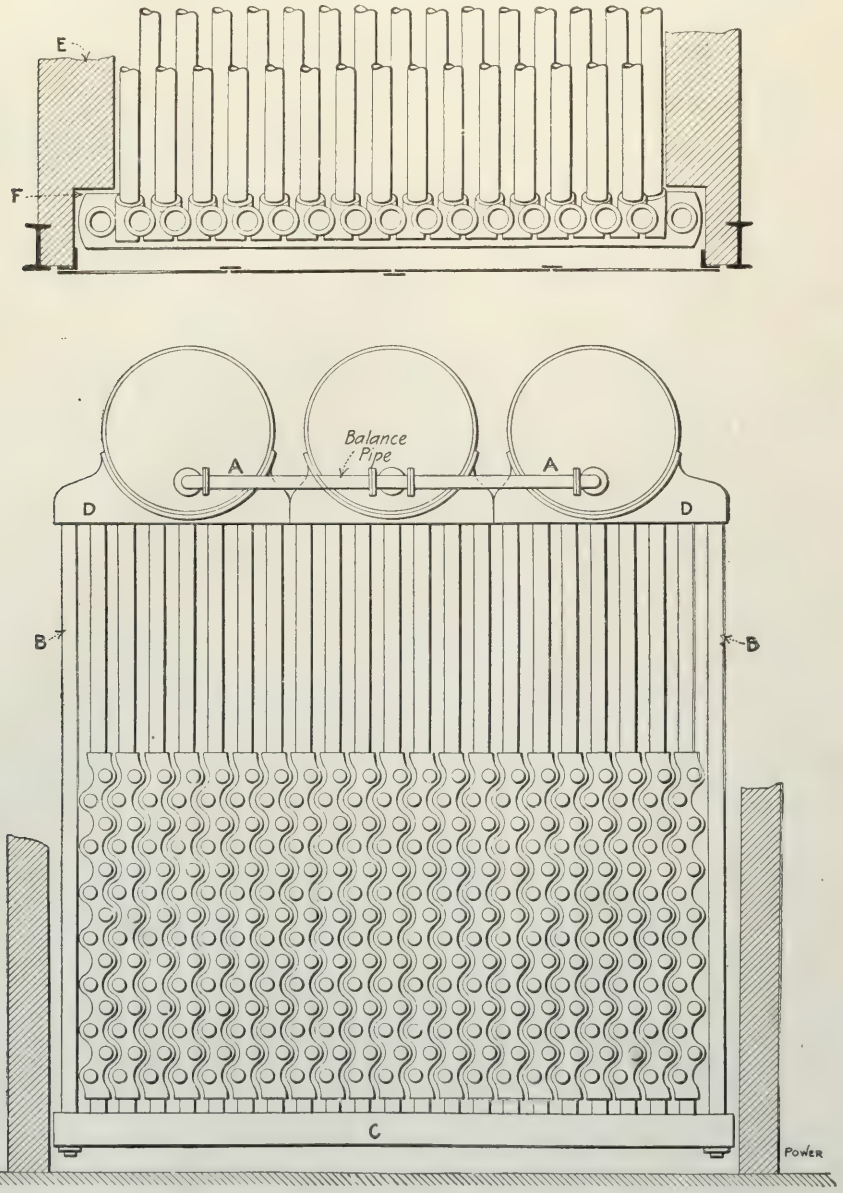


FIG. 1. REAR ELEVATION AND PARTIAL TOP PLAN OF WATER-TUBE BOILER, SHOWING BALANCE PIPES TO AID CIRCULATION

This trouble would be extreme if the mud drum of the boiler was stopped up with mud or sediment. If it is possible for him to attach a water column or gage-glass to both the front and rear heads of each drum, he would perhaps find a greater variation than he thought existed. The illustrations show improvements invented and patented by me to overcome this trouble, which is due to the drums supplying a proportionately larger amount of water to the upper rows than to the lower rows of tube. This causes steam pockets to form in the lower tubes which impede the circulation.

The rear end of the boiler is shown in Fig. 1. The horizontal balance pipe A aids in maintaining a uniform water level in the drums. This pipe should be connected high enough to avoid interference with the downward current in the cross-boxes. The vertical or balance pipes B connect the mud drum *C* directly to the cross-boxes *D*.

The brick setting or iron casing E, Fig. 1, has inner perpendicular pockets *F* into the lower ends of which are projected the ends of the mud drum. This protects the pipes from the heat of the gases and thereby prevents steam pockets

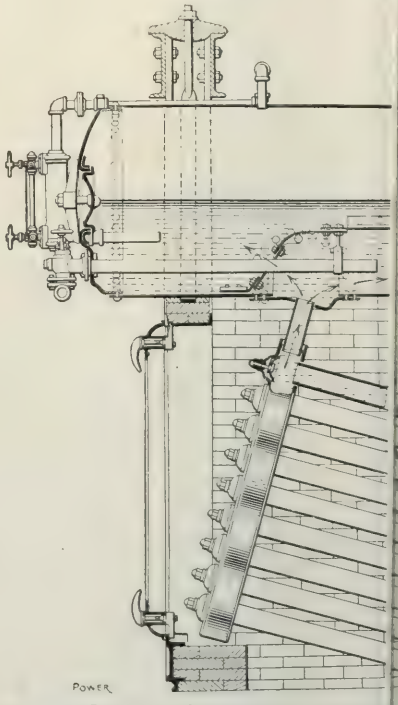


FIG. 2. PARTIAL SECTION OF BABCOCK & WILCOX BOILER

forming within them. This means a free circulation of water and a uniform water level because the lower tubes receive their supply simultaneously with the feeding of water to the upper tubes.

DANIEL WEBSTER

Brooklyn, N. Y.

B. H. Chester's trouble may be due to more than one cause. To begin with, he may have recently been running horizontal return-tubular boilers, which are noted for their steady water level, and he must not expect any water-tube boiler to carry an equally steady water level. I do not think he has anything to worry about.

I have known of a B. & W. boiler, while being forced, to show 2½ gages of water, and in less than 2 min., during which time the fires were banked, the water could be seen to creep down out of sight in the glass. This is due to the large amount of heat absorbed by the water while the fires were forced, and this shows that the condition of the fire has much to do with the water level. Again, the condition of the water will cause the level to be unsteady in any boiler. If the boiler is dirty and has not been washed out frequently, it will prime.

The baffle-plate shown in the partial vertical section, Fig. 2, is supposed to direct the ebullition toward the back end and the space around the plate should be sufficient to let water pass to the water column through the short pipe and be comparatively quiet, but in some cases I believe the space for the water to pass is too great, hence the fluctuating level. It may be that the baffle-plate has been dislodged. I would advise him to first wash out the boiler, and, if this does not improve matters, I would inquire as to the advisability of decreasing the space through the baffle-plate so that the water, as it boils up by the plate will not rush by it to the water column too fast.

A. C. WALDRON.

Revere, Mass.

Mr. Haight's Engine Trouble

A. I. Haight's engine trouble, mentioned in the May 7 issue, reminds me of two similar cases of broken pistons. One was on a simple 20x36-in. Corliss engine running at 103 r.p.m. with 150 lb. steam pressure. The piston was of the built-up type in which the follower plate was held in place by six 1⅛-in. tap bolts. On several occasions the heads of these bolts broke at the end of the thread, and on two occasions, the head and short section of bolt went through the exhaust valve without doing any damage. At another time the bolt head was pushed through the cylinder head without doing further damage. The last time this happened the bolt head caught between the cylinder head and piston, breaking the cylinder head, piston, piston rod, and crosshead shoe and twisting both exhaust-valve stems. A new cylinder head, piston, piston rod and crosshead were ordered and new valve stems made. The new piston was of the same type, but stud-bolts were used instead of tap-bolts.

The follower plate was counterbored on the back to slip over the collar on the studs, and the weakest point in the stud is just back of the collar, so that in case of breakage, the bolt and nut cannot get out and do harm. The reason the old bolt broke was that the piston was sprung out of true. Two different follower plates broke and were replaced with new ones, but as the engine was much needed it was kept in operation.

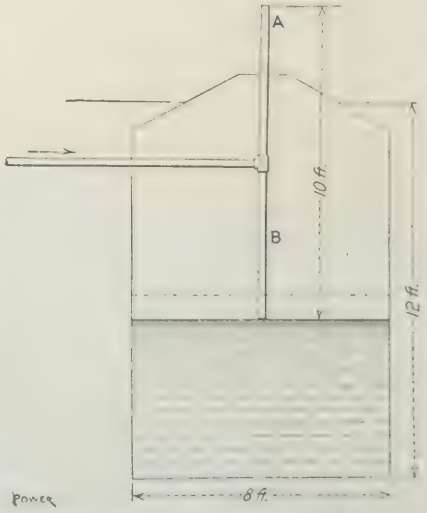
In the other case the cylinder was 32x48 in., having a piston valve and running at 85 r.p.m. The trouble was thought to be due to a piece of a broken snap ring being caught in the counterbore so that it broke out a piece of the follower plate which dropped between the cylinder head and piston, and at the next revolution the nut on the end of the 5½-in. rod pushed the head out, taking part of the cylinder with it and springing the rod. A new valve, piston, valve chamber and cylinder were required, as the valve chamber and cylinder was cast in one piece. In the first case the load on the generator stopped the engine before the throttle was closed, and in the second case the engineer closed the emergency throttle valve as the piston was making the stroke that pushed the head out. A remarkable thing about this accident was that there was a craneman asleep on the floor about 10 ft. directly back of the cylinder and, although he was badly scared, none of the pieces came within 6 ft. of him.

J. C. HAWKINS.

Hyattsville, Md.

Why This Peculiar Exhaust?

A. O. Fisher's inquiry under this heading in the June 25 issue is easy to answer. When the water in the well is



EXHAUST PIPE LED INTO WELL

lower than the end of the pipe *B*, the steam and water rise to point *A*, because nothing prevents the flow of steam to the atmosphere; hence, with the steam, a certain amount of water is lifted.

When the water in the well is above the end of pipe *B*, if the steam escaped at point *A*, it would form a vacuum in pipe *B*, as the end of pipe *B* is sealed by the water in the well. The water in the exhaust naturally takes the line of least resistance and falls through *B* into the well and, in falling, establishes an air current in *A* in the direction of *B*, and therefore carries the steam with it.

If the action of the steam and water coming out at *A* becomes a nuisance or disadvantage to Mr. Fisher, it can be stopped by extending pipe *B* further into the well, so that it will always have a water seal at its base.

C. H. PARSON.

New York City.

A. O. Fisher mentions that when the bottom of a pipe into which the exhaust pipe of his engine is connected, is covered with water, no steam is exhausted to the atmosphere, but, when the water falls low enough to uncover the bottom of this pipe, exhaust to the atmosphere does occur.

In times of high water the well acts as a jet condenser and the cool water in it condenses the steam in the lower leg of the pipe, thereby causing enough vacuum to draw all the steam down to the water so that none escapes from the top of the pipe.

G. H. MCKELWAY.

Brooklyn, N. Y.

Is the Packing Defective?

Replying to James Pierpont's inquiry under this heading in the July 2 issue, I would say that my experience with the style of metallic packing he describes extends over several years and I have found only one effective way of preventing it from leaking. After ascertaining that the steam is not leaking around the packing case (which can only be remedied by a renewal of the composition metallic ring between the cylinder end of the stuffing-box and packing case) remove and thoroughly clean the packing with gasoline, polishing the wearing surfaces to remove any dirt or scale lodged there.

When reassembled, give the piston rod a coating of a mixture of cylinder oil and graphite before replacing it in the stuffing-box. Unless the piston is scored or badly worn this should remedy the trouble. I always have an oil cup connected above piston rods using metallic packing and use cylinder oil as a lubricant and have experienced but little trouble.

L. A. LESHER.

Mandeville, Ark.

If the piston rod of Mr. Pierpont's engine is badly worn or fluted, the only remedies are to have it turned down or, better still, have a new one made. If it is in fairly good condition see that sufficient clearance is between the different sections to allow the spring to hold them against the rod without binding at the ends. Cut a piece of valve-stem packing, not larger than ⅜ in. and long enough to fit nicely in the stuffing-box; cut the ends square and butt them together. Put one such ring in the bottom of the box and another next to the

gland. This will prevent steam from leaking by the packing case. Draw up the gland as tightly as possible and it should be steam-tight.

H. C. MacAulay.

Rumford, Maine.

Mr. Pierpont's trouble is undoubtedly due to weakness of the springs that hold the metallic rings against the piston rod. Perhaps the rings are worn excessively; if so, new rings and springs should be used.

If the springs are at fault they may be given more tension by pulling the coils further apart. I would suggest that Mr. Pierpont remove and clean the packing and the stuffing-box. This would possibly correct the trouble.

R. G. Cox.

Harrisburg, Penn.

I have had some experience with the packing which Mr. Pierpont mentions and have always found it very satisfactory. If the rod is worn flat, the only way to get a tight job is to turn down the rod, taking off enough to make it round. If the rod is round, perhaps the rings do not fit it tight enough, in which case the rings should be removed and the springs shortened. In reassembling care should be taken to break the joints of the two rings in each case.

With this kind of packing a lead ring is placed in the bottom of the stuffing-box to prevent steam leaking out around the case. This ring is made from ⅛-in. lead pipe. Some engineers lap the ends but I have had better success by reaming out one end of the ring and shaving the other end to fit it. The ring should be made as large as the stuffing-box as it makes a better joint. In Mr. Pierpont's case it may be necessary to place a lead gasket between each of the cases, as steam leaking by the bottom set of rings will find a way of escape between the cases.

C. B. Hudson.

Lowell, Mass.

Cleanliness and Mechanical Efficiency

From C. E. Aldrich's letter under this heading in the June 11 issue, I infer that he is not wholly in favor of cleanliness in the plant. If he means cleanliness obtained at the sacrifice of efficiency, I agree with him. This thing in its relation to the plant is the same as its relation to the body. Most of us can do just as much work with a dirty face and overalls as we could with clean ones. In some plants this is done, but they are not pleasant ones to visit.

A person feels better when clean and a clean machine has the same effect on him. To clean regularly means that when repairs are necessary it does not require half a day's digging to find the thing to be repaired. No sane man would let a generator or motor be dirty. Why should he do it with anything else? A man with ambition and pride enough to keep an apparatus clean and polished on the outside will not neglect the inside, as Mr. Aldrich intimates.

The man who is ready with excuses for a dirty or pounding engine, leaky or plugged lubricators, choked ash and soot pits, etc., is spontaneous with excuses for anything else that is wrong. We all might take a lesson from the way such work is done on ship-board. If the management is reluctant to buy cleaning supplies it is unfortunate, but once it sees a dirty plant made clean it is encouraged to do so.

A. P. Hyde.

Binghamton, N. Y.

C. E. Aldrich is right in his conclusions about mechanical efficiency being more desirable in an apparatus than outside polish. But I see no reason to criticize that Western concern for offering a prize for the best looking apparatus, "outside and in," since that indicated that it is giving proper consideration to both and educating its men to obtain, not only an efficient plant, but a clean one as well.

I do not think Mr. Aldrich intended a criticism, but his letter leads one to consider it as such.

Lloyd V. Beets.

Nashville, Tenn.

Modern Pumping Plant

In the issue of May 7, Kirby Thomas presents a highly entertaining photograph of a pump run by a "Missouri hay burner" and with gears that should set the *American Machinist's* teeth on edge. But seriously, while the efficiency may be only 15 or 20 per cent., even when we neglect the thermal efficiency of the prime mover, that pump is all right. While I feel a trifle hurt, since I used to see as crude devices as this when I was a native son of California before the state became a chicken fence of electric lines, my personal feelings are not the point. There is often a tendency to over-refinement on the part of engineers, only held in check by the unprogressive ideas of the "old man," and this extreme case may serve to illustrate "Tightwad's" side of it.

Speaking of the general case where this type of water elevator is used, rather than any special case, it is to be doubted whether, even Mr. Thomas, if called upon by the *ranchero* to act as consulting engineer, and with all the data before him, would prescribe any other method. This one-mule power should be sufficient to raise about 50 gal. of water per minute to a height of 15 ft. with our mechanical efficiency of 20 per cent. If the mule were city fed it might be profitable to replace him, but a Mexican mule depends about as much on fancy foods, as his Vicksburg driver, who lives near the coal yards, depends upon city gas or electricity for cooking. "Fuel" for the mule is more or less cheap, costing almost nothing on some ranches; hence a mule is a good sort of furnace and boiler to use.

Where these installations are common, it happens that the service is decidedly intermittent, water being required for occasional irrigation of the small garden patches between an *arroyo* bed and the high ground—often but an acre or two in one place, or within range of one pump. If the "engine" was not busy at this, it would be hobbled and wandering off on three legs toward the foothills in search of a change in diet.

Why then own the mule at all? Because once in a while he is wanted for a trip to town, and it often happens that several are wanted. So the investment has to be made anyway and if he is fed anything at all it takes about as much to keep him "under steam" while loafing as while working. Even if he had to be bought for this purpose only, his price plus several generations of him would be less than a small electrical equipment, including transmission lines and an "infant" centrifugal pump.

E. G. Medart.

Quincy, Ill.

Electrolysis Pipe Corrosion Doubted

Mr. Phelps' suggestion, in the June 2[illegible] issue, that the destruction of the pipe was caused by electrolysis, gives very meage information, but it would seem to b unlikely, for he mentions that the plac where the trouble occurred was not nea either street railway tracks or pipes tha might lead to those tracks.

If the eating away of the pipe ha been caused by electrolysis from stra electric railway currents, the curren would have to have a way to reach th pipe from the tracks and a way to retur to them or to the power house throug pipes, water, or some other conductor.

If the gas pipe runs near the track at some place, even though distant fro the point where the trouble is, electrolysi might reasonably be thought the caus if some fairly good conducting path le from the damaged pipe to the powe house, but in the absence of this th theory would seem to be untenable an the cause should be looked for elsewher although where it would be hard to say.

G. H. McKelway.

Brooklyn, N. Y.

An oil-burning furnace equipment fo its new shops will be installed by th Portland (Ore.) Ry., Light & Power C

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Milky Appearance of Deep Well Water

Is the milky appearance of water when first drawn from a deep pipe well any indication of presence of impurities which may render the water unfit for use as a boiler-feed water? What is the cause of the appearance and how may its nature be determined?

T. W.

The appearance of the water when first drawn, is probably due to liberation of gas from the water, generally the liberation of carbonic acid in the form of minute bubbles which impart a whitish appearance to the water, and are mostly redissolved as the water clarifies. As few well waters are free of lime, the presence of carbonic acid tends to make a harder boiler scale.

The best test for presence of carbonic acid is to take a sample of the well water after it has clarified, and add an equal quantity of clear lime water. If carbonic acid is present a milky precipitate of carbonate of lime will be formed.

Blowing Off Boiler under Pressure

Why is it objectionable to blow off a boiler when under pressure?

H. C.

Because the less the time employed in relieving the joints of stress the greater the danger of disturbing the joints and causing them to leak.

To Find Pitch of Rivets

How can the pitch of the rivets be determined for a double-riveted butt and double-strap joint which is to have ⅞-in. rivets and a strength of plate between the rivet holes on the outer row which will be 82 per cent. of the strength of the solid plate?

W. S. F.

Let

P = Pitch;

t = Thickness of plate;

TS = Tensile strength of plate;

d = Diameter of rivets.

Then

$P \times t \times TS$ = *strength of solid plate*

and

$(P - d) \times t \times TS$ = *strength of plate* between the rivet holes on the outer row. The conditions require

$$\frac{(P - d) \times t \times TS}{P \times t \times TS} = 0.82 \qquad (1)$$

By canceling out t and TS from numerator and denominator of the first member of (1) we obtain,

$$\frac{P - d}{P} = 0.82 \qquad (2)$$

and as $d = 0.875$, the equation becomes

$$\frac{P - 0.875}{P} = 0.82 \qquad (3)$$

from which it is found that

$$P = \frac{0.875}{0.18} = 4.86 - \qquad (4)$$

or practically 4⅞ in.

Openhearth Steel

What is meant by basic and acid methods of manufacturing openhearth steel for boiler plates?

B. A.

The process is basic or acid, depending on the kind of furnace lining employed. The basic process is a dephosphorizing and desulphurizing one, and is the one most commonly employed, as it can use pig iron that is high in phosphorus; while the acid process is one in which the furnace lining is acid or silicious, and therefore not well adapted to remove phosphorus.

The growing popularity of the basic openhearth process is due to the exhaustion of low phosphorus ore.

Power Required for Ice Machine

How much horsepower is required to drive a 100-ton ice machine?

A. H. A.

One hundred and twenty to one hundred and thirty, according to the efficiency of the plant.

Merits of Scotch Boiler

How does the Scotch marine boiler compare with the horizontal return-tubular boiler as a stationary boiler where space is very limited?

T. R.

The Scotch marine boiler has the advantage of being free from infiltration of air through a brick setting, but shares the disadvantage of most internally-fired boilers, viz.; that little time and space are given to the gases to burn completely before coming into contact with the cool surfaces surrounding the furnace. It has the advantage of compactness, and for short runs, has the further advantage that there is no mass of brickwork to be alternately heated up and cooled off.

Cause of Engine Belt Running to Side of Pulley

What will cause the main belt of an engine to run to one side of the pulley when the receiving jackshaft is parallel with the engine shaft and both pulleys are evenly crowned?

J. N.

When iron pulleys are faced, the cut of the turning tool is usually made in one direction clear across the face and the shallow screw thread made by the turning tool has some tendency to urge the belt to one side of the pulley. But when iron pulleys are finished with an ordinary smooth cut of the turning tool, the ridges soon become worn down so smooth as to produce no side creeping of the belt, and if the belt continues to run to one side it is due to the fact that one side of the belt is tighter than the other, the tighter side running to the crown of the pulleys.

Screening Air Supply

What is the best way to prevent floating dust and grit from being carried with the air supply into an air compressor?

C. G. C.

By taking the air supply through a good sized duct which is divided across by a number of cheese-cloth screens.

High Duty Attachment for Pumps

Why is it impossible to use steam expansively in a direct-acting pump unless it is fitted with a high-duty attachment?

M. T. D.

In a direct-acting pump the resistance overcome by the piston is nearly the same at every part of the stroke. When steam is used expansively its pressure decreases during expansion, and a high-duty attachment is necessary for storing up as much energy before cutoff takes place as may be necessary, during expansion, to be expended as useful energy, in addition to the energy which is developed by expansion.

Study Questions

This Week's Questions
Last Week's Answers

71) Steam is turned into and condensed in a barrel of water until the weight of water is increased from 360 to 379.1 lb. and its temperature raised from 52.8 to 109.6 deg. F. The steam pressure is 79.8 lb. gage. What percentage of moisture did the steam contain?

(72) What is the indicated horsepower of a 28 and 52 and 72 by 54-in. triple-expansion engine running at 100 r.p.m., when the mean effective pressures are, in the high-pressure cylinder, 60 lb., intermediate 38 lb. and low 18 lb.?

(73) To withstand a shearing strain of 20,000 lb., how many ⅞-in. rivets will have to be used, if their ultimate shearing strength is 40,000 lb. per sq.in. and a factor of safety of 4½ is to be allowed?

(74) Assume a body weighing 2½ short tons moving without friction or air resistance. It has just covered 315 yd. in 179 sec. with no force acting upon it. For a period of ⅔ min., what force would have to act so that thereafter the body would cover 10,800 in. in $^1/_{50}$ hr.?

(75) To change the air in a room, 40x80 ft. by 12 ft. high, five times an hour, what quantity of air per minute would a ventilating fan have to discharge?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(66) Let

x = First engine's rated hp.;
y = Second engine's rated hp.

Then

$0.80x$ = Hp. first engine actually develops;

$1.20y$ = Hp. second engine actually develops;

and

$$x + y - 10 = 0.80x + 1.20y \quad (1)$$

but

$$0.80x = 1.20y$$

Therefore

$$x = \frac{1.20y}{0.80} = \frac{3y}{2} \quad (2)$$

Substituting in (1)

$$\frac{3y}{2} + y - 10 = 1.20y + 1.20y$$
$$3y + 2y - 20 = 2.4y + 2.4y$$
$$5y - 4.8y = 20$$
$$0.2y = 20$$
$$y = 100 \ hp.$$
$$x = \frac{3y}{2} = \frac{300}{2} = 150 \ hp.$$
$$0.80x = 120$$
$$1.20y = 120$$

The rated capacities are therefore 150 and 100 hp., respectively, and the power actually developed by each, 120 hp.

(67) The ratio of the voltages is as the number of turns, therefore,

$$110 : 2000 : : 100 : x$$
$$x = \frac{2000 \times 100}{110} = 1818 \ turns$$

(68) Let x = original amount.

After giving 1 lb. to an oiler there is left $x - 1$. Half of this goes to the boiler room, leaving $\frac{x-1}{2}$. Another pound is given an oiler, leaving $\frac{x-1}{2} - 1$ at the end of the first day. Call this quantity a.

Then, as the same thing is done each day, there will remain at the end of the second day

$$\frac{a-1}{2} - 1$$

Call this quantity b.

Then at the end of the third day there remains

$$\frac{b-1}{2} - 1$$

Call this c.

At the end of the fourth day there remains 1 lb.; hence

$$\frac{c-1}{2} - 1 = 1 \text{ or } c = 5$$

Then

$$\frac{b-1}{2} - 1 = 5 \text{ or } b = 13$$

and

$$\frac{a-1}{2} - 1 = 13 \text{ or } a = 29$$

and

$$\frac{x-1}{2} - 1 = 29 \text{ or } x = 61$$

The problem may be worked backward arithmetically, adding on each time what had been given out, starting with the 1 lb. left, as follows:

1 + 1 = 2 2 + 2 = 4 4 + 1 = 5	Fourth day
5 + 1 = 6 6 + 6 = 12 12 + 1 = 13	Third day
13 + 1 = 14 14 + 14 = 28 28 + 1 = 29	Second day
29 + 1 = 30 30 + 30 = 60 60 + 1 = 61	First day

(69) The formula for the efficiency of a hydraulic ram is

$$n = \frac{qh'}{(Q - q)h}$$

where

q = Quantity of water pumped;
h' = Height to which it is raised;
Q = Quantity of water supplied;
h = Head of the supply.

The problem gave $n = 60$ per cent.; $h' = 60$ ft.; $Q = 100$ cu.ft. per sec.; $h = 10$ ft.

Substituting

$$0.60 = \frac{q \times 60}{(100 - q) \times 10}$$

Simplifying

$$60q = (100 - q)\,6$$
$$60q = 600 - 6q$$
$$66q = 600$$
$$q = 9.09 \ cu.ft. \ per \ sec.$$

As 1 cu.ft. = 7.48 gal., the gallons per minute discharged is

$$9.09 \times 60 \times 7.48 = 79.59 \ gal.$$

(70) The pressure corresponding to 1 in. of water is 0.0361 lb. per sq.in. Then 20 in. of water is

$$0.0361 \times 20 = 0.722 \ lb. \ per \ sq.in. \ gage$$

For absolute pressure the atmospheric pressure must be added. As 1 in. of mercury corresponds to 0.491 lb. per sq.in. the atmospheric pressure at 29 in. of mercury is

$$0.491 \times 29 = 14.239 \ lb. \ per \ sq.in.$$

Then the absolute pressure required is

$$0.722 + 14.239 = 14.961 \ lb. \ per \ sq.in.$$

Power Gas from Coal Seam

The theoretically interesting scheme for producing gas directly in the coal deposit and utilizing it on the surface, is to be tried in both England and the United States, according to the statement by Sir William Ramsay before the recent meeting of the Institution of Mining Engineers, in furtherance of the proposition which he first brought forward about three months ago.

The scheme, according to *Mechanic World*, involves driving bore holes down to the coal seams; lighting the coal and conveying to it water and air to produce a combustible gas which is to be brought to the surface by bore-pipes and then used in gas engines to supply power for industrial and other purposes. As the gas temperature would be very high, the oil would distill in the pipes and be separated from the gas in some way and would also be conveyed to the surface for use.

If gas can be thus produced, its escape through cracks and fissures cannot be prevented, and the subterranean combustion of coal would lead to serious subsidences and collapses. Also, a fixed composition of the gas cannot be relied on. The incursion of water, which has already to be dealt with in ordinary coal-mining work, would also play a part in connection with the underground fire. Whether these and other difficulties are provided for, the scheme may safely be tried.

The author of the proposition states that the system is to be tried, hence coal-mining and financial interests are evidently prepared to incur the necessary expenditure.

Confessions of an Engineer

By R. O. Warren

Due to excessive friction in shafting bearings, needless power is expended.

Engineer Warren advises Wood to buy an engineer's level, the use of which astonishes the manager.

If one thing more than another irritates, it is interference, and I was sometimes nettled with Wood's attention to my power plant. It is true that he had caught me napping and neglecting opportunities more than once, but that did not cause me to look upon his interference with any additional favor, and I made up my mind that if the chance ever came I would jump on him with both feet. The opportunity came sooner than I anticipated.

Increased Load on Engine

I had noticed for some time that the main engine was cutting off later in the stroke than it had for months before, but I made no attempt to ascertain the cause of the additional load. The coal report also showed an increase in fuel consumption over what had been used when we first began keeping plant records.

One day when Wood came into the engine room I casually inquired if business had picked up and how many new machines had been put in operation. Wood replied that business had been a little slack for the past year and that no new machines had been added to the equipment.

The matter was dropped and I began to look about for the cause of increased power consumption. My power-plant apparatus was carefully gone over until I knew that the trouble was not there. The boilers were free of scale; therefore the increased fuel consumption could not be due to dirty boilers. It was "dollars to doughnuts" that the seat of the trouble was out in the factory.

I hunted up the old mechanic in charge of the repair work about the mill and asked him, in a way that indicated but slight interest on my part, how often he lined up the shafting.

"Gosh-almighty! Them shafts h'aint been lined since the factory was built. They don't need it, for when they were put up I put them there for keeps. I never have to do my work twicet."

Found Shafting Out of Line

When I got an opportunity, unobserved, I sighted along a line of shafting and found, with the aid of the targets used, that the shaft was out of level as well as out of alignment sidewise.

Here, then, was the root of the trouble. The shafting had been allowed to take care of itself and the increased friction in the bearings was the cause of the increased power consumption. I followed my investigations further and was convinced that while Wood had been attending to power-plant matters the factory shafting had been neglected, and it made me feel good.

Further talk with the old mechanic disclosed that the old method of lining the shaft had originally been used—that is, of stretching a wire parallel with the shaft and measuring from the shaft to the wire. The old man had used a spirit-level on the shaft from hanger to hanger and, although he had doubtless been careful in doing his work, the process must have been tedious and uncertain, and considerable error had crept in between one end of the long shaft and the other. With this information I awaited the next visit from Wood.

Don't Blame Me. It's up to You, Wood. You Remove Excessive Line Friction and there Will Be no More Trouble in Keeping Up the Speed

Speed Drops

In a few days he came and wanted to know why the speed had fallen below normal that morning. Matters could not have been better suited to my purpose.

"Below normal?" I exclaimed, as if surprised.

"That is what I said," snapped Wood. "Furthermore, this is not the only time the speed has dropped during running hours."

"I guess that is true," I replied. "I have noticed that the load on the engine has been steadily increasing of late and, what is more, it is not all productive work. There is something like a 30 per cent. friction load on this engine that does not belong there, but you can't come in here and place the blame on me; it is up to you, Wood. If you will remove some of that friction there will be no trouble in keeping up the speed, because there is not enough boiler capacity to overcome ten times as much friction as there is any need of. Of course, we can put on another boiler if you don't want to get onto your job of lining the shafting."

I was enjoying myself to the full, and talked to him like a brother.

"No, no!" hurriedly ejaculated Wood at my suggestion "There's no need of any more boilers. You say it is up to me to reduce the load. What do you expect me to do, stop manufacturing?" This with sarcasm.

Warren Gives Advice

"Hardly that," I replied. "But if I were superintendent I would take a 'slant' at the line shafting and see how much it is out of line. I have got a ten-spot that says the shafting is 1 in. out of true, and I will venture to say that it is even more."

"What makes you think so?" asked Wood, with surprise.

"I don't think it, I know it! What can you expect when a shaft in this mill hasn't been lined up since it was first installed, and then the old spirit-level and line system was used?"

"When this mill was built," replied Wood, "it was built extremely heavy and I don't believe that the shafting is out of line. I will say this, however, that I have not had the chance to look after it since I have been on the job. If there is anything in what you say, I am willing to eat crow. I will have the machinist start looking over the shafting as soon as we shut down Saturday noon."

Shaft-aligning Instrument

"Going to use a spirit-level and wire?"

"Of course, what else would I use?"

"Why don't you blow yourself to an engineer's level?"

"A what?"

"A shaft-aligning and leveling instrument. You can get a telescopic level like 'good' engineers and surveyors use, if you have the price, and that wouldn't break most men."

I said this with as much sarcasm as I could command. "But just to show you that it would be money well invested, I know of one firm that saved nearly $1000 in one year by keeping its shafting in shape. They had about 2000 ft. of it and went over it in something less than 50 hours. It would take some time to do the same job your old-fashioned way."

Wood winced, but was game and wanted to know more about the instrument.

"What does it look like; how does it work?"

Seeing he was getting interested, I explained: "The outfit consists of an architect's level and some targets. Some of these levels have a telescope that will magnify about 20 times and are good over a shaft 600 ft. long. That will cover anything we have got here.

"When you get ready to line up a shaft, put up a fixed target that is arranged for horizontal and vertical adjustment. A lantern behind this fixed target makes it an easy matter to work in dark places or at night, if necessary. A portable target can be hung from the shafting by a clamp so designed that the center of the target will remain the same distance from the center of the shaft, no matter what the diameter may be."

"Seems simple enough, but won't the cost of keeping the shafting aligned all the time be greater than the amount saved by reducing the friction?"

"Of course not; although the first cost of lining up will be greater than at any other time, you won't have to line up the shafting every day. About four times a year will be enough, because much of the machinery is light and wear on the shafting is not excessive."

What Is Saved

"Why, I know a big mill where the cost of lining shafting was about $500 and a saving of something like $1700 was pocketed annually over the cost of operating before the shafting was put in shape."

"Well, that sounds good and it may be as you say," rejoined Wood, who was dying hard; "but I don't see why shafting should require lining four times a year."

"The reason is because a building will settle, timber will warp and shrink, belts will pull sideways, up and down and at angles, and bearings will wear. There is no getting away from that."

"I guess those reasons will hold water. I believe that machine would come in handy for putting up countershafts, etc., as well," continued Wood, with a trace of warmth. "It wouldn't take long to train someone to use it, I suppose?"

"Humph; it will almost work itself. Any mechanic of ordinary intelligence can do the work. You could yourself," I added as a knock. "In fact, but two men are required to work the thing and the lining can be done in about one-fourth the time required with the old method. There is no bother with removing belts while doing the work, and it is a wonder to me that you don't buy one of these machines and be uptodate."

Wood mused for a time and then said: "How are you going to work where there are obstructions under the center of the shaft? There are several such lines in our mill."

Works Under All Conditions

"The easiest thing in the world, for in such a case a portable target can be attached to suitable rods so as to come free of the obstructions at one side. To tell the truth, I don't know of a condition under which the instrument cannot be used. Even brick walls are no barriers, as it is merely a case of drilling a small hole through in order to obtain a sight of the fixed target at the other end of the shaft.

"More than that, the level can be used to good advantage in putting up pipe lines, especially water pipes in which pockets are likely to form and freeze in cold weather, as the proper pitch can be obtained.

"In my opinion, you can't go wrong in getting one of these instruments, and the sooner the better, for, honestly, Wood, the way we are throwing away coal in overcoming useless friction would make the directors slice your salary if they knew it."

I had Wood on the run. He gave a feeble smile and said that he would look into the matter, adding it was just as important to look after the economical operation in the mill as it was in the power plant.

Wood Acts

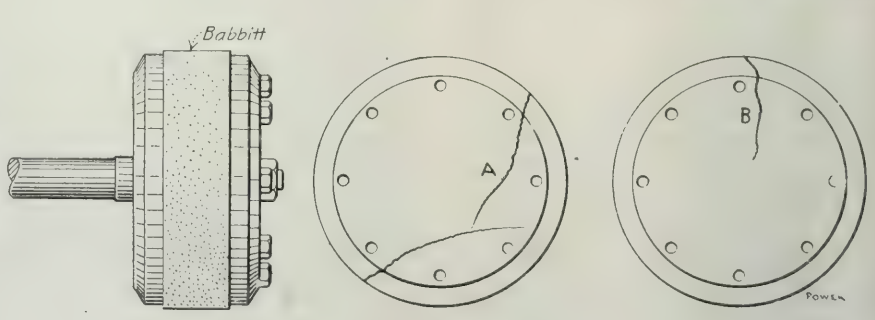
It wasn't many days before I noticed that the coal consumption was decreasing, and an investigation showed that Wood had purchased an engineer's level and was using it. I do not recollect the saving made by keeping the shafting aligned, but perhaps Wood's own words will give an accurate idea of the value of the device, for he said later, when speaking of its work: "No living man could have made me believe that those shaftings were so out of alignment, or that there was so much needless friction in the bearings."

Wood was game, however, and invited me to hand him another whenever I could. But I had more troubles of my own before I again caught him napping.

Emergency Piston Repair

The piston of a condenser air pump fractured when the condenser lost its vacuum and the pump ran away. The fracture in the 12-inch piston is shown at A and that in the follower head at *B*.

As this air pump must be kept going the engineer decided to try a babbitt piston. The rings, which were also fractured in the accident, were removed and the space between the front head of the piston and the follower plate was filled with babbitt, turned off to a smooth surface and of a diameter to fit the cylinder. The piston was placed in the cylinder, and the machine started up and was operated for three weeks before a new piston was secured.



How the Piston Was Repaired

Stone Boiler Setting

On the outskirts of the little town of Duryea, Penn., on the banks of a small stream, can be found an old dilapidated sawmill in which the belts are patched, the wheels run askew, the flooring is missing in places and the entire plant has the appearance of having seen its best days. The old mill is

FIG. 1. OLD MILL WITH ITS 6-FT. WATERWHEEL

shown in Fig. 1, with its little 6-ft. waterwheel that furnishes power when there is enough water.

When the water is low, a 40-in. by 14-ft., return-tubular boiler and an old slide-valve engine are pressed into service. The boiler is a curiosity, inasmuch as there is no brickwork included in its foundation or setting. The setting is of stone, held together by mud or a certain kind of fireclay with which is mixed salt and wood ashes. The setting is shown in Fig. 2.

The boiler is fitted with a ball-and-lever safety valve, but the lever is loaded with two ordinary lever weights, and, although the owner of the place maintains that the boiler will blow off at 100 lb. pressure, the weights on the lever would suggest that this pressure is far below the actual blowing-off point.

In spite of the fact that the boiler is of uncertain age, a pressure of between 90 and 100 lb. is carried. The owner, insists that the shell is as strong as ever, but admits that the tubes are "a bit weak." Some day the neighborhood will probably be startled by a "mysterous" boiler explosion.

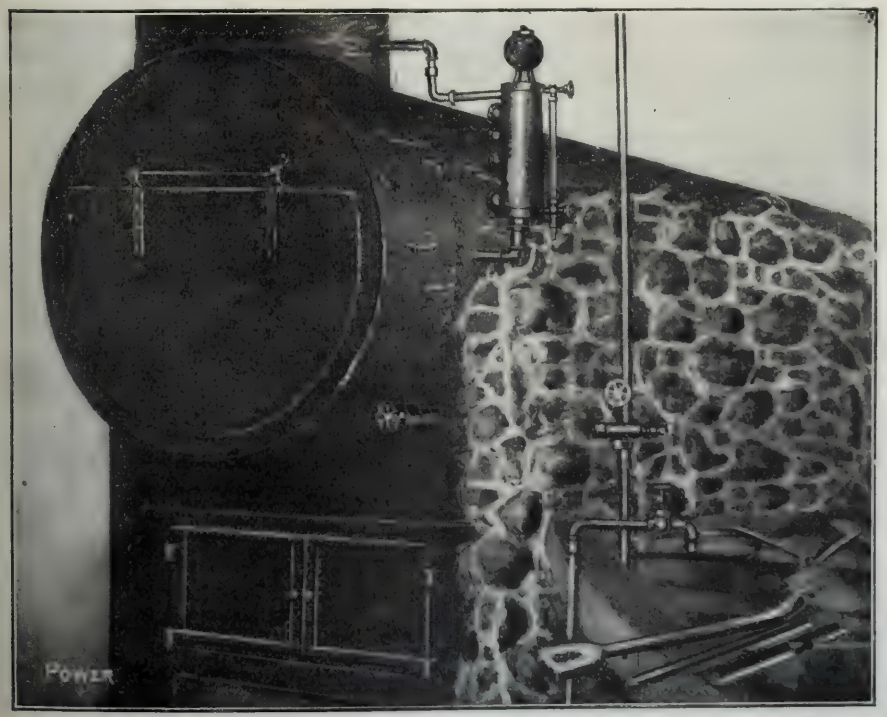


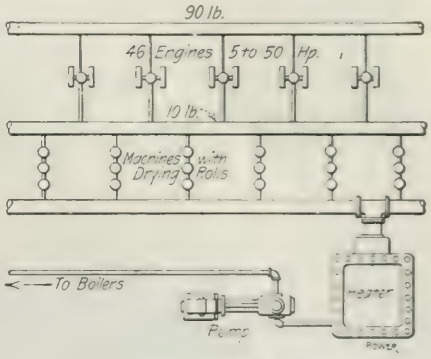
FIG. 2. STONE BOILER SETTING

Saving the Returns

Much of the loss obtaining in textile work is due to the waste of return condensation, sometimes to its contamination by dye stuff and more usually because proper provision has not been taken to pipe the returns to suitable apparatus for using it over again.

A notable instance can be found at the Garner Print Works, Garnerville, N. Y., wherein the chief engineer, Robert L. Sheraton, utilizes the condensed water coming from 50 small engines which are used throughout the plant. The steam for these engines is taken from a high-pressure main, and the exhausts of each are connected to a single exhaust-steam header.

In the process of manufacture, drying machines are used which require a large amount of steam in the drying drums. Steam for this purpose is taken from the main exhaust-steam line, and after passing through the several machines at a pressure of 10 lb., it is discharged into a second exhaust-steam main. It is then conveyed to a large open heater, and then pumped back to the boilers, without undue waste of steam from any of the auxiliary engines, except that which may occur due to piston-rod leakage, etc.



EXHAUST STEAM PIPING

Under former conditions, it was wondered why the returns coming to a certain heater were at such a low temperature, and one day it was accidentally discovered that the return pipe which runs underground was in the direct path of a cold-water discharge through which waste water passed to a creek, and this waste water was discharged directly upon the hot-water returns to the heater.

The elimination of this source of waste, and adding other improvements have enabled several boilers to be cut out of commission, with a corresponding decrease in fuel consumption.

Old Centennial Boilers

A part of the installation of steam boilers at the Dutchess works of the Garner Print Works, Wappinger Falls, N. Y., is a battery of an old type of boilers seldom seen on this side of the water, three of which have an interesting history. At this old boiler plant, which is about to be discontinued, there are 10 boilers, three of the Galloway type and seven drop-flue boilers.

The Galloway boilers were made by Galloway & Sons, Manchester, England, in 1876, and were brought over to this country and installed at the Philadelphia Centennial. When they had served their purpose they were removed to their present position and have been in use ever since, running at a pressure of 50 lb. per square inch.

The other boilers are what were commonly known as the drop-flue type, in that there are three flues attached to a header connection at both ends of the boiler and two flues through which the return gases pass to the stack. These boilers are fitted with gage-glasses and try cocks, but the Galloway boilers are equipped with the gage-glasses only, there being two on each boiler, one to be used in case the other gets out of order. These Centennial Galloway boilers are 7 ft. in diameter, 28 ft. long, and have two furnaces each 36 in. in diameter. They will soon be displaced by new, uptodate water-tube boilers.

On the main steam header, which extends over the top of the boilers, are several interesting expansion joints, one of which is shown in Fig. 2. They are made of wrought-iron disks, which are bolted together with suitable packing inserted between the two inside faces of the flanges. These expansion joints are fitted into the main steam header, which is constructed of cast iron.

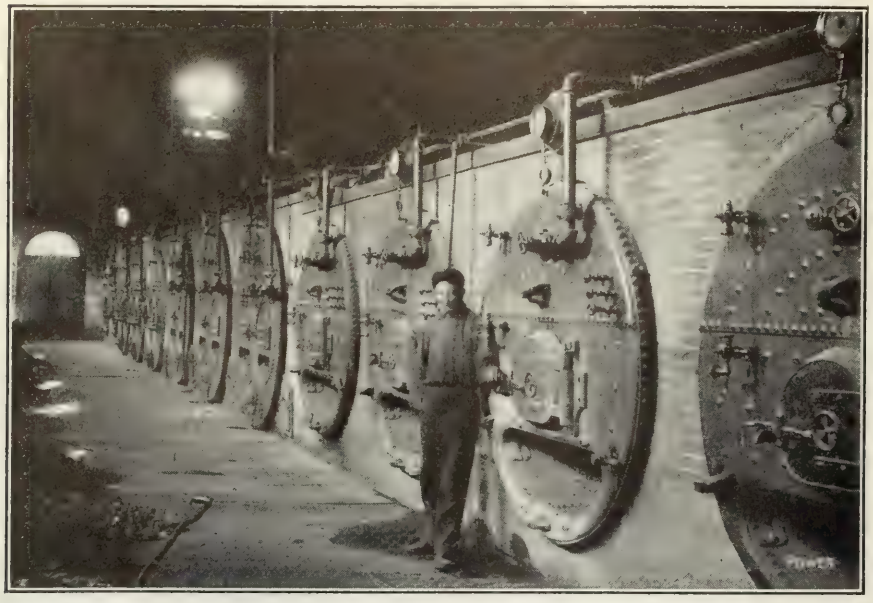
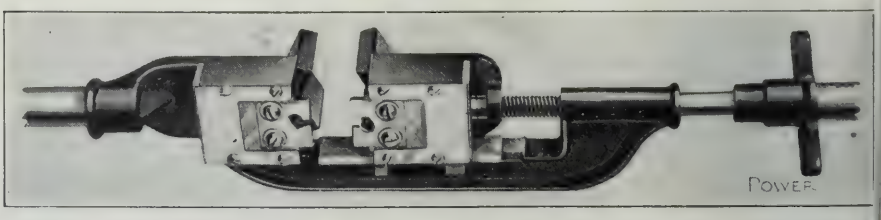


FIG. 1. OLD CENTENNIAL GALLOWAY BOILERS

FIG. 2. WROUGHT-IRON EXPANSION JOINT

Square End Pipe Cutter

This tool severs a pipe the same as a lathe tool cuts off a piece of bar metal. The knives are fed automatically and as they revolve cut a narrow slot around the pipe. The operator turns the tool by two handles similar to using a die stock. The knives make a square pipe end on which the threading dies will easily start. The tool does not split the pipe and saves the time necessary for reaming the inside of the pipe. The form of the knives regulates the depth of the cut. The tool is manufactured by the Borden Co., Warren, Ohio.

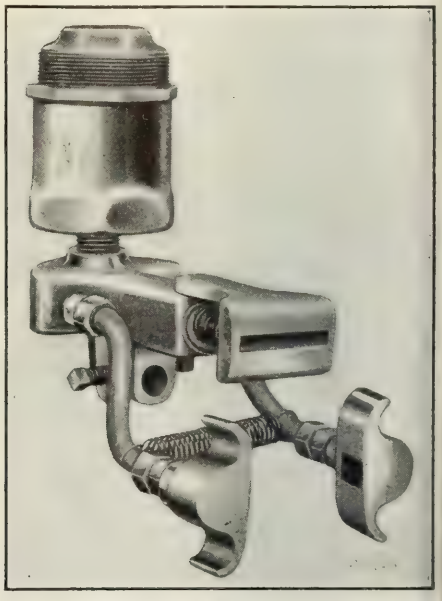


SQUARE END PIPE-CUTTING TOOL

Elevator Guide Lubricator

The "Economy" elevator guide lubricator is built in several sizes. The body is a hollow casting, to which is attached a compressed-air grease cup. An occasional turn of the top nut compresses the air confined in the cup sufficiently to force grease out through the arms of the lubricator to the side rails in a thin ribbon of grease where the guide shoes bear.

The distributors are kept against the face of the rail by springs. The device, illustrated herewith, is manufactured by the Peterson Engineering Co., 50 Church St., New York City.



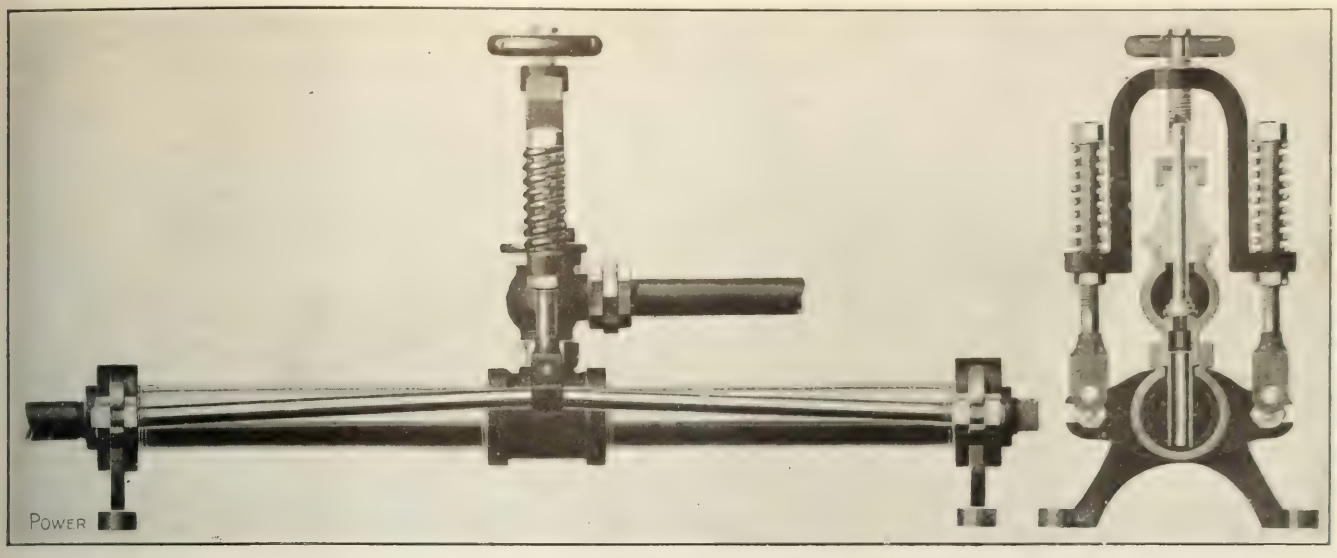
ELEVATOR GUIDE LUBRICATOR

Komo Steam Trap

The body of the trap consists of two tubes of a metal having a high coefficient of expansion. These are joined at the center by a T carrying the valve and are screwed at their outer ends into castings carrying the lugs upon which the trap is set. Each of these castings has an internally threaded opening, one of which is used as the inlet to the trap and the other is plugged.

The end castings carry side lugs or brackets which are connected by the upwardly bowing side rod shown. As they are drawn together by the contraction of the receiving tubes, the side rods are buckled upward with a movement very considerably multiplied by reason of the toggle-joint action raising the valve through the connection shown in the engraving. The inner tube reaching nearly to the bottom of the trap seals it against the escape of steam, the entrance of which expands the receiving tube and closes the valve before all the water has escaped. The springs upon the rods connecting the side rods to the yoke hold the valve to its seat against the steam pressure and permit the trap to be adjusted to different pressures and temperatures; the handwheel at the top is used for blowing out or adjusting.

It is made by the Linton Machine Co., 26 Cortlandt St., New York City.



LONGITUDINAL AND SECTIONAL VIEW OF THE KOMO STEAM TRAP

Dourte Valveless Pump

The horizontal type of Dourte valveless pump shown in Fig. 1 is designed to handle very trashy matter, weeds, or other material being sheared off by the edges of the ports.

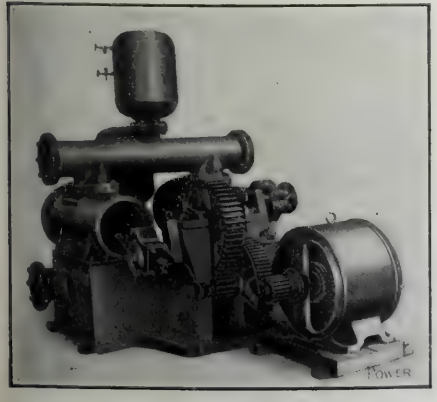


FIG. 1. DOURTE VALVELESS PUMP

The cross-section of one cylinder, Fig. 2, shows its design. A miter gear is keyed on the end of the crank and meshes with a similar gear on the end of the connecting-rod, which is connected to the piston by means of a universal coupling. When the crank revolves it pushes the piston back and forth and rotates it at the same time. The piston is of the trunk type, open at both ends, but has a partition in the middle. The head end has a port which alternately registers with the suction and the delivery ports. The delivery port is in a separate saddle which rides on the piston and is held down by adjustable springs.

In the pump shown, the suction port is cored out of the solid cylinder, but for service where the wear is great, the suction port is cast in a separate saddle which can be easily renewed when worn.

In the illustration, all ports are blanked, but as soon as the pump starts the suction port will begin to open. When the crank is at 90 deg. from the position, the piston port will be exactly over the suction port. If on the dead center and opposite the present position, the ports will all be blanked again. When the crank is straight down, the piston port will be exactly under the delivery port. If the pump is reversed or the piston is set at 180 deg. from its present setting, it will pump from the delivery to the suction; in other words, the flow will be reversed.

This pump is manufactured by the Dourte Valveless Pump Co., Longmont, Colo.

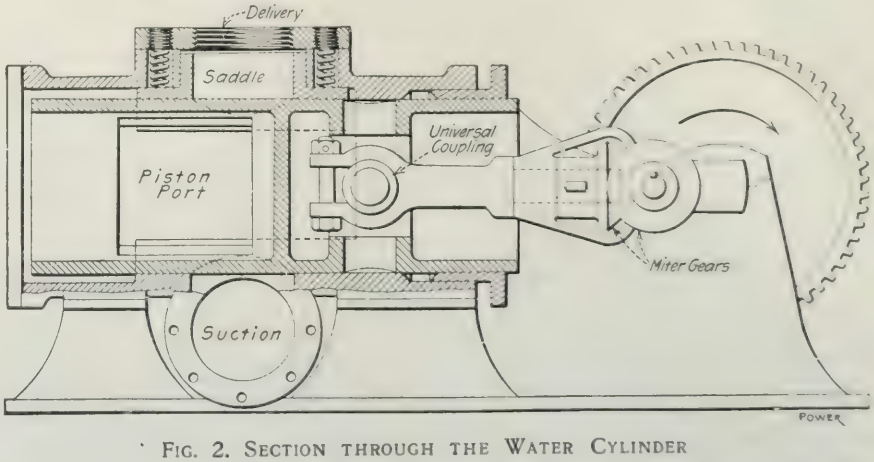


FIG. 2. SECTION THROUGH THE WATER CYLINDER

Wood in Pulleys

Wood is used in many places where it was not thought of before. It was perhaps the split idea that first made the wood pulley popular, and that idea is freely used in metal as well as wood pulleys. But now one finds many wood pulleys where there is but little call for the split. A sample instance is on motors for driving machines. When dynamos and motors were new, wood pulleys were not a part of them; now they are an important part wherever belting is used. There are places where the metal pulley is essential, and will always be used, but where it is not absolutely necessary the wood pulley seems to be getting the call right along.—*The Wood Worker.*

Homemade Automatic Engine Stop

By C. A. Ware

Description of the construction and action of an electrically operated automatic engine stop which can be made in any ordinarily equipped machine shop.

With comparatively simple facilities at hand an automatic engine stop can be made which in the present case was applied to eight 42 and 86 by 60-in. vertical engines, direct-connected to alternators. The stop is electrically operated; a layout of the circuit for one engine is shown in Fig. 1.

The small governor *G* is driven from the tail shaft of the engine; its speed bearing a constant ratio to that of the engine. In case the engine overspeeds the governor balls move outward, due to centrifugal force, and the contact *D* is drawn over until it touches the two contacts and closes the circuit through the trip coil *C* (Fig. 2) which immediately attracts its plunger *P* and by means of the lever shown rotates the hook *H* to the left, after which the arm *A*, being unsupported, is pulled down by the weight *W*, closing the butterfly valve and shutting off the steam. When the arm *A* swings downward the small switch *S*, shown mounted on the upper side of the arm, moves away from the two spring contacts which it normally touches while

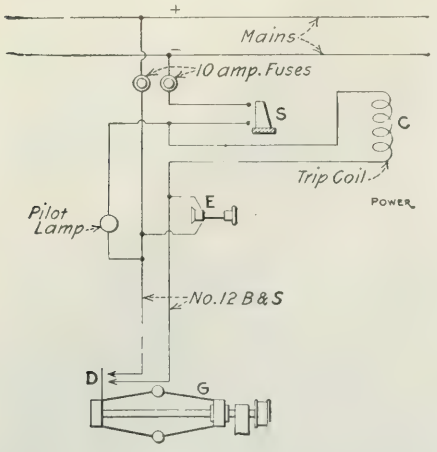


FIG. 1. LAYOUT OF ELECTRICAL CIRCUIT

A is held up and the valve is open. As soon as the switch leaves the contacts the current is interrupted at this point to prevent arcing; as the engine slows down the governor contact *D* is gradually moved away from its contacts.

When the hook *H* trips, the arm *B*, connecting with the pull rod as shown in Fig. 2, moves downward, and to avoid any opposing force the head of the pull rod should be slotted, allowing the pin in the arm to move freely downward.

Referring to Fig. 1, the wires can be connected by means of a double-pole, double-throw switch either to the direct-current exciter busbars, or to the storage battery, each of 110 volts. The pilot lamp and the cutout switch *E* are mounted on a column close to the throttle valve.

When starting up and when shutting down the engine, the switch *E* is pulled to operate the butterfly valve and keep it working freely and show that everything is operating properly. As a number of these cutout switches may be located anywhere about the building, the pilot lamp should always be lighted as it gives a visual indication by going out if any failure of the current occurs. By making the connections as shown in Fig. 1, any failure of the switch *S* to make contact would also cause the pilot lamp to go out.

All supporting brackets, levers, links, etc., used in this device were made in a machine and blacksmith shop having only ordinary equipment. The contacts of the switch *S* were made of two pieces of thin phosphor bronze, as this material retains its elasticity. These contacts and also the piece of sheet brass in contact with them are mounted on pieces of hard fiber which insulate them from the supporting metal parts. The coil *C* and its frame, while not homemade in this case, could be made without any difficulty. The frame could be made from ½-in. flat iron bent U-shape with a flat piece to close the end. Care should be taken to get the joints as true and close fitting as possible.

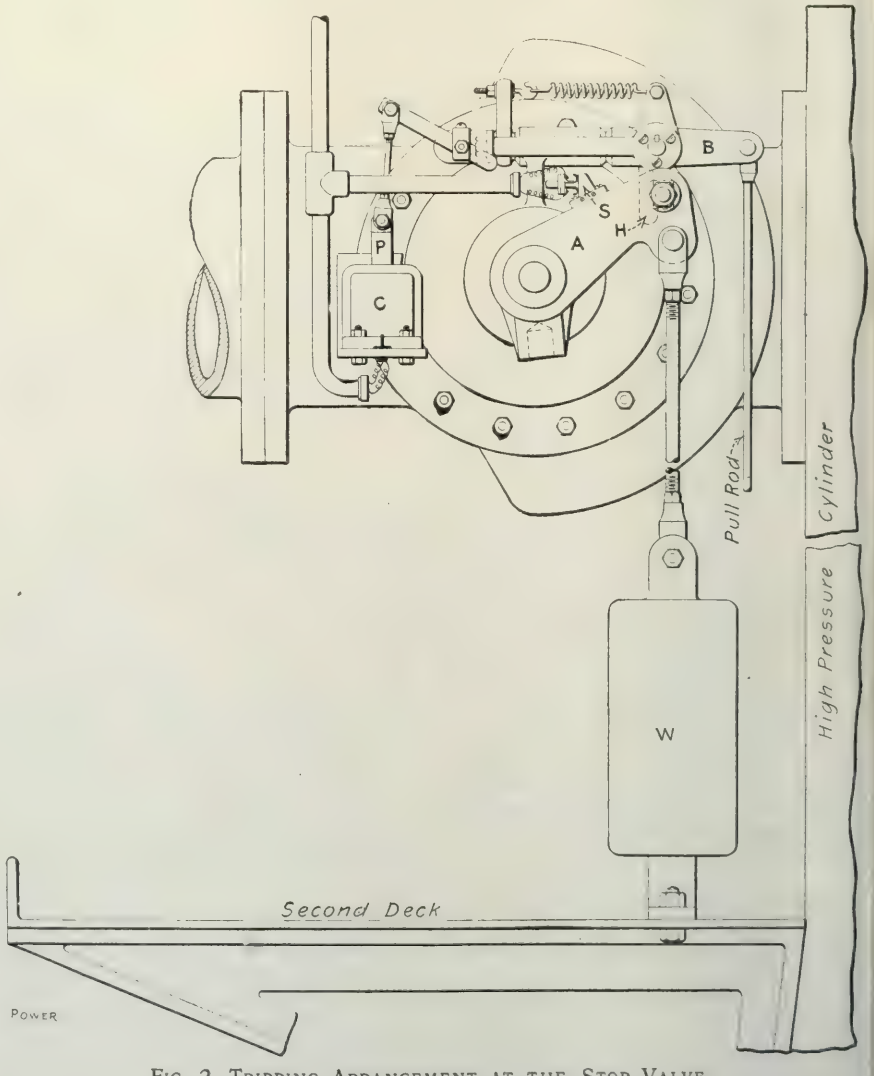


FIG. 2. TRIPPING ARRANGEMENT AT THE STOP VALVE

The coil, about 4 in. long, can be mad by winding with 2 lb. of No. 22 B. & S gage magnet wire (this amount fo 110 volts). The plunger *P* is 1¼ i in diameter and is 3 in. long. A bras sleeve is put inside the coil and extend up flush with the surface of the iro frame. This prevents the plunger fro

sticking to the frame, as it would do if there were actual contact between the two; the sleeve also protects the inside of the coil. A soft steel plug should be fitted to the lower end of the magnet frame and extend up through the coil for about 1 in. to provide a stop for the plunger. A thin disk of brass or copper should be fastened to the upper end of the plug to prevent actual contact with the plunger and the sticking together which would result.

The circuits are wired with No. 12 B. & S. wire in ¾-in. loricated conduit. The governor was purchased from a firm making a specialty of manufacturing them for this purpose.

BOOKS RECEIVED

FARM GAS ENGINES. By H. R. Brate. The Gas Engine Publishing Co., Cincinnati, Ohio. Cloth; 195 pages, 5x7 in.; illustrated; indexed. Price, $1.

BUILDING AND FLYING AN AEROPLANE. By Chas. B. Hayward. American School of Correspondence, Chicago, Ill. Cloth; 142 pages, 5½x8½ in.; 52 illustrations; tables; indexed. Price, $1.

NEW PUBLICATIONS

PERKINS TABLES, for use in Calculations for safe working pressure on boilers. By Lyman B. Perkins. 360 pages, 5x7½ in., thin paper, flexible leather covers. Published by the author at 38 Huntington St., Hartford, Conn. Price, $5.

The author, a graduate of Annapolis, has been employed for over 25 years by the Hartford Steam Boiler Inspection & Insurance Co., and much of this time has been occupied in making calculations relating to the strength of boilers. The tables are the outcome of the demands and developments of this experience.

The volume starts with a table of the bursting pressure of the hollow cylinder without a joint, and covers shells from 26 to 96 in. in diameter, from ¼ to ¾ in. thick and having tensile strengths from 45,000 to 62,000 lb. There are 11 pages of this table. The bursting pressures there given must be multiplied by the joint efficiency to get the bursting pressure, and most of the book is devoted to joint efficiencies. In this table can be found the efficiency of any practicable joint, and special tables are given for the conditions imposed by the Massachusetts Board of Boiler Rules. In addition are tables of head bracing, of segmental areas, collapsing pressures of furnaces and flues, directions for reinforcing openings, staying flat surfaces, sizes of pipe and boiler tubes, decimal equivalents, etc., and many useful formulas.

In its preparation Mr. Perkins has done a stupendous amount of work which, having been once thus reliably done, will not need to be done over and over again by boiler designers and inspectors who have access to this volume.

CENTRAL STATION HEATING. By Byron T. Gifford. Heating & Ventilating Magazine Co., New York City. Leather, 208 pages, 5½x8 in., illustrated, numerous tables. Price, $4.

This book sets forth the advantages of central-station heating and is written from the standpoint of one interested in furthering this line of work. The chapters on rates, pipe-line construction, business management and franchises are excellent and have undoubtedly been drawn from the extended experience of the author. The data on radiation losses for different classes of conduit and insulation for street mains under varying conditions are also commendable. The factors are so given that the radiation losses for the various methods of insulation can be readily determined with fair accuracy.

The book is of doubtful value to designers of central heating plants wishing more than approximation, excepting the portion devoted to pipe-line construction. This is due to using arbitrary constants which, although excellent for the author's practice, might lead others to err when designing for different conditions.

The curves on pipe sizes combine all factors and allowances involved in the drop in temperature and loss of friction head. This makes them inadvisable for general use unless the complete formulas and factors are known. As nearly as the reviewer can check the curves for water, the friction heads are low. For a 20-deg. drop with a given output and drop in friction head, other authors give a discharge that would make the drop 26 deg. for the same size of main.

The author intimates that considerable difficulty has arisen from deterioration of mains, and mentions the use of comminglers and the furnishing of domestic water supply. It is poor practice to draw domestic water from heating mains, for the air contained in the make-up water tends to cause rapid deterioration. Comminglers would facilitate the removal of the air but might not wholly obviate the trouble.

It is stated that the friction of the moving water in the mains develops heat, thus returning some of the mechanical energy used in circulation. Also that heat is lost at the station, due to the increased pressure on the mains necessary to circulate the medium. If the exhaust of the pump is utilized in heating the water it is hard to see how increased pressure head on the system at the station would cause any particular loss on the water system other than that due to the steam efficiency of the pumping apparatus. This might be true on a steam system inasmuch as an increase in pressure is accompanied by a corresponding rise in temperature. Increased pumpage causes lower average temperatures of the circulated water, increased heat transmission through the heater and radiating surface and less drop on the system, all of which are advantages.

In considering the losses of water mains the radiation losses are calculated properly, but the constant 0.046 for cost of pumping is variable, depending on the steam consumption of the prime mover, which in this case would be about 60 lb. per horsepower-hour. The amount of heat returned to the main should be figured with the mechanical equivalent of heat in each case and the steam consumption of the prime mover determined and the heat developed from friction deducted. At 26 lb. of steam per horsepower-hour the loss in the pump will just balance the heat returned from friction. Any higher steam consumption requires an addition to the losses in mains. This will show high velocities and small mains to be advantageous.

The pipe-line design for hot-water systems should be arranged so that the distance will be balanced as well as the friction heads on the different circuits, thus doing away with individual regulation of outlets and the use of choke valves. It makes automatic heat control unnecessary and enables all sections to be regulated by the change in water temperature and vacuum at the central power house.

An arrangement of mains is shown that would require more or less throttling in spite of the best hydraulic calculations unless very low velocities and large mains were used. By making the supply and return flow in the same direction this difficulty is avoided for all velocities and the increased cost would be insignificant compared with the advantage gained.

The author considers that a surface condenser or closed heater cannot be made to condense all the steam with the same facility as a barometric condenser and an open heater. This is not true for a closed water system which can be operated under vacuum equally well with a barometric or surface condenser. The water to condense the excess steam passes through the condenser proper and should be kept out of the heating system. Its temperature can be regulated exactly by the degree of vacuum, the temperature of the outboard water of the heating system approaching that of the vacuum within 2 or 3 deg. in a properly designed plant. This requires large pumpage, high velocities and small drop in temperature to be successful.

When the heating system is used solely as a condenser the degree of vacuum is merely incidental and interferes with the regulation of the heating temperature as the water becomes overheated and at the same time the vacuum is reduced.

Hot-water boilers are given attention, and, although used in some plants, they are not as easily regulated as high-pressure steam boilers and a live-steam heater. In a by-

product plant the hot-water boilers would be an unnecessary investment, to say nothing of being idle in summer. It is this unnecessary investment in boilers that makes the central heating plant so attractive in connection with the central power plant.

In the discussion of some chapters to the exclusion of others it has not been the intention to detract from the excellent matter contained therein, but rather to call attention to some of the points differing from the practice of the reviewer. As before stated, the subject has been approached more from a business than an engineering standpoint, and this addition to the limited literature on the subject should succeed in its mission of awakening station managers to the advantages of the combined central heating and power plant.

Universal Craftsmen's Convention at St. Louis

On June 6 to 9 the tenth annual convention of the Universal Craftsmen Council of Engineers was held at St. Louis, Mo.

The entire second floor of the Planters' Hotel was engaged by the convention and tastefully decorated. The exhibitors' booths were in the dining room, and the delegates met in the large reception hall, adjoining which were the meeting rooms of the national secretary and treasurer and executive committee. The convenience of these arrangements reflects great credit upon the local committee, and particularly on Chairman Samuel J. Hunt.

There were 137 delegates in attendance, and considerable important business was transacted. A charter was granted to the German engineers of Greater New York, and a new council will be formed in which the work will be carried out in German. The financial condition of the organization was reported to be first-class.

The opening exercises of the convention took place on Tuesday morning. Chairman Edwin S. Hallett introduced the Rev. Brother Early, who pronounced the invocation. The address of welcome was made by Samuel J. Hunt. Walter J. Cadwell, past grand worthy chief, responded, after which an address was given by Grand Worthy Chief Thomas H. Jones. After the announcement the committees were appointed, and the meeting adjourned.

On Tuesday afternoon the visitors, in automobiles, visited Art Hill and the museum where the official photograph was taken. In the evening the ladies went to the Demar Gardens.

On Wednesday afternoon the delegates made a tour of the Anheuser-Busch Brewery, and in the evening were the guests of the Steam Engineers Club of St. Louis. In the evening the ladies were entertained at Forest Park Highlands.

Delegates at the Meeting of the Universal Craftsmen Council of Engineers, St. Louis

On Thursday afternoon the company enjoyed an excursion on the Mississippi and in the evening a vaudeville entertainment in the convention hall at the hotel.

During the convention, Mrs. O. N. Pomeroy was presented a cut-glass set, Mrs. Thomas H. Jones a silver service, and Worthy Chief Thomas H. Jones a past chief's jewel.

The election of grand officers resulted as follows: James U. Bunce, worthy chief; H. C. Senn, assistant worthy chief; William Armstrong, secretary; N. J. Burdick, treasurer; William J. Weyker, warden; Charles F. Siegrist, guard; P. H. Early, chaplain; H. E. Terry, chairman of the board of trustees; Samuel J. Hunt, trustee; F. W. Townsend, trustee.

Boston, Mass., was selected for the 1913 convention.

Exhibitors

Garlock Packing Co., Jenkins Bros., Dearborn Drug & Chemical Works, Reeves, Skinner Machinery Co., Lunkenheimer Co., Power, John E. Angell & Co., Armstrong Cork Co., Keystone Lubricating Co., Capen Rubber & Belting Co., Schaeffer & Budenberg Manufacturing Co., V. D. Anderson Co., Triumph Ice Machine Co., Western Valve Co., Lagonda Manufacturing Co., Walter L. Flower Co. Wagner Electric Co., A. Leschen & Sons Rope Co., Frank Adams Electric Co. C. E. Squires Co., Broderick & Bascom Rope Co., M. H. Foundry & Manufacturing Co., Elliott Co., Diesel Engine Co. McLeod & Henry Co., International Acheson Graphite Co., George T. Mathews & Co., William Kemp, Arrow Boiler Compound Co., Indestructible Fan Blade Co. H. W. Johns-Manville Co., Bogardus Co. Price Hill, Heine Safety Boiler Co., F. C. Lemire, Liberty Manufacturing Co. Mound City Oil & Supply Co., J. Brockman Manufacturing Co., L. M. Sperry A. S. Doxey, Ohio Injector Co., Hahn & Mayer.

Illuminating Engineers' Convention Program

At the sixth annual convention of the Illuminating Engineering Society, at Niagara Falls, Ontario, Sept. 16 to 19, a report will be presented dealing with recent progress and developments in the lighting industry both in this country and abroad.

The committee on nomenclature and standards will present a report dealing with certain definitions and terminology of illuminating engineering. "Steel Mill Lighting" will be the title of a report by C. J. Mundo, chairman of the committee on illumination of the Association of Iron and Steel Electrical Engineers.

F. W. Goodenough, chairman of the council of the (British) Illuminating E

gineering Society, will read a paper, "High-Pressure Gas Lighting." The program as thus far arranged includes the following papers: "The Status of High-Pressure Gas Lighting," George S. Barrows; "Recent Developments in Gas Lighting," R. F. Pierce; "Indirect and Semi-Indirect Illumination," T. W. Rolph; "Recent Developments in Series Street Lighting," C. P. Steinmetz; "Research Methods," E. P. Hyde; "The Problem of Heterochromatic Photometry and a Rational Standard of Light," H. E. Ives; "Reflection from Colored Surfaces," Claude W. Jordan; "Diffuse Reflection," P. G. Nutting; "A Study of Natural and Artificial Light Distribution in Interiors," M. Luckiesh; "The Physiology of Vision," T. A. Woodruff; "The Efficiency of the Eye Under Different Systems of Illumination," C. E. Ferree (this paper will be a report of a research carried on for the American Medical Association); "A Proposed Method of Determining the Diffusion of Translucent Media," E. L. Elliott; "Illumination Charts," F. A. Beuford; "The Determination of Illumination Efficiency," E. L. Elliott; "An Absolute Reflectometer," P. G. Nutting; "Colored Values of Illuminated Surfaces," Bassett Jones, Jr.; (this subject will be presented in the form of a series of experimental demonstrations).

One session is to be devoted to discussing miscellaneous phases of illuminating engineering, to bring forward certain features not covered by the formal convention papers.

Kelvin Memorial Window

It is proposed by the engineering societies of Great Britain and the United States that a memorial window to the late Lord Kelvin (Sir William Thomson) be erected in Westminster Abbey.

American engineers are invited to contribute to this tribute of respect to the foremost scientist of his day and an engineer distinguished in his application of scientific knowledge to enterprises of world-wide importance. Individual subscriptions need not exceed $10, but those who wish may increase their contribution.

Coutts & Co., 40 Strand, London, W. C., are the treasurers of the fund.

Hot Bearing Stops Trial

A hot truss-bearing on the port shaft of the torpedo-boat destroyer "Henley" halted its standardization trial on July 20. The destroyer had been over the course off Rockland, Maine, 21 times when the mishap occurred, making the runs at speeds ranging from 12 to 29 knots an hour. The last mile was made in 2 min. 4 sec., which would give about the contract speed of 29½ knots.

Halfway down the course on the last of the top-speed runs one of the truss-bearings was seen to be smoking, and the ship was immediately headed for Rockland. In all other respects the destroyer made a fine showing.

Flywheel Explodes in Flouring Mill

A flywheel explosion occurred at the Hull & Draper flour mill, Salem, Ill., July 16, a few minutes before shutting-down time. The engine was an old slide valve fitted with a throttling governor. The explosion was caused by excessive speed due to the governor belt breaking. The wheel was about 10 ft. in diameter and ran at a normal speed of 75 r.p.m. The broken spokes and rim showed flaws in the metal. One piece of the wheel weighing about 800 lb. was thrown about 400 ft. The hub remained on the shaft with the spokes broken off close.

The engineer was not in the engine room at the time of the explosion. He did the firing and helped in the mill, receiving a wage of $6 per week.

Minneapolis Electric Co. Under New Management

Active management of the Minneapolis General Electric Co., which was purchased from Stone & Webster some time ago, was assumed, Aug. 1, by H. M. Byllesby & Co. Vice-President Arthur S. Huey personally superintended the transfer. Gen. George H. Harries, for some time president of the Louisville Gas Co. and vice-president of the Consumers Power Co. and of the Minneapolis General Electric Co., has been assigned general supervision over the property. Samuel Kahn took charge as acting manager to serve temporarily or until a permanent manager is appointed. Mr. Huey announced that the company would proceed to develop 35,000 hydro-electric horsepower on the St. Croix River above the present 20,000-hp. development at Taylor's Falls. He stated also that the company contemplates further water-power development on the Mississippi River amounting to approximately 80,000 hp., which would give a total of not less than 160,000 hydro-electric horsepower, including several smaller developments, available to Minneapolis and St. Paul and vicinity. The properties at Minneapolis and St. Paul will be connected by transmission lines to the water powers of the Consumers Power Co., at Cannon Falls, and Mankato also will be tied in by a transmission line running south from St. Paul.

Dartmouth (Eng.) to Honor Newcomen

Desirous of honoring the memory of Thomas Newcomen, who first worked out the idea of a piston in his atmospheric engine, the pioneer of the present locomotive and stationary steam engines, the citizens of Dartmouth, England, propose to establish in his honor a museum in that town. If the funds collected prove insufficient to maintain a museum, an obelisk is suggested.

Many prominent Englishmen are in sympathy with the project and have subscribed liberally.

Cotton Manufacturers Meet

The semiannual meeting, No. 93, of the National Association of Cotton Manufacturers, will be heid at the Hotel Griswold, New London, Conn., Sept. 9 to 11, 1912, Governor Baldwin will deliver the address of welcome.

The most important papers to be presented are: "Cotton Growing in India," "Cotton Shipments and Reclamations," "Dust-proof and Water-proof Mill Flooring," "Economic Use of Lubricating Oils," "Humidifiers," "International Tariff Relations," "Lithographic Method of Cotton Printing," "Present Day Relations between the Foreign-born Operative and the Mill Management," "Raising Cotton in California," "Reports of Committees," "Selection of Sites of Cotton Mills," "Spinning Values of Different Grades of Cotton," "Tape Driving for Spindles," "Wastes in the Cotton Manufacture," "Workmen's Compensation Laws in the United States."

SOCIETY NOTE

During the summer months, every Tuesday evening at the Modern Science Club, Brooklyn, will be a social event.

PERSONAL

James T. Whittlesey, chief engineer of the Public Service Electric Co., Newark, N. J., has resigned to engage in consulting practice in San Francisco, Calif.

R. L. Brunet, industrial engineer of the Public Service Electric Co., Newark, N. J., has been appointed electrical and gas engineer of the department of public works, Providence, R. I.

D. D. Kimball, 15 West Thirty-ninth St., New York City, has been appointed to represent the American Society of Heating and Ventilating Engineers as chairman of the Section on Ventilation of the Department of Industrial Hygiene and Sanitation of the American Museum of Safety, 29 West Thirty-ninth St., New York, N. Y.

Moments with the Ad. Editor

F. A. Shoemaker, M. E., of Buffalo makes a specialty of designing power, light and heating plants.

Where one engineer deals with a single problem, he deals with a hundred of them.

Naturally he gets a broad outlook on the whole question of power-plant installation and maintenance.

And if he, who may be considered "authority" on the question, gets from Power's Selling Section the real value and help that he does, does it not follow that *you* will find there the answers to many questions that confront you?

This is worth reflection on your part and so is Mr. Shoemaker's letter which follows:

In reference to your article "Is there Anything *Not* Advertised In Power That *Ought* To Be?" I beg to state that as I have made a business of power plants as a Mechanical Engineer for a great many years and at all times have read POWER with a great deal of interest, I assure you that I do know of a great many things that *are* advertised in POWER that ought to be, and no doubt a number of things that are *not* advertised that should be.

One thing that I can say that may cheer you up some is the fact that I very seldom dictate a specification for a power plant or heating plant but what I refer to the advertisements in POWER for a number of articles I wish to use, as I am very busy and can more quickly find the address of the manufacturers by referring to the advertisement in POWER than to wade through cabinets of catalogues even though I keep those catalogues filed in perfect order and strictly classified. In this way POWER has been of a great benefit to me, also from practical experience with the merit of a very large number of appliances in the make-up of a power plant, I find the live ones and good ones avail themselves of the opportunities offered in your advertising pages.

There may be a large number of very good articles that a person or company have not been putting on the market very long who feel they can not stand the cost of high class advertising, and still another class of manufacturers who think they have been in existence so long that they do not have to advertise, but my experience as a power plant designing engineer has conclusively proven to me that both of these classes of people are missing a great many sales by not keeping their goods before the public. I know of several manufacturers who do not even have a man on the road who have built up enormous businesses and made large profits within the last fifteen years by carefully selecting high class advertising, and I cannot see how any manufacturer with a good article can expect to reach the largest number of customers interested in power plants without advertising in POWER.

A very large part of orders placed through my specifications every month have been such things as I have taken note of in your advertising spaces and descriptive matter in the body of your magazine

I feel that I can congratulate you as well as mysel on your comprehensive advertising and the benefit I have derived from the same.

* * *

After all is said and done, the ma who consistently reads Power advertis ing and buys advertised goods is goin at his work along the right lines—

He's buying on a satisfaction-assure basis.

POWER

Vol. 36 NEW YORK, AUGUST 27, 1912 No. 9

Available Power Plant Information

When something goes wrong with our "innards" we generally call in the doctor. He looks over the "plant," gets a line on the symptoms, diagnoses the case and prescribes the proper remedy—sometimes.

Professionally, friend Doc. is, in a manner of speaking, a sort of kinsfolk to the operating engineer. But he has this advantage, his "machines" are built pretty closely on the one model. In a dozen power plants there may be twelve differently designed or constructed engines, boilers, pumps, etc. By making a specialty of one line of work—say, erecting boilers—a man may become familiar with most of the types manufactured, but no one operating engineer can know offhand the constructive details of the plant equipment in half-a-dozen medium sized plants.

While many engineers stay in one plant long enough to become familiar with it, unfortunately all do not have that stamina. Perhaps restlessness may lead an engineer into new fields where he does not always increase his store of knowledge. In making these changes, he is "up against it" as everything is usually set up and in operation; he has to step in and keep things going.

While he knows the general operating principles of the engines or boilers he does not know all their construction details, especially their insides. Suppose they begin to show symptoms of ailment. If he has drawings he can study their construction and reason out the probable cause of trouble and apply the remedy. If not he turns "surgeon," taking things apart and hunting for the trouble.

When new machines are installed blueprints, sectional drawings and instruction cards are generally furnished by the manufacturers. These, perhaps, have been turned over to the engineer in charge. He takes care of them for a while, after which they are probably lost or go with him when he leaves.

In some plants the drawings and general instructions are set up in glass-covered frames near the machines. This is a good plan, but the best way is to have all drawings, instructions and notes of changes, alterations, etc., carefully filed in the general office. Then the information is always at hand. Manufacturers are always glad to furnish such matter as will be helpful to the successful operation of their products.

Maps or plans of piping systems, especially underground, should always be available. Then they can be studied so that in case of an accident or an emergency the man in charge will know directly what to do.

Measuring Movement of a Steam Main

By W. N. Stevens

By means of an instrument called a flexometograph, the movement of a point upon a steam main is recorded upon a 10 to 1 scale. The movement of a flange near the end of a 16-in. main 230 ft. long, securely anchored in the middle, was found to be seven-tenths of an inch under working conditions.

In reporting the hearing at Washington by the Navy Department on the "U. S. 1912 Standard" for flanges and flange fittings, POWER referred to experiments by the engineers of the New York Edison Co. to determine the movement of a steam main under working conditions. Through the courtesy of the company the writer is permitted to lay before the readers of POWER a description of the apparatus used and some of the results obtained.

To meet the special requirements of this test, an instrument called a flexometograph was designed. This is shown in front and end views in Fig. 1 and consists of a drum *D* revolved once in 24 hours by a clock *C*. It is carried upon a steel frame work which rests upon some fixed support, as a bracket upon the wall of the building. The flange or other point, the movement of which is to be registered, is attached to the turn buckle *T*, which connects the ends of a wire passing around the hub of the pulley *P* and the idler *p*. The pulley *P* has a diameter ten times that of its hub, and the motion is transmitted by the cord which passes around its periphery and the idlers *F* and *f*. To this cord is attached the slide *S*, which carries the recording pen and has a motion ten times that of the flange, the movement of which is to register.

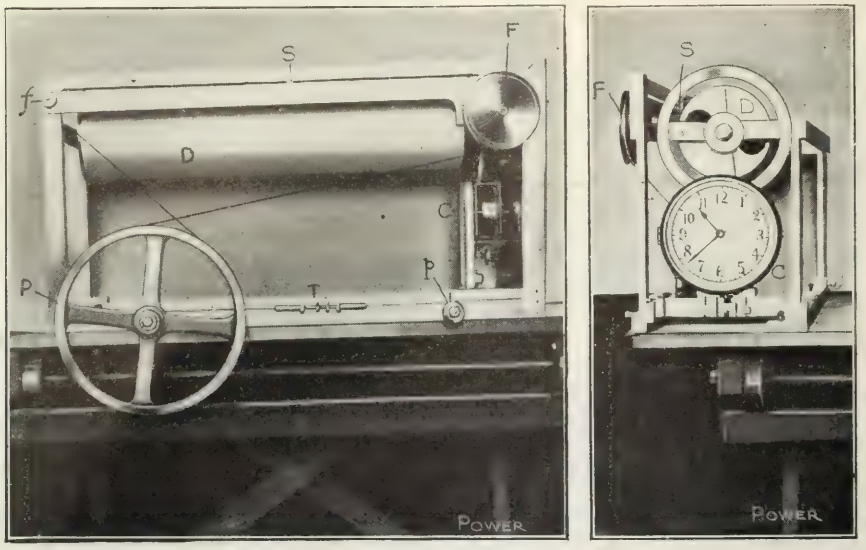


FIG. 1. THE FLEXOMETOGRAPH

The steam header to which this was applied is some 230 ft. long and is shown in elevation in Fig. 2. The location of the flexometograph is indicated at *F*. The pipe is securely anchored at the center and is supported at intervals on rollers to permit of free movement incident to expansion and contraction without undue strain on the flanges and bolts along the line of the header.

Fig. 3 shows the record for Tuesday Jan. 29, 1907, which is characteristic of those obtained on other days. Upon the original record the heavy horizontal rulings are 1 in. apart, so that each of them indicates a movement of $\frac{1}{10}$ of an inch of the point to which the instrument is attached, while the intermediate rulings indicate hundredths of an inch. The dotted line indicates the variation of temperature in the main at the same time.

As the pipe was under steam pressure throughout the test, the curve shows the movement during the operation of the power house. No record was taken of the movement from a cold pipe to a pipe under steam. The pressure varied from 182 to 200 lb., superheat being used as indicated by the temperature.

It will be seen that the point *S* on the steam main was never at rest. Its movements are continuous throughout the life of the power house. As the outlet flanges of the tees in this main remain parallel to the axis of the main, the leads to the boilers and engines must not only be bent during the movement, but are given a compound bend due to their own expansion as well as that of the main. It is obvious that these leads should not be stiff, but must be as flexible as possible; otherwise the strain on the tees will be enormous. It is most important that no extra-heavy pipe be used in these leads. Extra-heavy pipe defeats the ve

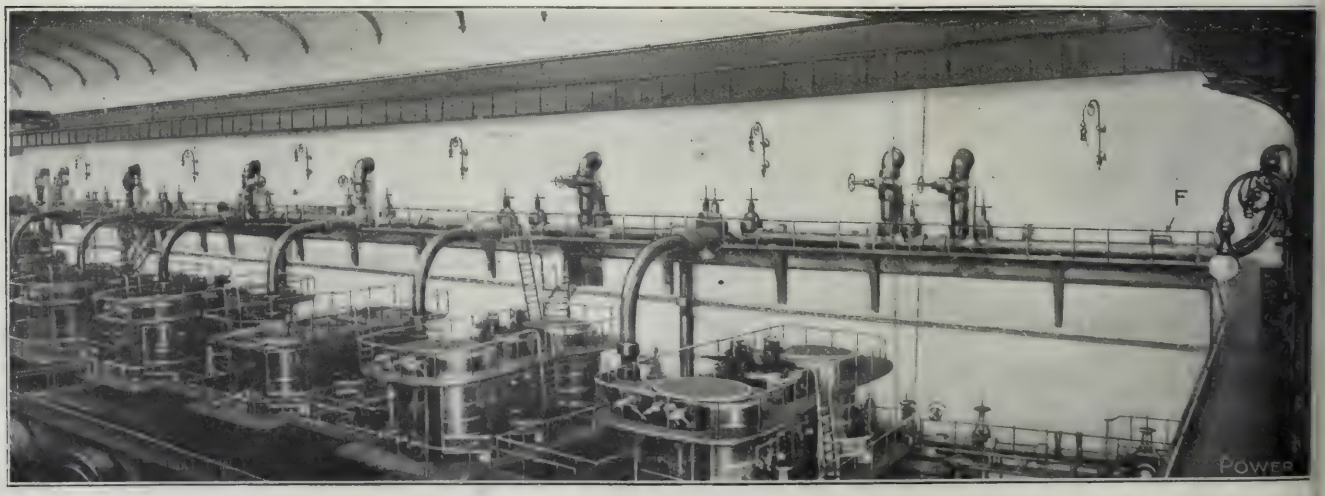


FIG. 2. STEAM HEADER 230 FT. IN LENGTH AT WATERSIDE STATION OF THE NEW YORK EDISON CO. FLEXOMETOGRAPH APPLIED AT

purpose for which it is employed. It lowers the factor of safety of the tee below the factor of safety of standard weight pipe. This is based on the pipe being made with a Van Stone flange or a welded-on flange and the tee being an extra-heavy cast-steel fitting.

Eight-inch standard weight wrought-steel pipe requires 4000 lb. pressure per sq.in. to burst it. The factor of safety is 20 if the steam pressure is only 200 lb. It is therefore amply strong. With the fittings, however, it is quite different. The never ceasing movement of the pipe

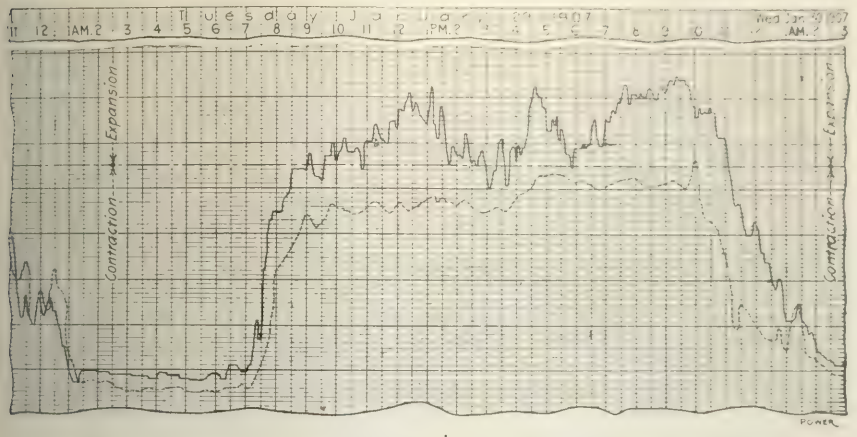


FIG. 3. FLEXOMETOGRAPH RECORD OF PIPE MOVEMENT AND TEMPERATURE VARIATION

subjects them to strains that may be many times those due to the steam pressure, and they are bound to crystalize in time. They must be very heavy and strong and have no abrupt changes in the thickness of metal if they are cast fittings.

Where a high degree of superheat is employed, the movements are correspondingly greater. It is therefore submitted that cast-iron fittings are not good enough for superheated steam. They should be steel. Steel castings can be obtained that are free from shrinkage strains and are sound.

If one could obtain forged steel fittings, they would be still better. The daily cycle which this point *F* follows also shows the importance of so supporting steam piping, that its movements are freely made and that it must be securely anchored at the center of its length.

"How much experience have you had with boilers?" asked the chief engineer of the new applicant.

"Oh, I've been around them quite a lot."

"Of course you know how to handle the injector?"

"Oh, yes."

"What is an injector?"

"Why, the injector, as I understand it, is a little thing that screws on the side of the boiler, with a rubber hose that leads into a barrel."

The Myrawatt, a New Power Unit*

BY H. G. STOTT AND HAYLETT O'NEILL

The object of this communication is to introduce a new unit of power which, if adopted, will afford a basis for comparing all converters of energy, thermal and mechanical, and also will be international in its character, as it is merely a new multiple of the watt.

In American and European practice, there are many empirical units, the use of which is restricted to a distinct territory. A few of the more important are horsepower, boiler horsepower, kilowatt, cheval à vapeur, pferde-kraft or pferdestärke and poncelet. Obviously an engineer in attempting to compare data from a foreign country, is compelled to face a confusion of terms, which usually can be made intelligible only by laborious calculation.

Again, in the United States there are such units as boiler-horsepower and horsepower, which, while similar in sound, have no logical connection; and one has yet to find where the "horse" comes in.

With the rapid development in electrical measuring instruments, and, until recently, a corresponding lack of development in steam-flow measuring instruments, the term kilowatt has become more and more used as the one unit of power output.

The term became a necessity with the growing favor of steam turbines, and all direct-connected units where it is impossible to measure accurately the mechanical and the electrical power separately.

To form a connection between the boiler or producer output, the term "myrawatt"—derived from the Greek "myria," meaning ten thousand, and the term watt—is proposed.

For the purpose of standardization, the British thermal unit, 1/180 of the heat required to raise one pound of water from 32 deg. F. to 212 deg. F. and the equivalent evaporation from and at 212 deg. F. (970.4 B.t.u.) is used (Marks and Davis).

From this, the following equivalents are obtained:

1 foot-pound	= 0.001285	B.t.u.
1 [illegible]	= 2.547	B.t.u. per hr.
1 [illegible]	= 2.510	B.t.u. per hr.
1 pferde-kraft	= 2.510	B.t.u. per hr.
1 poncelet	= 3.20	B.t.u. per hr.
1 kilowatt	= 3.415	B.t.u. per hr.
1 boiler horsepower	= 33,479	B.t.u. per hr.
1 myrawatt	= 34,150	B.t.u. per hr

The last two are practically the same, differing by only 2 per cent. The usual practice is to rate water-tube boilers on the basis of one boiler-hosepower per 10 sq.ft. of heating surface.

COMPARISONS IN EFFICIENCIES AND RATES OF OUTPUT WITH VARIOUS TYPES OF ENERGY CONVERTERS IN TERMS OF THE MYRAWATT

1. Boiler output:

Nominal rating = 600 boiler h.p.

Total draft head. in. water gage	Boiler h.p.	Myrawatts
0.589	817	800
1.170	1174	1151
1.730	1375	1348

2. 5000-kw. engine:

Kw. Output	Lb. steam kw.-hr.	B.t.u. kw.-hr.	Kw. output Mw. input	Per cent thermal efficiency overall
3100	18.2	21,500	1.59	15.9
4977	17.2	20,160	1.70	17
6772	18.5	21,600	1.58	15.8

3. 5500-kw. high pressure turbine:

Kw. Output	Lb. steam Kw-hr.	B.t.u. Kw-hr.	Kw. output Mw. input	Per cent thermal efficiency overall
2283	20.16	23,100	1.48	14.8
5350	16.86	19,160	1.78	17.8
8183	16.39	18,450	1.85	18.5

4. 15,000-kw. engine—low-pressure turbine.

Kw. Output	Lb. steam Kw-hr.	B.t.u. Kw-hr.	Kw. output Mw. input	Per cent thermal efficiency overall
8,347	13.34	15,740	2.17	21.7
11,240	13.19	15,660	2.18	21.8
16,172	15.07	17,500	1.95	19.5

5. 56-in. low head water-turbine:

Brake h.p.	Brake kw.	Kw. output Mw. input	Conversion eff per cent overall
294	219	7.83	78.3
283	211	8.41	84.1

6. Steam plant efficiency:

Lb. coal per kw-hr.	=	2
B.t.u. per lb. coal	=	14,250
B.t.u. per kw-hr.	=	28,500
Kilowatts per myrawatt input to boilers	=	1.2
Plant efficiency	=	12 per cent

7. 2000-h.p. Koerting gas-engine direct-connected blower

Kw. Output	Cu.ft. gas Kw-hr.	B.t.u. Kw-hr.	Kw. output Mw. input	Per cent thermal efficiency overall
5206	146	14,220	2.4	24

With modern plants, notably those in marine service, operating from two to five times this rating, the ordinary method of determining nominal boiler capacity could be stretched 2 per cent. without materially affecting the present rating; i.e., the boiler might be rated at 34,150 instead of 33,479 B.t.u. per hour for each 10 sq.ft. of heating surface.

The myrawatt as a unit of boiler or producer output, and correspondingly a unit of input to all kinds of dynamical machinery, is fixed in value by the watt,

*Presented at the Boston Convention of the Am. Inst. Elec. Engrs.

and by its very sound gives a clue to its meaning.

To compare efficiencies of direct-connected units and eliminate the various factors of quality of steam, pressure and vacuum, the term, "B.t.u. per kilowatt-hour" has been used. If we use the term myrawatt,

$$\text{Per cent. overall efficiency} = 10 \times \frac{\text{kilowatts output}}{\text{myrawatts input}}$$

Also, with the thermal efficiency of the engine known, the heating surface in the boiler room is determined (assuming the 10 sq.ft. rule) thus; 2 kw. per myrawatt input to an engine is equivalent to 20 per cent. thermal efficiency of the engine and the heating surface in the boiler room equals kilowatts engine output × 10/2.

Obtaining an exact figure of the same with the boiler-horsepower unit involves a tedious operation.

The efficiency of internal-combustion-engine-driven units of all cycles, such as Diesel, Brayton, Otto, etc., is determined by rating the heating value of the fuel i myrawatts; thus

$$\text{Per cent. efficiency} = 10 \times \frac{\text{kilowatts outpu}}{\text{myrawatts inpu}}$$

With hydraulic machinery, again, ratin the water-power input to the wheels i myrawatts, the formula is the same.

Thus, in the term myrawatt lies a sim ple, logical and universal means of com paring outputs and inputs of all classes c energy converters, the meaning of whic will be clear to all engineers wherever piece of electrical machinery is used.

Jersey City High School Power Plan

By Warren O. Rogers

Several types of main generating units have been installed, water-tube boilers and all necessary auxiliaries for cleaning, ventilating and heating the school building.

For completeness of equipment, the newly installed steam-power plant of the Technical and Industrial High School, Jersey City, N. J., is said to stand second to none installed in similar institutions in the United States.

The new power house is in the hollow square formed by the school building. It is of brick and well lighted and ventilated.

Although numerous types of machines have been installed, they are not to be used for experimental purposes, but the students will be allowed to visit the engine room and obtain practical information regarding steam-plant operation by observation and explanation.

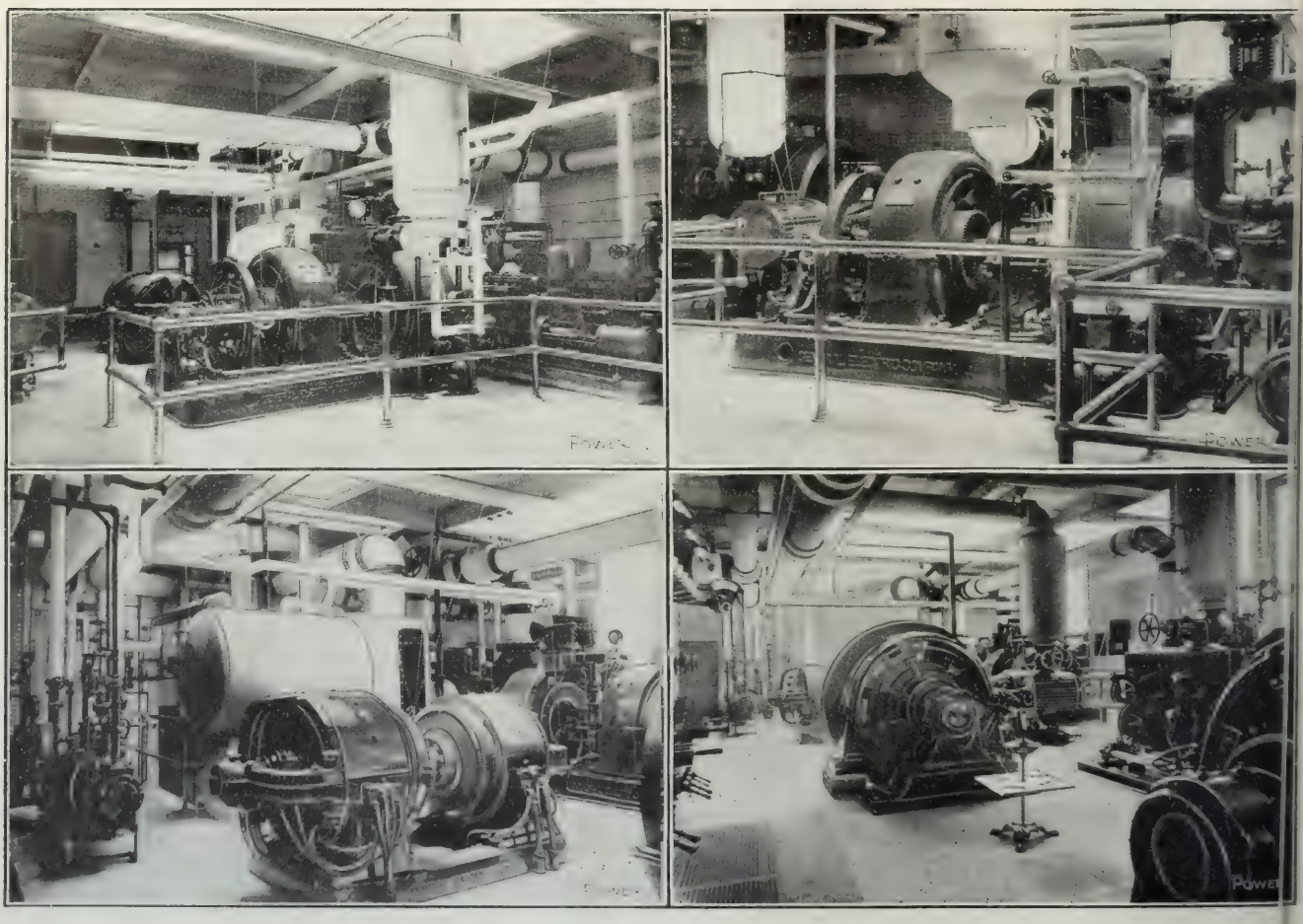


FIG. 1. FOUR VIEWS OF THE ENGINE ROOM

The restricted floor area compelled crowding the machinery somewhat, but even with this handicap the units have been so installed that each can be di assembled for repairing, etc., witho undue trouble.

ENGINE ROOM

In the engine room are two stea engines, one steam turbine, a moto driven generating set, and pumps f boiler feeding, hot-water supply and t heating system. Of the two engines t smaller is a side-crank, medium-spe unit, running 200 r.p.m., and direc coupled to a direct-current generat This unit at present supplies all t power for lighting, fans, motors, c., but when the industrial departments ha been completed and are in operation, he

large four-valve engine directly connected to a direct-current generator running at 200 r.p.m., will have to be used.

At about the center of the room is a 200-kw. horizontal steam turbine, running noncondensing, connected to a direct-current generator of 800 amp. capacity. The turbine is protected by an automatic safety stop valve on the steam line. The fourth generating unit is a motor-driven set taking current from outside commercial lines at 500 volts. The motor is of 150 hp. capacity, running at 900 r.p.m., and is coupled to a direct-current generator of 400 amp. capacity. This unit is a reserve for use in emergency or to take overload from the other units. The four views shown in Fig. 1 give a good idea of the general arrangement of the engine room, also the pump used for the various purposes. The steam-driven units all have separators, and oiling systems. The load on the three-wire system is balanced by either of two balancing sets, each of 40 kw. capacity.

Boiler Room

Fig. 2 is a view in the boiler room, which contains four 265-hp. water-tube boilers, carrying 135 lb. steam pressure. Each set of two boilers is served by a stack 100 ft. high. The furnaces have shaking and dumping grates. The feed pipes and drips from the water columns are brass.

Buckwheat coal is burned and is hand-fired. It is delivered to a large storage bin under the yard pavement and directly in front of the boilers. The coal wagons are unloaded through a hatchway in the pavement into this room, from which fuel is conveyed to the boiler room in a wheelbarrow. The ashes are removed from the ashpit into barrels hoisted to the ground level by an electric hoist. The blowoff pipes are galvanized iron and discharge into an iron tank, from which the accumulated water is removed by an air ejector to the sewer. The air is furnished by a small air pump, on the wall of the engine room. The draft is controlled by a damper regulator. A forced-draft system is installed but has not been used so far. The draft fan is driven by a 27-hp. turbine running at 1600 r.p.m.

The feed water of the boilers is heated in a closed heater, the supply to which is automatically controlled by a regulator on the feed pump.

Fig. 2. Partial View of the Boiler Room

EQUIPMENT OF THE JERSEY CITY HIGH SCHOOL POWER PLANT

	Equipment	Make	Use	Kw.	Volts	Amp.	R.p.m.	Size	Pressure lb.	H p	
1	Engine	Fleming Harrisburg	Main unit				200		130		Harrisburg Foundry & Machine Works.
1	Engine	McIntosh-Seymour	Main unit				200		130		McIntosh-Seymour & Co.
1	Turbine	Curtis	Main unit	200			1800		130		General Electric Co.
1	Turbine	Sturtevant	Forced draft fan				1600		15	27	B. F. Sturtevant Co.
1	D. C. Generator	General Electric	Main unit	200	250	800	250				General Electric Co.
1	D. C. Generator	General Electric	Main unit	200	250	800	200				General Electric Co.
1	D. C. Generator	General Electric	Main unit	200	250	800	1800				General Electric Co.
4	Water tube boilers	Heine	Main units						135	265	Heine Safety Boiler Co.
1	Fan	Sturtevant	Forced draft				1600				B. F. Sturtevant Co.
1	Motor	General Electric	Main unit	250	500	250	900			150	General Electric Co.
1	Generator	General Electric	Main unit		250	400	900				General Electric Co.
1	Electric hoist	Gurney	Ash								Gurney Electric Co.
1	Heater	Reilly	Feed water					31½			Griscom-Spencer Co.
2	Balancers	General Electric	Distributing lines	14	125	112	1425				General Electric Co.
2	Stacks	Custodis	Smoke					100 ft.			Alphonse Custodis Chimney Construction Co.
4	Water columns	Reliance	On boilers								Reliance Gauge Column Co.
1	Air pump	Westinghouse	Sump ejector								Westinghouse Air Brake Co.
	Valves	Crane	Main pipes								Crane Co.
	Piping	Wrought iron	Steam and water								Evans, Almiral & Co.
1	Scale	Fairbanks	Coal								Fairbanks Co.
1	Safety stop	Automatic	Turbine								Schulte & Koerting.
8	Lubricators	Detroit	Engines and pumps								Detroit Lubricator Co.
1	Lubricator	Richardson-Phenix	Engine								Richardson-Phenix Co.
2	Air compressors	American	Heating system					8x8x8	130		American Air Compressor Works.
2	Pumps	Worthington	House service					7½x6x10	130		Henry R. Worthington.
2	Pumps	Simplex	Heating system						130		Union Steam Pump Co.
2	Pumps	Outside packed	Boiler feeder					10x6x10	130		Henry R. Worthington.
1	Filter	Blackburn & Smith	Water supply								James Bogg & Co.
4	Regulators	Standard	Pumps								H. Mueller.
1	Vacuum cleaner	Richmond	House								McCrum-Howell Co.
1	Switchboard	General Electric									General Electric Co.
12	Traps	Anderson	Drips and returns								The V. D. Anderson Co.
2	Washers	Standard	Air washing					No. 4			Standard Regulator Co.
2	Centrifugal pumps		Air washing water					2½			Platt Iron Works.
2	Motors	Ft. Wayne	Circulating pumps		250	2-8					Ft. Wayne Electric Co.
4	Fans	Sturtevant	Air circulating				1800				B. F. Sturtevant Co.
7	Fans	Sturtevant	Air circulating					7 ft.			B. F. Sturtevant Co.
4	Motors	Direct current	Air fans		250	77		7 ft.		20	General Electric Co.
7	Motors	Direct current	Air fans		250		185			15	General Electric Co.

Pumps

Two outside-packed pot-valve pumps deliver water, after filtering, to the boilers. The exhaust from the engines and pumps passes through a grease extractor. There are also two duplex service pumps and two simplex pumps used on the vacuum system when it is operated. The heating system has thermostatic control, the air supply to which is furnished by two simplex air compressors. All drips from the building are piped to traps discharging into a return tank, and are pumped through a filter to the heater and on to the boilers. The vacuum pumps discharge water from the heating system into a tank near the ceiling, from which it flows by gravity into a second tank, to be pumped to the boilers as feed water.

The school is ventilated by four 7-ft. fans, driven by 20-hp. motors, and seven 7-ft. exhaust fans in the attic, driven by 15-hp. motors.

The current generated by the main units is controlled by switches on a blue-marble switchboard, at one end of the engine room, illustrated in Fig. 3.

The piping in the engine room is suspended from the ceiling by rods and cross hangers, and in some instances tee irons are inserted in the concrete ceilings and beams, carrying I-beams at right angles from which the hangers are suspended.

There are 170 motors in all throughout the various departments averaging 3 hp. each, and 4200 incadescent lamps and 30 arc lights.

Although there are no radical departures in power-plant installation, the completeness of the plant shows that those in authority are alive to the need of providing suitable equipment from which the students taking the industrial course can learn something of uptodate practical operation.

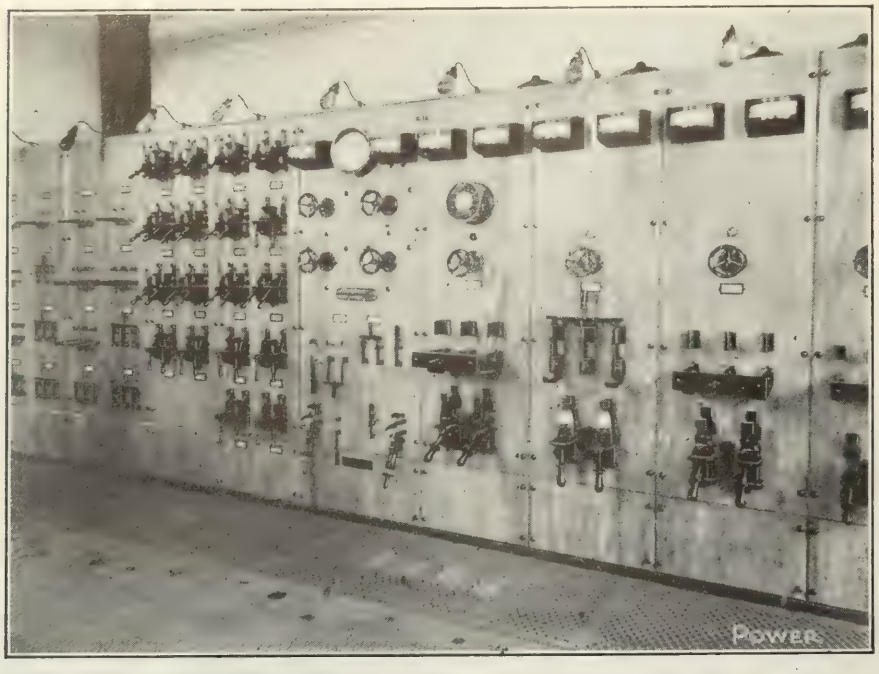


Fig. 3. Switchboard

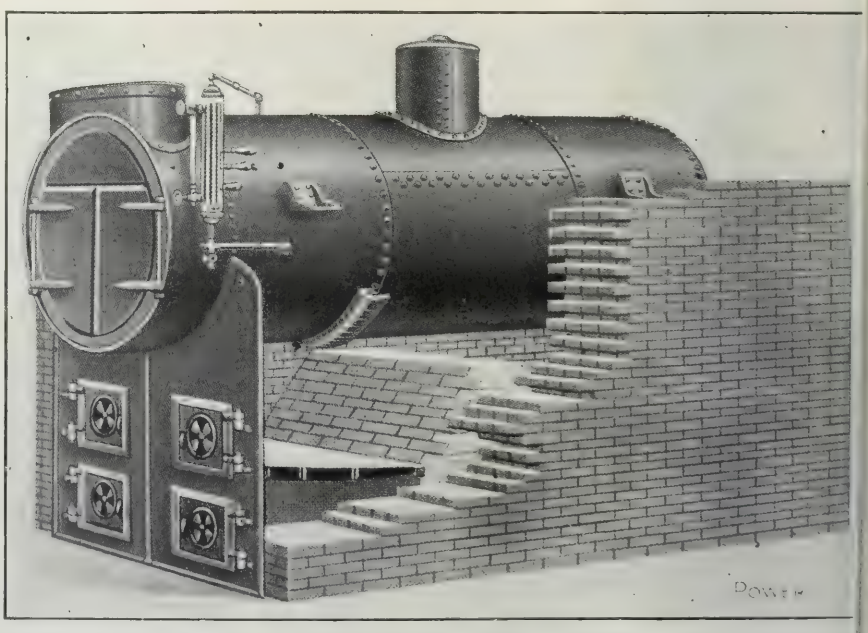
Girth Seam Protector

Horizontal return-tubular boilers are more or less susceptible to trouble from cracks at the girth seam over the fire. One authority estimates that as high as 50 per cent. of the boilers in use develop girth-seam cracks sooner or later. In many cases, such cracks are harmless and if the boiler can be kept tight little attention need be paid to them. However, there is always danger that a crack will extend beyond the rivets and become serious.

The cracks are due to the concentration of heat at the seam where the courses overlap making a double thickness of metal. The distance from the outside to the inside at this point is too great for the water to keep the edge of the outer course cool and hence when the fire-door is opened the inrush of cold air causes extreme contraction and expansion strains and the frequent cracks.

To protect the girth seam by shielding it from the direct impingement of the hot gases and flames or the drafts of cold air, the J. J. Gage Boiler Girth Seam Protector Co., 802 Conover Building, Dayton, Ohio, has placed on the market the device shown attached to a boiler in the accompanying illustration. A piece of ⅝x2-in. bar steel is rolled to conform with the circumference of the shell and is fastened to it by four stud bolts. On this bar are hung a number of special tile of hard refractory clay which fit tightly against the boiler shell as shown. These tile fully cover the seam and also prevent any part of the steel bar or the bolts from coming in contact with the flame.

The protector may easily be attached to any boiler without disturbing the setting.



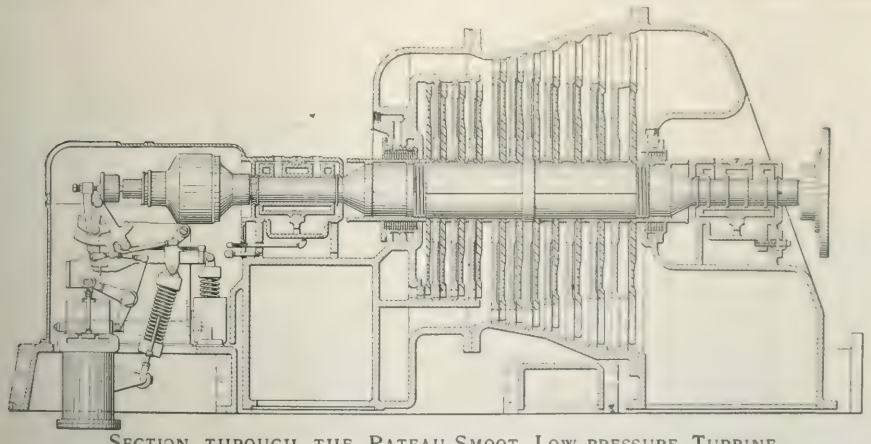
The J. J. Gage Boiler Girth Seam Protector

The latest thing in engineering-society journalism is a periodical published in two languages. The last two issues of the two-monthly *Journal* of the Society of Hungarian Engineers and Architects of America have each contained two leading articles, one printed in Hungarian and one in English. The English articles in each case deals with engineering developments in Hungary, while the articles in the Hungarian language describe engineering work in America. This plan might well be adopted by like societies.

Rateau-Smoot Low Pressure Turbine

The Illinois Steel Co. recently installed a low-pressure turbine at its South Works, South Chicago, to utilize the exhaust steam of several reversing engines. The exhaust steam is directed into Rateau steam regenerators so designed that the intermittent flux of exhaust steam is transformed into a steady flow, taking care of full load on the steam turbine not only during the mill cycle, but when the mill cycle is interrupted for periods not exceeding two minutes. The steam turbines are mixed-flow machines of the Rateau-Smoot type.

A cross-section of the turbine is shown herewith. It has an entirely independent high-pressure section. High-pressure steam passes through the high-pressure wheels and, after expanding, enters the low-pressure section. When working on live steam the turbine is a complete high-pressure machine. The low-pressure steam is admitted direct to the low-pressure wheels from around the high-pressure section of the turbine; thus the highest possible economy is attained when working on either high- or low-pressure steam.



SECTION THROUGH THE RATEAU-SMOOT LOW-PRESSURE TURBINE

The governor is so designed that the pressure of the low-pressure steam controls the flow of high-pressure steam, and no live steam is admitted to the turbine before the low-pressure steam has become deficient, or when it has reached atmospheric pressure, in this instance. This occurs when the steam regenerators have been delivering steam during a shutdown of the mill engines for more than two minutes, and during this period the turbine has been carrying full load.

The turbine has a capacity of 3000 kw. when using low-pressure steam exclusively, and when using high-pressure steam can carry a continuous load of 4000 kw.

Only one turbine is in operation, the second turbine being under construction, and will be ready to operate in about 30 days. The bearings are ring-oiled with water-cooled bushings. This is said to be the largest turbine in the world having this form of lubrication and bearing cooling.

The stuffing boxes are built up of a number of consecutive rings of carbon blocks separated from each other by diaphragms and having springs to hold the carbons in contact with the shaft. No adjustment is necessary during the life of the box.

The governor is direct acting. The flyballs and spring against which they act are mounted within a steel shell fixed on the high-pressure end of the turbine shaft. The motion of the flyballs is transmitted by links to the high- and low-pressure throttle valves. The governor is powerful and controls the throttling valves with a high degree of regulation.

When first put into service the turbine regulation was adjusted for a 3 per cent. variation in speed when changing from no load to full load. When placed in parallel with other machines it was observed that this turbine took all of the load when it increased and dropped most of its load when the load decreased, because the other machines with which it was running in parallel were not so closely regulated, and it was necessary to readjust the governor to give a 6 per cent. difference between no load and full-load speed before it would carry its proper share of the load on the entire system.

The governing mechanism is so arranged that the action of the flyballs is to hold the valves open; therefore, the breakage or disconnection of any of the links between the flyballs and valves results in the valve closing automatically by its own weight, which, in the case of the high-pressure valve, is assisted by a closing spring.

The governor is equipped with a releasing trigger, which can be operated by hand or by the overspeed device and acts to disconnect the valves from the flyballs and the governor, permitting the valves to close whenever it is desired to shut down the turbine.

A single hand-operated lever restores both high- and low-pressure valves into proper connection with the governor and also replaces the tripping device. The governor can be tripped and restored to action within two or three seconds, and when the turbine is running loaded the safety trip can be actuated by hand and the valves put back in commission without appreciably disturbing the load carried by the machine, due to the heavy flywheel effect of the turbine and alternator, and also to the extreme rapidity with which a man can operate the governor.

The mixed-flow part of the governor comprises a piston actuated by the low-pressure steam, which rises and falls with the pressure in the low-pressure steam main. When the low-pressure steam has a higher pressure (in this case some 20 lb.), the piston takes its proper position and causes the flyball governor to actuate the low-pressure valve for its full travel while holding the high-pressure valve in a closed position. As the quantity of low-pressure steam decreases, the piston which it actuates descends, giving a greatly increased opening to the high-pressure valve and decreasing the opening to the low-pressure valve until it reaches its lower limit, under which conditions the low-pressure valve is always in a closed position and the high-pressure valve is actuated by the flyballs for its maximum displacement.

An intermittent position of the control piston gives a simultaneous opening to both high- and low-pressure valves, thus causing the turbine to run on both sources of steam supply.

By this arrangement the turbine tends to run exclusively on low-pressure steam, when its pressure is above that of the atmosphere, but when the pressure falls to atmospheric the turbine takes less exhaust steam and more high-pressure steam, the governor being a regulator of the low-pressure steam supply.

A feature of this governor is that the load carried by the turbine does not change appreciably when the turbine goes from high- to low-pressure steam, because the change in the source of steam for driving the turbine is accomplished without any change in speed and, therefore, without any shifting in the position of the flyball governor. When the turbine is carrying half load its flyball governor assumes an intermediate position, which is exactly the same whether the turbine is driven with high- or low-pressure steam, or a mixture of both.

The governor effects this change by means of the low-pressure steam and not by the speed of the flyball governor, which, however, is always in direct control of both high- and low-pressure valves and has absolute speed control of the turbine regardless of which kind of steam it is operating upon and also regardless of the performance of the selective device which determines whether the turbine shall run on high- or low-pressure steam.

Shaft Governors

By Hubert E. Collins and Edwin F. Williams

Several shaft governors are described, and the manner in which the governor controls the valve movement is gone into in detail.

Several tables are given which show the various positions of the valve under different conditions.

The shaft governor is especially adapted for engines of comparatively high rotative speed, and the closeness of regulation in such engines is often remarkable. It is possible for the governor to hold the speed practically constant through all ranges of load and even with sudden variations of load.

A number of American engines are designed for moderately high rotative speeds, using Corliss, piston or flat valves in which the admission valves are driven direct from a shaft governor; the exhaust valves are independent and separately driven from a fixed eccentric. On such engines the steam valve both opens and closes the admission, the governor shifting the steam eccentric so as to change its angular advance and throw, keeping the lead nearly constant, thus varying the cutoff. The valve-gears of nearly all these engines are so constructed as to modify the plain eccentric movement of the eccentric rod by certain angular movements of the rocker-arm which accelerate the valve movement during the admission period.

This is done to give wider steam-port opening at any given point of cutoff, avoiding the necessity of abnormally long ports and large clearances where a limited travel of valve is constructively essential.

High-speed American engines have one valve for both steam and exhaust. The governor is on the crankshaft and controls the position of the eccentric, shifting it to alter its angular advance and throw, keeping the lead nearly constant. The range of cutoff is from zero to approximately 72 per cent. of the stroke. Incidentally the opening and closing of the exhaust varies with the varying cutoffs, only in a lesser degree. The same governor is applicable to all valve systems where the admission and cutoff are effected by one valve, whether the exhaust is dependent, as in the single-valve type, or independent as in the four-valve type.

Various forms of shaft governors are used with the single-valve control accomplishing practically the same eccentric movement only in different ways. Fig. 1 shows sectional and side elevations of a governor mounted on the end of a shaft, and driving the valve from an eccentric pin.

With the governor weight O in the position shown, the latest cutoff will occur, and at the position P no steam will be admitted. Between these positions intermediate cutoffs will be obtained. The arm to which the eccentric pin E is attached is pivoted on one of the governor casting arms at A, and at the other end is connected with the opposite end of the weight arm at D by the link CD. As the weight moves from its inner position O to its outer position P, the end of the eccentric arm moves from C to X, carrying the eccentric pin E from E to F. This is the governing movement.

The governor springs hold the weight at O until the engine is up to normal speed when the centrifugal force of the weight will equal the pull of the springs. An increase of speed will cause the weights to move outward until the two forces just balance, and if sufficient will set the weight against the outer stop S, closing the valve and shutting steam off entirely.

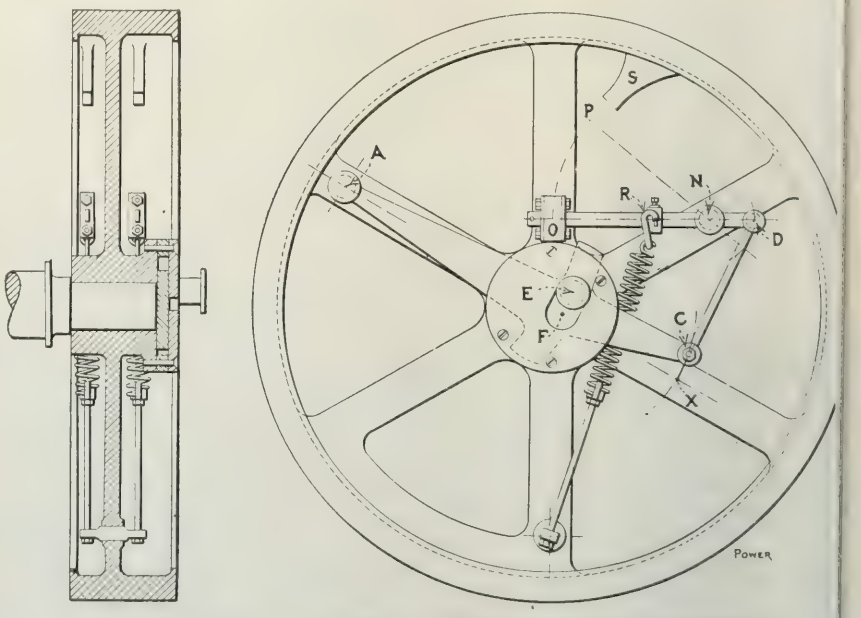


FIG. 1. SECTIONAL AND SIDE ELEVATION OF SHAFT GOVERNOR

In all direct-driven governors, the valve travel being equal to the governor eccentric-rod leg's lateral movement, the distance from A to E is equal to the distance from A to the center of the shaft, plus the steam lap of the valve, minus about $\frac{1}{32}$ in. to cover the port entirely when the eccentric is in its position of least throw. The valve travel with the weight at the inner position is commonly four times the steam lap and its travel with the weight at the outer position is invariably equal to, or slightly le than twice the lap.

In Fig. 2, the outer circle represe the eccentric center when given its gre est throw (the weight being at its in position), and the inner circle the pa of the eccentric center when given su cient throw to give the valve a tra equal to twice the steam lap. If t were the shortest throw given at governor, no steam would be admitt The eccentric center would be located D when the crankpin is at K and governor weight would be at the ou stop. With the engine standing on de center and the governor weight mov backward and forward between the in and outer limiting stops, the eccent would pass back and forth along the l from C to D. This line is called even lead line. In practice it is not f lowed for the reason that it does not mit of lead. The controlling or gove ing movement is commonly made al the line EJ either straight or sligh curved. The shaded portions of the d gram representing no admission, it

be seen that there is positive lead at E and negative lead at J, the controlling movement crossing the even lead line at F. When the weights are at their outer stops the eccentric center is at F, a point within the no-admission zone, where no steam can be admitted at any position of the piston.

In Fig. 3, the outer circle represents the path of the crankpin, the central circle the path of the eccentric center at greatest throw, and the inner circle the path of the eccentric at least throw. The line AP is the even lead line.

The piston valve is shown in the position it would have were the eccentric anywhere along the even lead line. Whether the governor weights were in or out, the valve would be at the point of opening when the piston is at the beginning of the stroke—the crank being on the dead center.

Evidently, were the eccentric center permanently located at *A* and the crank rotated in the direction of the arrow, the valve would immediately open and stay

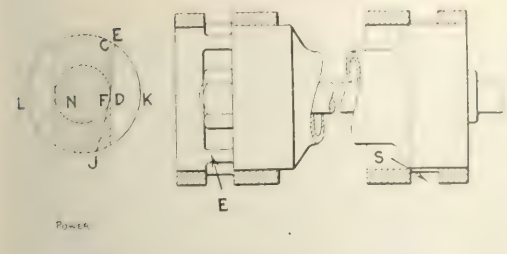


FIG. 2. REPRESENTING ECCENTRIC CENTER AT GREATEST THROW

FIG. 3. REPRESENTING PATHS OF CRANK PIN AND GREATEST AND LEAST ECCENTRIC THROW

so until the point *RA* was reached. During this period of admission, the crankpin would have traveled from *E* to *S*, an angular distance equal to that from *A* to *R*. Disregarding for the present the angularity of the connecting-rod, the point of cutoff corresponding to the crank position *S* is 75 per cent. of the piston stroke.

With the governor weight at the inner stop therefore, the valve admits steam 75 per cent. of the stroke. With the weight at the outer stop, the eccentric center being at *P*, there will be no admission period. With the weights at an intermediate position, the eccentric center being at *B*, the admission period would be from *B* to *D*, the corresponding crank angle being *ECF* and the cutoff being at 30 per cent. of the piston stroke.

The lap being fixed and the travel of the valve variable, the lineal port opening and the admission period may be expressed in terms of the lap. For example, with the eccentric center at *A*, the lineal opening *PG* equals the lap *CP* and the travel of the valve is four times the lap *CP*. With the eccentric at *B* the lineal opening is 0.187 of the lap, and the valve travel is 2.374 times the lap.

In Table I is given the valve travel and the lineal port opening referred to the steam lap uncorrected for the connecting-rod angularity.

TABLE 1

Per cent. of admission	Travel of valve referred to lap	Lineal opening referred to lap
10	2.08	0.0416
20	2.208	0.104
30	2.374	0.187
40	2.54	0.29
50	2.79	0.395
60	3.123	0.565
70	3.625	0.81
75	4.00	1.00

When a single valve for both steam and exhaust is under governor control, the exhaust opening and closing points vary with the cutoff, being early when the cutoff is early, and late when the cutoff is late.

In Fig. 4, the outer circle represents the path of the crankpin, the next circle the path of the eccentric center at full throw, the next its path when the governor has reduced the throw sufficiently to occasion a 30 per cent. steam admission, and the inner circle no steam admission. The shaded part of the diagram indicates no admission, its breadth just equaling the steam lap. When the eccentric has full throw, its center will be at the opening point *F*, when the crankpin is at *A* on a dead center, and the admission period will be equal to the arc *AB* terminating at *B*, which is 75 per cent. of the stroke, the angular length of the arc being 120 deg. The exhaust side of the valve having no lap, steam is released at *H*, 30 deg. ahead of the opening point for steam admission, and the crankpin being on center, and piston

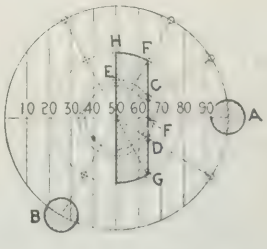


FIG. 4. PATH OF CRANK AND ECCENTRIC

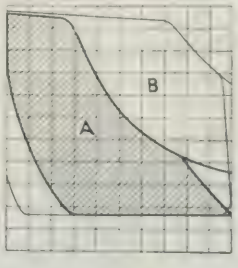


FIG. 5. VALVE ACTION AT TWO POINTS OF CUTOFF

at the beginning of the stroke, the point of exhaust closure and opening takes place at the 150 deg. angle, which corresponds to 86 per cent. of the stroke.

Assume the governor to have shifted the eccentric center to *C*. The angular advance will be increased to 56.5 deg., the steam admission will have been shortened to 30 per cent. of the stroke and the exhaust action will have been transferred to *E*, which is 56.5 deg. ahead of *C* and the exhaust will open at 77.5 per cent. of the stroke and close at the same per cent. of the return stroke.

With the eccentric center shifted to its extreme inner position at *F*, no steam will be admitted and the exhaust opening and closing point will be advanced to the 90-deg. crank angle, which is 50 per cent. of the stroke.

The following table gives approximate points of exhaust opening and closure and crank angles corresponding to varying points of cutoff:

TABLE 2

Period of steam admission, per cent.	Part of stroke, crank pin and valve, per cent.	Part of stroke, crank pin and eccentric, per cent.	Crank angle at point of cutoff, deg.
5	62.5	37.5	27.0
10	67.0	33.0	37.5
15	68.5	31.5	45.5
20	73.0	27.0	[illegible]
25	75.0	25.0	[illegible]
30	77.5	22.5	[illegible]
35	80.0	20.0	[illegible]
40	82.5	17.5	[illegible]
45	83.5	16.5	[illegible]
50	85.0	15.0	90.0
55	86.8	13.2	96.0
60	88.0	12.0	101.5
65	90.0	10.0	[illegible]
70	91.5	8.5	113.5
75	93.0	[illegible]	120.0

Fig. 5 shows the valve action at two different points of cutoff, namely, 30 per cent. and 75 per cent. The shaded area gives the distribution of pressures for 30 per cent. admission. The exhaust opens at 77 or 78 per cent. of the stroke and closes at the corresponding position on the return stroke, giving a compression period of approximately 22.5 per cent. of the stroke. From the unshaded diagram *B*, showing a three-quarter cutoff, it will be seen that the exhaust opens and closes much later, so that very little compression occurs. The governor, therefore, controls by compression as well as by admission; adding to or taking from the area of the indicator diagram in both ways simultaneously.

Lead can be obtained in this valve system only by moving the governor around the shaft in the direction of rotation until the desired lead is obtained with the governor weights in, eccentric at full throw and the crankpin on the dead cen-

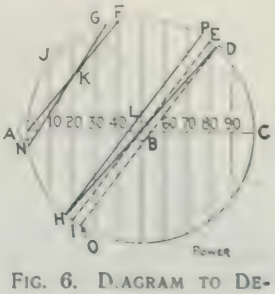


FIG. 6. DIAGRAM TO DETERMINE THE EXHAUST CLOSING AND OPENING PERIOD

ter. When secured to the shaft in a position to give lead at the full throw of the valve, the governor will diminish that lead until it becomes zero on the shorter cutoffs and negative at the shortest travel of the valve. Hence the valve should be given excessive lead at full throw, say 5 per cent. of the valve travel.

The effect of lead is to advance the position of the eccentric relative to the crank for all cutoffs. This hastens the exhaust and limits the steam admission.

Lead affords a greater lineal valve opening for a given cutoff, the approximate increase being equal to one-half the lead.

Assume a valve having 5-in. travel to have sufficient lap to cut off at 30 per cent. of the stroke without lead. The lap and point of opening may be found from Table 1. If the valve is given 5 per cent. lead (¼ in.) the lap may be reduced ⅛ in. and that amount will be added to the lineal opening of the port at full travel.

Generally there is no lap, either positive or negative, on the exhaust side of the valve. When the per cent. of cylinder clearance is large, however, positive exhaust lap is beneficial as it gives more compression and, incidentally, a later release. With small cylinder clearances, a little negative exhaust lap is advisable. In all cylinders having, say, 8 per cent. clearance, make the exhaust lap zero. When the clearance is from 10 to 14 per cent., make the exhaust positive equal 5 per cent. of the full valve travel. If the clearance does not exceed 5 per cent., the exhaust lap should be negative, approximately 2½ per cent.

A simple method of determining the points of exhaust opening, closing, etc., originated by Prof. John E. Sweet, is shown in Fig. 6.

Draw the outer circle equal in diameter to the travel of the valve. Then draw a circle whose radius *KL* equals the steam lap. Then, if there is no steam lead, draw from *A* the straight line *AF* tangent to the steam-lap circle, intersecting the travel circle at *F*. This is the point of cutoff without lead. By dividing the diameter *AC* into ten equal spaces and erecting perpendiculars at each division terminating in the travel circle it is easy to determine the per cent. of the stroke corresponding to any arc of crank movement beginning at zero.

It will be seen that the cutoff at *K* is at 40 per cent. of the stroke. Draw the straight line *HD* parallel to *AF*, passing through the center at *L* and terminating in the circumference of the outer circle at *D* and *H*. The former is the point of exhaust opening, approximately 82 per cent. of the stroke, and the latter the point of exhaust closure, approximately 18 per cent. of the stroke, when there is no lap on the exhaust side of the valve.

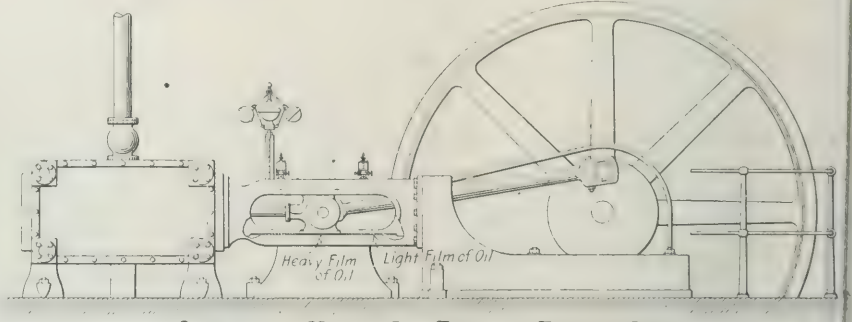
If lead is given the valve, draw the small circle at *A* with its radius equal to the lap, and draw the straight line *NG* tangent to this and the lap circle terminating in the travel circle at *G*. This is the point of cutoff with lead, approximately 35 per cent. of the stroke. Draw the straight line *IE* parallel to *NG*, passing through the center *L* and terminating at *E* and *I*. The point *E* is the exhaust opening and the point *I* the exhaust closure with steam lap, but with no exhaust lap.

If there is exhaust lap, draw the small circle at *L* with a radius equal to the exhaust lap and the straight line *OD* tangent to the circle at *L* and parallel to the line *NG* when there is steam lead, or parallel to the line *AF* if there is no lead. The exhaust will open at *D* and close at *O*. If the exhaust lap is negative draw the straight line *HP* tangent to the exhaust-lap circle *L* on the other side and parallel to the line *AF* if there is no steam lead, or parallel to the line *NG* if there is steam lead. The terminals *H* and *P* will represent the exhaust closure and opening, respectively. It will be seen that positive exhaust lap hastens the closing and retards the opening, while negative exhaust lap does just the reverse.

This method is strictly accurate throughout, covers the whole ground, and is the most simple and direct of any known to the writers.

Crosshead Pressure on Guide

Engineers have for years discussed the advantages and disadvantages of running an engine over and under. An engine will operate more quietly when running over than when running under, because it is not necessary to adjust the crosshead shoe so closely and there is no drop to the crosshead from the upper to the lower guide.



LIGHT AND HEAVY OIL FILM ON ENGINE GUIDE

When an engine is running under the pressure is between the bottom crosshead shoe and the lower guide at both ends of the stroke. But the center of the upper guide sustains the crosshead thrust at midstroke.

In the accompanying sketch the lightly shaded portion on the guides at the ends represents the thin oil film as it will show on the guide when an engine is running under. The dark shading represents a heavy deposit of oil, showing that the crosshead shoe is forced up against the upper guide during the central portion of the stroke and that it drops to the lower guide at each end of the stroke.

Moisture in Exhaust Steam

For many reasons it is highly desirable to ascertain the quality of the exhaust steam of engines and turbines, but, owing to the fact that the moisture is not evenly distributed throughout the cross-section of the exhaust discharge, this is usually a very difficult problem.

One company reports its experience in investigating this subject to the Prime Movers Committee of the National Electric Light Association as follows:

"To make a practical determination of the percentage of moisture present in the exhaust of the 46 × 94 low-pressure cylinder of a vertical cross-compound engine, operating on 25 in. of vacuum, with steam at 175 lb. pressure and with 4 deg. of superheat, openings were made into the exhaust chamber and a nozzle inserted, which was piped through a Carpenter separating calorimeter and a small condenser and connected to the dry vacuum pump which served the engine.

"The relative amounts of water collected in a given time by the calorimeter and by the condenser were, of course, an indication of the percentage of moisture carried by the exhaust steam. The same apparatus was later connected to the exhaust chamber of a steam turbine operating on 27½ in. vacuum. The averages obtained in each of these two tests indicated that the exhaust of the reciprocating engine carried 14 per cent. of moisture and the exhaust of the turbine carried 16 per cent. of moisture, although determinations made at different parts of the exhaust pipe of the engine gave results varying from 6 to 28 per cent., and the determinations made in different parts of the turbine base gave results varying from 6 to 18 per cent."

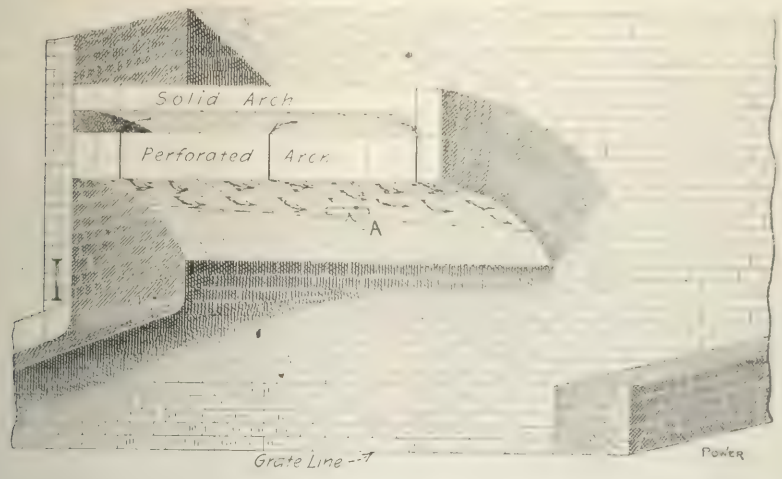
Probably the most carefully conducted investigation of this subject is that reported by Messrs. Stott and Pigott in a paper read before the American Society of Mechanical Engineers in December, 1910, and published in the *Transactions for* 1911, Vol. XXXII, page 69.

It is reported that the Government Bureau of Standards is about to make an investigation of this general subject.

Ventilated Furnace Arch

The accompanying figure shows the Carr ventilated furnace arch, designed and patented by E. E. Carr, Monadnock Block, Chicago, Ill. The object of the design is to prevent the arch from becoming overheated, thereby prolonging its life and reducing the cost of maintenance.

The structure really consists of two separate arches, an upper solid and a lower perforated arch. As the main or combustion arch is being built, series of openings are formed in the joints between the brick through which the draft



VENTILATED FURNACE ARCH

causes a proper supply of air to circulate. These openings lead into the air chamber between the main arch and the single-ring arch built over it, the space between the two being sealed off at the rear. Air from the outside is admitted at any convenient point in the boiler setting, as at *A*.

In traveling over the main arch and through its openings, the air conducts the excess heat away from it, thus tending to prevent the brick from being so softened by the heat as to cause the arch to become unstable and fall. This makes it possible to build the arch very flat and thereby secure the added advantage due to the more even distribution of the flame and gases across the boiler tubes.

An Economical Trapping System

At the saltpeter works of Battelle & Renwich, Jersey City, N. J., a novel steam trap system has just been started up, which not only takes care of the condensation from all the apparatus throughout the mill and returns it to the boilers, but is so arranged that the waste vapor from the vents of the traps is condensed and returned with the regular condensation.

The mill apparatus consists of 19 special kettles for boiling. Each kettle is equipped with what is known as a pit coil; that is, the drip from the coil instead of passing through the bottom of the kettle is carried up over the edge, so that the water of condensation is raised 6 ft. before it is discharged. The system also includes a dry-room and drying machines. The approximate steam capacity required for the boiling alone is about 600 hp. The boiling point of some of the liquids in the kettles is as high as 270 deg., so that live steam is required.

On account of the fact that the coils are submerged in the kettle and the condensation has to be lifted some 6 ft. over their sides it is necessary to keep up a continuous flow of steam to carry this water along. To effect this the special No. 20 Lytton trap with which each kettle is equipped is bled in the steam space, the escaping steam being collected in a common main and utilized in the dry room. The traps discharge their water into return mains of which two are used running back through the factory to a receiving tank in the boiler room. The temperature of this returned water varies between 260 and 282 deg. The condensation from the dry room is also returned to the boilers so that no steam nor vapor is allowed to escape.

On the boilers are located two 4-in return traps and one 1½-in. return trap, which take care of the return water from the factory at the temperatures mentioned and return it automatically to the boilers. The condensation from the live-steam mains is also trapped and collected in a separate return which discharges into the receiver in the boiler room.

By the installation of this system the saving in coal alone amounts to 2.5 tons per day, while the boiling capacity of quite a number of the kettles is increased from 15 to 25 per cent. It has not only made a saving in coal and increased the capacity of the kettles, but it is feeding the boilers with about 90 per cent. of pure water which prevents the formation of scale and makes an additional saving in the amount of water that would have to be purchased from the city. The system was devised by D. J. Lewis, Jr., sales manager and trap expert of the Lytton Manufacturing Corporation.

Uehling CO_2 Meter Improvements

The Uehling Instrument Co., Passaic, N. J., has recently introduced several improvements in its CO_2 machines and waste meters. One of these is a consequence of experimental work completed about a year ago, and since then, instead of caustic solutions, dry absorbers have been installed on practically every CO_2 machine out in service, and have been furnished with all new equipments.

The dry-absorber cartons are cylindrical, about 3 in. in diameter and 1½ ft. long, encased in wax pressboard, so that they are easily handled. The absorbent, a flakey substance called natron, is packed in layers within the carton, which is placed in the cylindrical chamber. To renew the absorbent it is only necessary to remove a few wing nuts, withdraw the used carton and insert the new one.

The new type *M* recorder is for service in connection with CO_2 meters or the pneumatic pyrometer (operating on a similar principal), or the waste meter which measures both CO_2 and the temperature. It is a light-pressure or vacuum recorder for vacuums, or pressures, or both, from 0 to 60 in. of water column. When this is supplied for recording CO_2, the chart is calibrated from 0 to 20 per cent. CO_2, and when supplied for recording temperature, it is calibrated up to 1000 deg., or for other ranges as desired. This recorder supercedes a somewhat larger gage, but is operated on the same U-tube principle, being filled with a special oil and involving no levers, pins, springs or similar devices.

In connection with the waste meters, the CO_2 and temperature may be recorded on separate type *M* instruments properly calibrated, or a new instrument just developed may be used and the temperature and CO_2 recorded on the same chart. This instrument, the type *DM*, is similar to the type *M*, but carries two pens and gives a chart with a double reading.

An engineering congress will be held in connection with the exposition at San Francisco in 1915 to commemorate the opening of the Panama Canal. At a meeting of representatives of the four national engineering societies, it was agreed to recommend to these organizations and to the Society of Naval Architects and Marine Engineers that they, together with members on the Pacific Coast, underwrite the expenses of the congress.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Synchronous Booster Rotary Converters with Compound Interpoles

Since the ratio of the direct to the alternating-current voltage in a synchronous rotary converter is practically constant under all conditions of load, to change the voltage of the direct current delivered, necessitates changing that of the applied alternating current. This is undesirable as most converters are operated on constant potential alternating-current circuits.

Several methods are in use for obtaining the desired direct-current voltage regulation to compensate for line drop, but all have their limitations. The synchronous-booster type of rotary converter, however, is now being strongly recommended for this purpose as having many advantages over the other methods. This type of machine, as now built consists of an ordinary rotary converter, having the armature of an alternating-current generator (booster) with the same number of poles, mounted on the same shaft next to the collector rings. By controlling the booster field, the voltage can be made to boost or to lower the alternating-current potential to the converter, thus varying the direct-current voltage in the same ratio.

In the smaller machines the field of the booster is supported on brackets from the main frame of the converter, but the larger machines have the booster frame mounted on the bedplate similar to the main field structure.

Interpoles as applied to rotary converters fulfill the same functions as their more familiar application to direct-current generators and motors; that is, they compensate the armature reaction tending to distort the field, and the neutral plane of the latter remains fixed at all loads. Consequently it is unnecessary to shift the brushes with a change in load. Their application to rotary converters is not new, but a combination of compound-wound interpoles with a synchronous booster converter, represents one of the latest products of the Westinghouse Electric & Manufacturing Co.

Fig. 1 shows the usual wiring connections for a synchronous-booster type rotary converter with compound interpole windings. The booster field is permanently connected in series with the shunt windings on the interpoles. The shunt interpole winding is connected to buck the series interpole winding when the booster is raising the alternating-current voltage, and assists the series winding when the booster is lowering the voltage. This may be explained as follows: As already mentioned the series interpole winding moves the neutral back so that the axis of commutation is at right angles to the axis of the field. This condition is satisfactory as long as there is no current in the booster field, but upon exciting this, the neutral position is shifted backward. The shunt interpole winding, being in series with the booster field, and connected so as to oppose the series winding on the interpoles when the booster is raising the voltage, bucks the interpole flux and effectually prevents the neutral brush position from shifting. With normal field

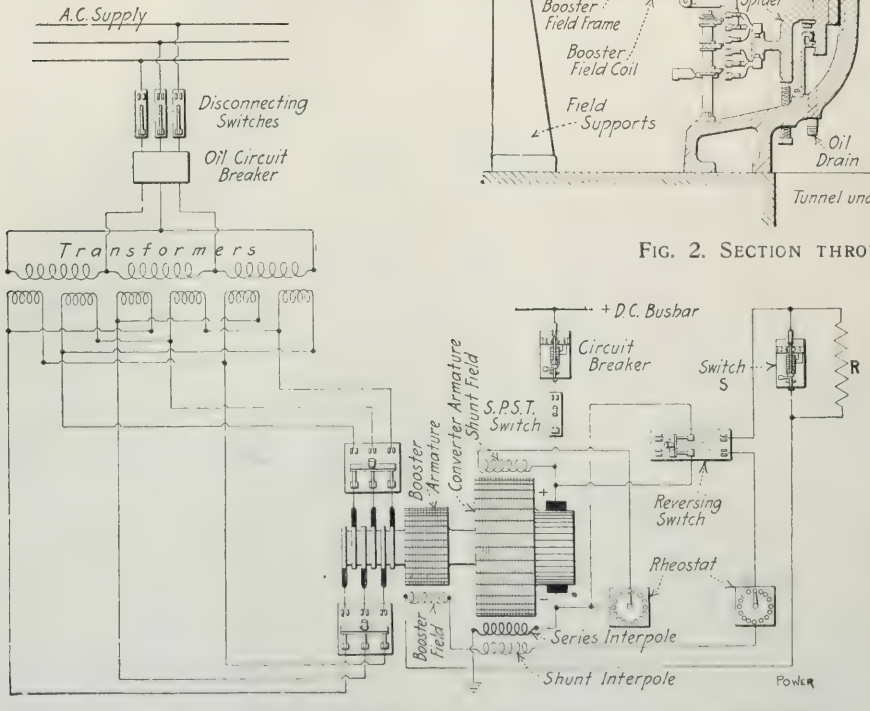


FIG. 1. TYPICAL DIAGRAM OF CONNECTIONS FOR SYNCHRONOUS BOOSTER ROTARY CONVERTER WITH COMPOUND INTERPOLES

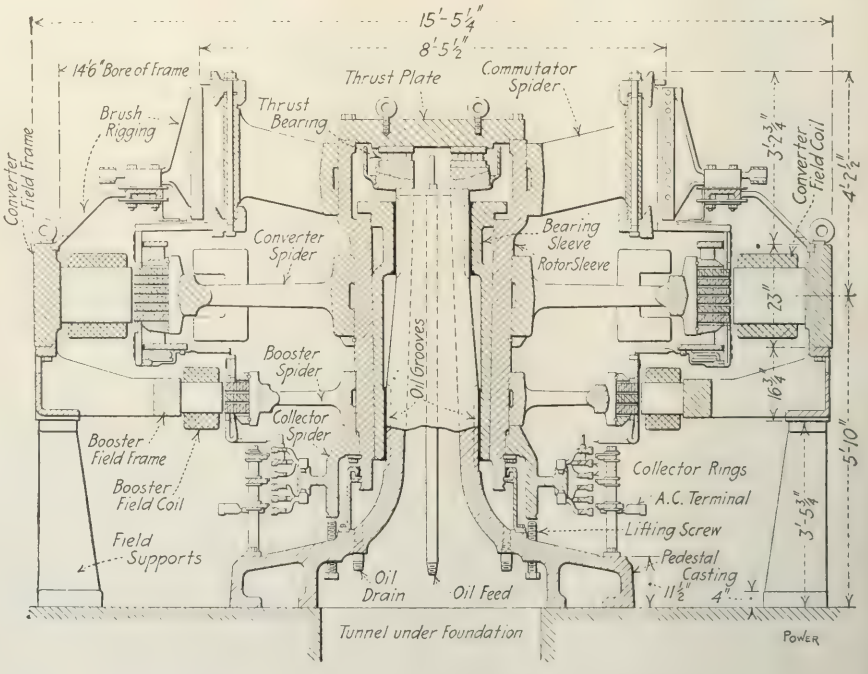


FIG. 2. SECTION THROUGH 3000-KW. CONVERTER

on the booster, it is capable of raising the direct-current voltage about 15 per cent., and with full booster field, the interpole flux is practically zero at full load, because the ampere turns of the series interpole coils are approximately equal to those of the shunt interpole coils.

To lower the direct-current potential, the polarity of the booster field and the shunt interpole coils is reversed. In this case, the booster tends to move the neutral position ahead (since the booster itself is now acting as a motor); consequently, a stronger interpole flux, obtained by both interpole windings acting together, is required to hold the neutral where it belongs.

The switch *S* in Fig. 1 is so attached to the circuit-breaker that it closes when the circuit-breaker is closed, and the resistance *R* is thereby short-circuited. Upon opening the circuit-breaker, the resistance is introduced into the circuit of the booster field and the shunt interpole windings by the opening of the switch *S*. It is necessary to introduce this resistance, which is slightly greater than the combined resistance of the booster field and the shunt interpole windings, in the shunt interpole circuit to prevent the neutral from shifting when the circuit-breaker is tripped under load. The series interpole winding is, of course, cut out of

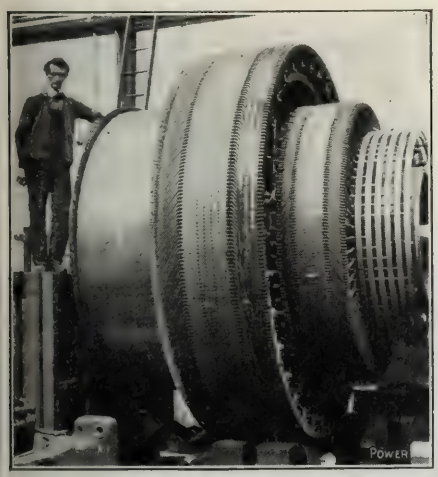


FIG. 3. ARMATURE FOR CONVERTER AND BOOSTER

circuit by the opening of the breaker and the shunt interpole winding in that case, having nothing to buck or help it, depending on whether the booster is raising or lowering the voltage, would move the neutral position either forward or backward, and cause sparking at the brushes if it were not for this resistance.

Rotary converters of this type operate very successfully and commutate at 50 per cent. overload without sparking.

Two 3000-kw., vertical, synchronous-booster type, commutating-pole, rotary converters have just been built and tested by the Westinghouse Electric & Manufacturing Co, for the New York Edison Co., to be installed respectively in its Clinton and its Crosby St. substations. These machines are of unusual interest because they are the largest vertical rotary converters ever constructed and because they are the only vertical converters built involving the synchronous-booster and commutating-pole features.

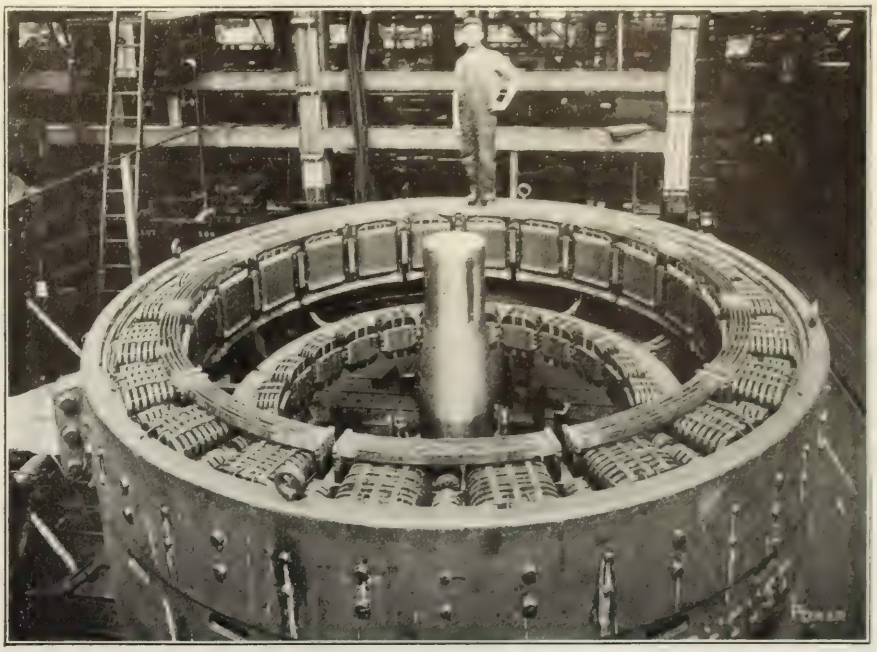


FIG. 4. WINDINGS OF 3000-KW. CONVERTER FOR N. Y. EDISON CO.

The converters deliver a normal direct-current voltage of 270, and are for 25-cycle, six-phase operation. The direct-current voltage variation obtained through the booster feature is 15 per cent. opposing and 15 per cent. boosting from the normal voltage, giving a total direct-current voltage variation of 30 per cent. These machines were made of the vertical type to conform in general construction to the other converters in the substations and start from the direct-current side.

Fig. 2 shows the detail construction of the machines. The converter armature and frame are mounted above the booster. The commutator is at the top of the machine and the collector rings at the bottom. The armature shown in Fig. 3 consists of a rotary-converter armature of the usual construction mounted on the same shaft with the smaller booster armature, which is connected in series with the converter armature so that when the booster field excitation is varied, the alternating-current voltage impressed on the converter armature will be decreased or increased accordingly.

The main or converter poles (Fig. 4) are shunt wound, and the copper grids imbedded in the main pole faces do not extend from pole to pole as in the ordinary machine but are cut off flush with the sides of each main pole; this construction is used on all commutating pole converters, which forms the most effective starting and damping winding. The booster poles are shunt wound, and the booster field is arranged for hand regulation.

A roller thrust bearing is arranged at the top of the pedestal to take the weight of the revolving element. The bearing rests on a plate which has a spherical seat carried on the pedestal so that perfect alignment is assured. The roller bearing can be taken out by removing the top plate of the machine. To assume the weight of the rotating part when it is necessary to remove the top plate, six 1¾-in. bolts are provided that turn through the lower portion of the pedestal flange. When necessary to take off the top plate, these bolts are screwed up until they raise the rotating element slightly and assume its weight.

Lubrication is effected with a gravity oiling system, the oil being forced up through the center of the pedestal and discharging at the top through a nozzle into a cylindrical chamber within the roller bearing.

Heating City by Electricity

Officials of the electric plant in Stavanger, Norway, have asked the city council's permission to electrically heat business houses and dwellings. Stavanger has a population of about 30,000, and 25,000 hp. can be developed by its water-power facilities.

According to present plans, no consumer will be allowed a consumption exceeding 100 hp. The price per horse-power per year will be only about $7 as the cost of production will be very low. The council also has under consideration the heating of Stavanger's public buildings by electricity.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Some Recent European Gas Engine Plants

By Austin C. Lescarboura

In Europe the advantage of using the waste gases from blast furnaces and coke ovens is becoming more fully recognized, especially with the appearance of engines which have been especially designed to operate on the quality of gas obtained from such sources. Gas engines operating on blast-furnace gas are found to give as high as 28 per cent. efficiency, while the best steam engine or turbine would not produce over 15 per cent. Tests have shown that with the same quantity of gas, the gas engine will furnish double the power that would be produced by a steam plant where the gas is burned under boilers to generate steam. Smelting plants are accordingly adopting large gas engines largely for generating electricity or driving blowers.

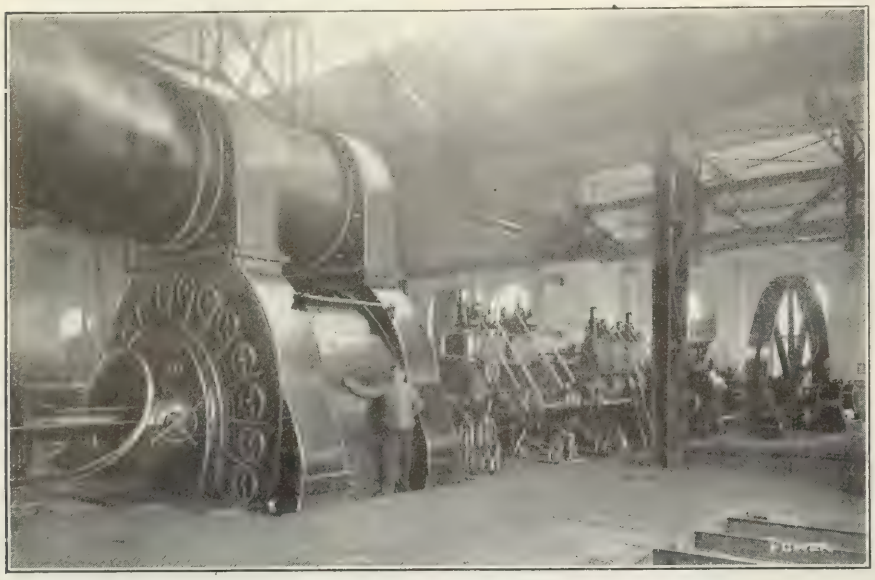


Fig. 1. Nürnberg Gas Engine and Blower, at Röchling Works

The gas from a blast furnace is said to represent about 34 per cent. of the total energy of the furnace. If this gas be burned under steam boilers which have a low yield, only approximately 4000 boiler horsepower can be obtained from a furnace producing 250 tons daily. The same gas in a gas engine would develop about 10,000 hp., according to results obtained with Nurnberg engines. Where the machines throughout the shops are steam operated, the loss in the piping is considerable for an extensive plant, and the economy of steam engines used on intermittent runs is low, so that probably not over 2000 hp. is realized from the total 4000 hp. of the gas-fired boilers. On the other hand, the 10,000 hp. from gas engines operating on one blast furnace, after deducting, say, 2500 hp. for running blowing machines, cranes, pumps and other machinery, leaves 7500 hp. to convert into electricity for operating rolling mills and other machines. This power may also be used in electric furnaces, or possibly be made a source of additional income by selling current.

The problem of operating gas engines from coke-oven gas is somewhat more complicated, owing to the different types of furnaces in use, and although the gas is not as suitable as that from blast furnaces, there are still several points in favor of using it in gas engines. From the waste gases of a recuperative coke oven burning 200 tons of coal per day, from 1500 to 1800 hp. can be obtained through gas engines. With the new gas producers of the Nürnberg type, using fine or waste coke which would otherwise be lost, considerable extra power can be obtained.

The Nürnberg gas engine is well adapted for operating blast-furnace blowers, and a typical installation of this kind is illustrated in Fig. 1, which shows the blower at the Röchling steel works, near Carlshütte, Germany. This plant also has two groups, each of 5000 hp., of Nürnberg engines operating on blast-furnace gas. The large blower is connected directly to a gas engine and has hollow disk pistons, allowing water-cooling when working at high pressures. As the blast-furnace blower has to run at variable speeds, the engine speed is controlled by changing the cutoff of the gas-inlet valve and by hit-and-miss governing.

An important installation of these engines is in the Georgs-Marie mining and metallurgical works, near Osnabrück, Germany, Fig. 2, which includes six gas engines, giving a total of 8990 hp. These engines are connected directly to direct-current dynamos, which furnish electricity

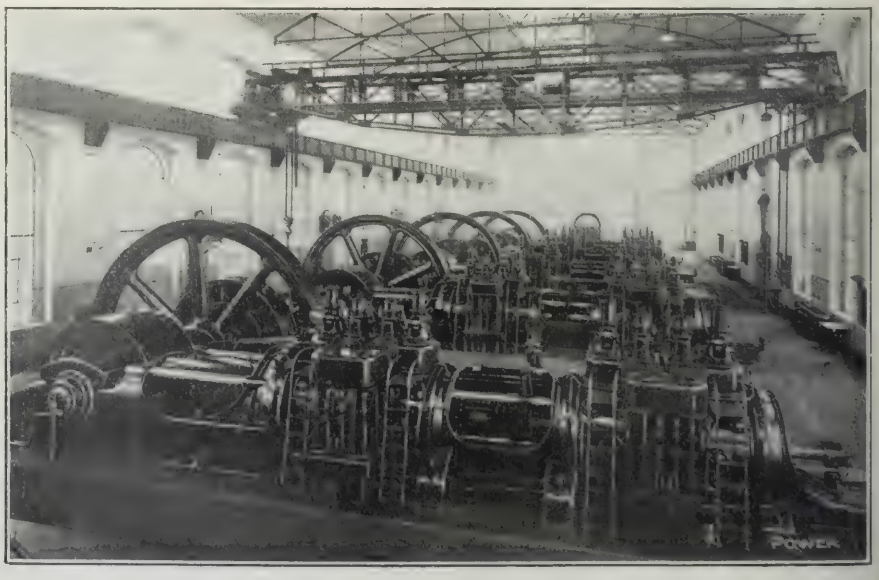


Fig. 2. Plant at Georgs-Marie Mine

for use on the premises. Another German plant using blast-furnace gas is the Rombach metallurgical works, at Lorraine, which has engines aggregating 29,-

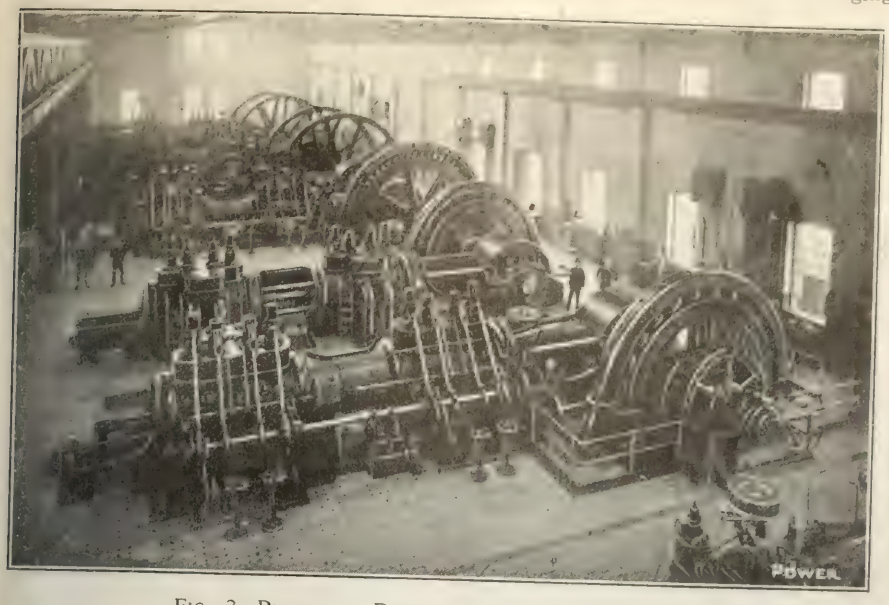


FIG. 3. PLANT OF ROMBACH METALLURGICAL WORKS

120 hp., shown in Fig. 3. Some of the engines drive generators having a combined output amounting to 11,500 kw., and the others are connected to blowers

Elyria Oil Engine

An oil engine embodying many of the advantages of the Diesel type, yet at a slight sacrifice in economy, avoiding the mechanical refinements, excessive weight and the auxiliaries of the latter, is being built by the Elyria Gas Power Co., Elyria, Ohio, in sizes from 25 to 150 hp.

Although the operating principle is similar to the Diesel, several departures have been made in its application. Chief among these is the use of lower pressures and an unjacketed cylinder head and false piston end, with which it is said to be possible to burn the heaviest fuels with a maximum pressure not exceeding 400 lb. per sq.in. without missed ignition, so common at light loads.

The two-stroke cycle has been adopted to decrease the weight and consequently the price per horsepower, and to simplify the construction from the standpoint of the user by eliminating the valves and valve-gear from the working cylinder. Although there is a tendency toward lower thermal efficiency, due to lower compression, this is compensated through having less jacketed surface exposed.

Some sacrifice would be expected in a modification of any well established type, and the Elyria engine has a fuel consumption somewhat greater than the standard four-stroke-cycle Diesel engine of the same size. The makers claim, however, that this is not due to any loss in thermal efficiency but rather to a lower mechanical efficiency, because of the extra work in scavenging the cylinder.

As Fig. 3 shows, there are two pistons —the working piston and the scavenging or differential piston; the displacement of the latter is approximately 1.9 times that of the former. The two pistons are separate castings held together by four studs. The wrist-pin is carried in the forward or differential piston, producing

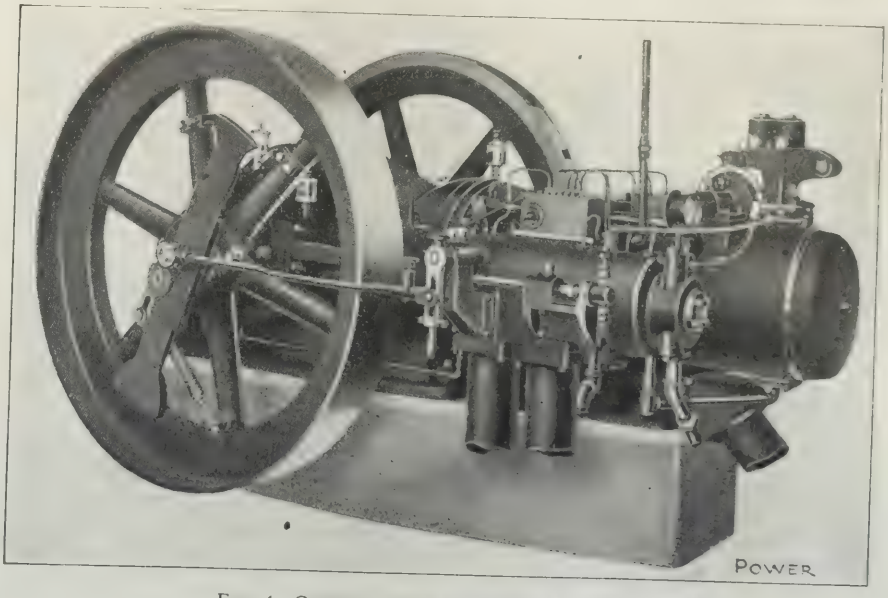


FIG. 1. GENERAL VIEW OF ELYRIA OIL ENGINE

the effect of a crosshead, as all side thrust is relieved from the working piston. To take out the working piston requires only removing the cylinder head, which is free from piping, and the four nuts which secure the two piston castings together.

The piston head is a separate piece which tends to maintain a high temperature and incidentally permits symmetry of the piston casting. The six rings used are accessible for inspection by removing the cap over the inlet port at the top of the cylinder.

To provide air for injection, three stages of compression are employed; the first is effected by the differential piston and the second and third by a separate compressor eccentric-driven from the main shaft, as shown in Fig. 2. Through the pipe shown in this illustration, air is taken past the piston valve and into the differential cylinder. It is here compressed in the first stage to about 5 or 6 lb., at which pressure the scavenging is effected.

The larger cylinder of the compressor takes air at this pressure from the scavenging, or differential, cylinder and compresses to 175 or 200 lb.; this air is delivered to a receiver alongside the machine, which is also the starting tank.

The third stage air (that used for fuel injection) is taken from this middle-stage tank through the governor-controlled valve to the high-pressure cylinder of the compressor and thence direct to the injection valve. This pressure will vary from 500 lb. at light loads with easily atomized fuel, to as high as 1000 lb. per sq.in. with overloads and the heaviest fuels.

Among the advantages claimed for this means of air control over standard Diesel practice, are that the first-stage compression is delivered by the scavenging cylinder without extra mechanism; that only a short length of piping is needed and no tanks for storing high-pressure air; and that for all load conditions the air pressure and volumes are automatically controlled.

The air in the main cylinder is compressed to 400 lb. which is the maximum pressure in this cylinder as no explosion occurs. To provide against the contingency of loss of pressure in the receiver while the plant is idle, a small gas or

gasoline engine and compressor are furnished with each installation.

The fuel pump is driven by an ordinary Rites governor in the flywheel, which varies the stroke of the pump plunger, thus proportioning the amount of fuel to the load. The driving mechanism allows disconnecting the governor and operating the pump by hand to supply oil for starting. Similarly in shutting down, the oil supply is thereby immediately cut off.

When starting the engine cold, gasoline is sprayed into the cylinder through the fuel valve and, to insure combustion, batteries, a vibrator coil and plug are provided. No commutator is used, consequently, a continuous shower of sparks is thrown into the cylinder, which serve to augment the heat of compression and insure burning.

Within 60 sec. from the first introduction of gasoline it is claimed the switch can be thrown out, the heavy oil immediately turned on and full load thrown on the machine. The working piston is lubricated from a force-feed pump, which also furnishes the small amount of oil required by the compressor cylinder. All

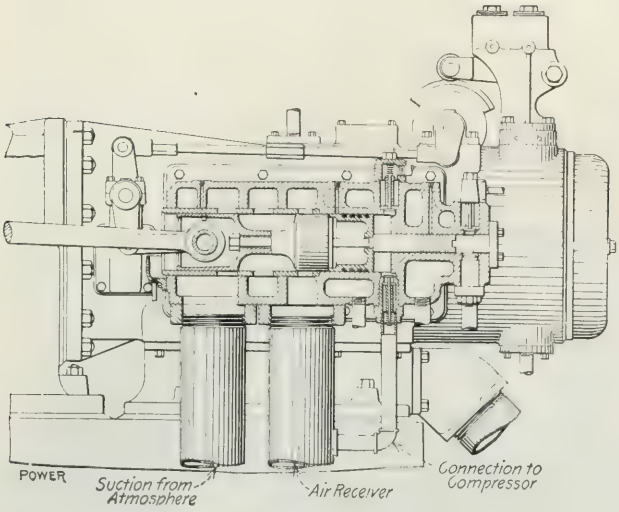


FIG. 2. SHOWING SECTION THROUGH AIR PUMP

important bearings, such as main bearings, crank-bearing eccentric, scavenging piston, or crosshead, and crosshead pin, are lubricated by a continuous oiling system. Oil from an overhead tank after filtering is fed to the bearings, from which it runs to a sump tank and is then pumped back to the overhead tank.

The cooling system is simple, having only one inlet and outlet. All water is admitted around the compressor cylinder from which it goes to the working cylinder and is there discharged. According to the manufacturer, with water entering at 60 deg. F., only 3 gal. per horsepower-hour is required.

This engine is being marketed under a fuel guarantee of 0.65 lb. of oil per brake horsepower at rated load and not to exceed 0.75 lb. at half load.

A Diesel Engine Explosion

By Arthur H. Allen

A singular accident occurred on July 10, at the municipal plant at Bray, a small town in Ireland. A Diesel engine of 50 hp. was being started up, when a violent explosion took place, killing one man outright, blowing a foot off the chief engineer and injuring his assistant. According to the latter, the compressed air in the starting reservoir had failed and, lacking means to compress a fresh supply, a 40-cu.ft. cylinder of compressed oxygen was obtained for starting the engine—a course recommended by the makers of the engine. The blast chamber was charged to 700 lb. pressure from this cylinder, and one of the starting reservoirs was also charged to about 350 lb. The assistant then started the engine in the usual manner, and, going on the platform, put the starting lever into the running position. He noticed that the engine did not appear to be running properly, but before he could ascertain what was wrong, the fuel valve in the cylinder cover exploded; the flame shot along the blast pipe and reached the blast reservoir, which then exploded also.

There was nothing unusual in the use of oxygen, although carbon dioxide is perhaps preferable, and doubtless if this gas had been used the explosion would not have occurred.

Several experts examined the plant after the explosion, and the general opinion was that it was due to the violent combustion of the oxygen with oil contained in the blast reservoir or in the passages connected with it. Some were of the opinion that using pure oxygen for starting was dangerous, although sanctioned by the engine builders. Without doubt the engine cylinder, the pipe connections, and the reservoirs were filled with air highly charged with oxygen—perhaps nearly pure oxygen—and the rise of pressure on ignition far exceded the normal pressure when working with air, overcoming the pressure in the pipes and reservoirs; but the fragments of the latter showed no indication of defective construction or material, and the necessary conclusion is that an explosion actually took place in the reservoir. The difficulty is to account for the presence there of volatile oil or combustible gas in sufficient quantity to explode.

Explosions have taken place in connection with the transmission of power by compressed air, due to using lubricating grease or oil, and apparently this case was similar. Presumably the oil had crept along the pipe into the air reservoirs from the air-compressor cylinder, or perhaps when the excessive pressure rise took place in the engine cylinder sufficient oil was blown back along the blast pipe to produce the result.

According to the latest information, there was no trace of oil on the fragments of the receiver, but this may have been due to the intense heat and violence of the explosion. That the connection from the blast receiver to the fuel valve was a copper tube 14 ft. long and only ⅜ in. in diameter, indicates that the explosion could not have been due to transmitted pressure, but flame could certainly travel along such a pipe. Normally, there is an almost continuous stream of air from the receiver to the fuel valve; therefore fuel oil could not creep along the pipe into the receiver. More probably the oil came from the air-compressor cylinder, the lubricating oil from which might reach the receiver either as vapor or by creeping.

In any case, the incident conveys a warning that the presence of oil in the air reservoirs is dangerous, and that in starting a Diesel engine in an emergency with "bottled gas," it is wise to use carbon dioxide rather than pure oxygen.

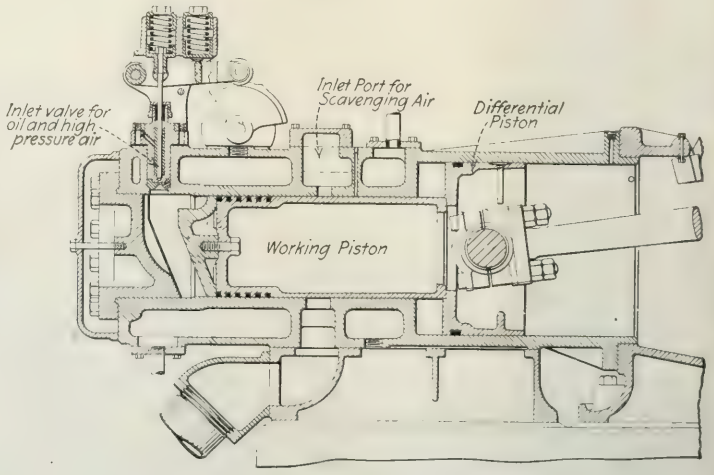


FIG. 3. SHOWING SECTION THROUGH PISTON AND PORTS

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Means for Accumulating Refrigeration—III

By Charles H. Herter

When an ice-machine builder is requested to furnish a refrigerating plant to be operated for, say, 12 hours per day in summer, with no brine pump moving during the night, and the temperature is stipulated not to rise more than a certain number of degrees Fahrenheit during the stop, he resorts to the use of brine tanks. In these the brine is lowered in temperature when the machine is in operation or some ice is formed and in this way refrigeration is accumulated for use during the night.

In the second article it was shown that when the insulation is placed outside the building walls, or a brick lining is erected, they acquire low temperatures and act as barriers to the incoming heat, because a portion of this heat is expended in raising the temperature of the walls. A more common method for preventing a 3-deg. temperature rise when the machine is shut down for some time, which for many products is the limit of safe permissible variation, is to use brine-storage tanks. Even this is practicable only on a small scale. In large installations refrigeration is either maintained continuously day and night, or a large body of brine is cooled off during the day and circulated by a pump through pipes in the rooms during shutdown. With the latter, direct-expansion ammonia pipes are usually abandoned and the plant operated on the brine-pipe system.

Brine Storage Tank System

In the following a brine-tank equipment will be laid out for the 12,000-cu.ft. meat room referred to in the first article. The heat leakage into this room has been calculated at 300,000 B.t.u. per 24 hours. Besides maintaining the temperature of the room it may be necessary to cool each day one-fourth of the contents, or 12,500 lb. of meat, received from refrigerator cars, through a range of 15 deg., which requires

$$12{,}500 \times 15 \times 0.8 = 150{,}000 \ B.t.u.$$

An allowance of 50,000 B.t.u. will cover the additional work required for renewal of air, heat due to presence of workmen, lights, etc. The total work in the day time will be 350,000 B.t.u. and at night 150,000 B.t.u.

Brine-storage tanks are usually operated without ice being formed therein intentionally. While by taking advantage of the latent heat of fusion of frozen brine much more refrigeration can be stored per cubic foot of brine, certain difficulties make this plan undesirable for most plants. However, the theoretical analysis may be interesting, and will be given later.

For economy, a plant should always have enough cooling surface to allow operating with a suction pressure and evaporating temperature as high as permissible. In the present case, owing to the need of cold brine in tanks, this cannot exceed 15 lb. gage, the ammonia vapor being then at —1 deg. F.; otherwise the temperature differences become so small that the brine tanks and cooling pipes require inconveniently large surfaces, and the plant is not large enough to warrant operating with two different suction pressures.

During the night the air rises from 32 to 35 deg. and the calcium-chloride brine from, say, 6 to 22 deg. Thus a cubic foot will take up

$$(22 - 6) \times 52 = 832 \ B.t.u.$$

the necessary brine capacity being

$$\frac{150{,}000}{832} = 180 \ cu.ft.$$

which is 1 cu.ft. per 66⅔ cu.ft. of room volume. To hold this quantity, use six tanks each 10 ft. long, 12 in. wide and 3 ft. deep. Longer tanks are inconvenient for transporting and erecting, while assembling in place is usually out of the question, owing to the lack of working space in the bunkers or coil lofts. The cooling surface of the four walls of each tank is 66 sq.ft., or a total of 396 sq.ft. During the night the true mean temperature difference between the air and the brine is 18.85 deg. The coefficient of heat transmission k in B.t.u. per square foot per hour per degree temperature difference will therefore be

$$\frac{150{,}000 \ B.t.u.}{12 \ hr. \times 396 \ sq.ft. \times 18.85 \ deg.} = 1.674$$

The above method of computing the available tank cooling surface and coefficient k is as yet not followed by all refrigerating engineers. Those who shirk going into "details" or are anxious to hand in an attractive bid, usually specify only one or at most two tanks for the whole body of brine. Manufacturers' illustrations show only one tank, although the necessary cooling surface is not then obtained, the dead brine remote from the side of the tank not being able to do its share in absorbing heat. With the weak air circulation realized, each square foot of tank surface can only absorb heat at the rate of from 1 to 2.5 B.t.u. per degree per hour, the exact value varying with the amount of difference and extent of frost on the surfaces. In the foregoing example a value between these two figures has been taken.

For satisfactory results it is well to observe the temperature of the brine as it offers a convenient means for determining the work done. For this a long-stem thermometer is very useful, but if not available, an ordinary short thermometer may be set into a small open-top tin can fastened to the end of a stick in the manner of a milk measure, and the whole submerged. By keeping the can for a few minutes at a certain spot the true temperature prevailing there will be obtained because the brine in the can will then have acquired the temperature of the particular portion of brine in the tank.

If the thermometer admits of close reading, a difference in temperature will be detected at various points of the tank owing to the absence of good circulation. When stirring the brine, an average temperature is obtained. The purpose of the can is to keep the bulb of the thermometer in contact with brine which has acquired the temperature of the brine whose temperature is to be ascertained; otherwise the bulb on being lifted out of the tank and exposed to the air will at once become cooled, due to evaporation. It also prevents the thermometer from being influenced by a stratum of possibly colder brine encountered while on the way to the surface. This method of taking the temperature of liquids can be used also with freezing tanks, storage vats, etc.

A pipe connection should be arranged near the top edge of each tank, unless they are intercommunicating when a single pipe will do, not so much to facilitate filling as to serve as an overflow; a similar connection with a valve should be provided for draining. Tanks must always be at least 6 in. from the nearest wall to allow room for air to circulate. The tanks in the bunker overhead should be so arranged that the natural flow of the air is lengthwise with the tanks and not at a right-angle thereto.

A substantial insulated bunker floor is usually provided, even where no tanks are used, inclined down toward the down shaft at a slant of ½ in. per foot or more, to allow the drip water from

the pipes on thawing to get to the drip gutter as fast as possible to avoid humidifying the air, and from there out through the waste pipe, which is 2 in. or larger. This pipe must have an easily accessible waste-pipe trap under the bunker to prevent escape of cold air. The bunker floor must be water-tight to protect the goods stored and to keep moisture out of the insulation.

For covering the floor, zinc, sheet lead, copper or galvanized iron, with either white or no painting, is preferable to built-up roofing, because it is not so easily pierced and is capable of absorbing much less radiated heat. With the air temperature at zero and below, the metal covering, if not free to move, is apt to cause trouble owing to contraction, and then a calked floor, roofing or asphalt may be used.

The object of insulating the bunker floor is to prevent dripping at the under side, and to assist the air circulation. As the cooling devices rest on timbers directly above the bunker floor, the timbers laid level and parallel to the air flow, the floor becomes chilled, and unless insulation, 2-in. thick corkboard or its equivalent, with no air spaces, is used to prevent the underside of the ceiling from getting cold, the fresh warm air entering through the door will deposit its moisture directly on the ceiling instead of on the frosted surfaces, and a "wet ceiling" is a nuisance. In primitive refrigerators the cold air descends from spaces between the bunker-floor joints, the air eddying around in the upper part of the storage space to the detriment of uniform temperature.

A positively guided circulation of air should always be provided for. The cold air, to do effective work, must be admitted to the storage space at a low level, sweep over the goods, become warmed and gradually rise to the bunker to be recooled. With no fan, circulation is brought about by the difference in temperature of the cold and warm air, but as this difference is only 1 to 2 deg., the difference in height of the respective columns of air must be taken advantage of. The simplest way to do this is to provide an apron, insulated with, say, 1-in. corkboard, forming with the wall a down shaft reaching within 2 ft. of the floor. On the opposite side is a similar apron, insulated with 2-in. corkboard to prevent early cooling of the warm air, forming the up-duct and extending only from the bunker floor to within a distance of the ceiling, leaving an area equal to the free area of the up-duct. This duct should be along the warmest wall of the room where the air rises naturally, so as to favor the natural air circulation.

It is also better if the underside of the bunker floor, with either no beams projecting or with the beams parallel to the air flow, slants from the cold-air toward the warm-air duct. Thus the vertical distance from the cold-air inlet to the warm-air outlet is as great as possible, resulting in a noticeable energetic air circulation which leaves the room dry and sweet.

The up-and-down air duct areas should each equal not less than 10 and, if possible, 15 per cent. of the ceiling area. In the room under discussion, measuring 20 x40 ft., a down shaft, 40x3 ft., should be provided in the center of the room, and one up shaft, 40x1½ ft., along each side wall. Instances might be cited where in rooms exceeding 16 ft. wide only one up-duct and one down-duct, each 12 in. wide, were provided and where the temperature in the storage space remained 10 or more degrees higher than in the loft, because the circulation was choked.

Sometimes the brine tanks are made narrower than 12 in. and hung all around the walls. This saves the cost of the insulated bunker floor, drip gutters underneath being sufficient to catch the meltage, but the air circulates around in eddy currents and wall space must be sacrificed. It is, however, the only arrangement that can be used when the room is not high enough for a bunker.

Occasionally several brine-storage cylinders are hung from the ceiling or walls, made of 12 in. spiral riveted pipe and containing a loop of ammonia pipe passing through stuffing-boxes in one of the heads. This arrangement is effective, but wooden gutters lined with sheet metal must be used to catch the drip. The cylinders being closed, the density of the brine remains unaltered. Like all refrigerating surfaces, the cylinders must be kept about 12 in. from the ceiling to prevent condensation thereon of the vapor in the air. A pet-cock to show overfilling should be used.

As during the day, the brine tanks contribute to the work of refrigerating, sufficient submerged cooling piping must be provided to abstract, say, 300,000 B.t.u. every 12 hours. The logarithmic mean temperature difference between the ammonia and brine is 13.45 deg., and the heat transmission for 1¼-in. ammonia pipe (not using the "flooded" or gravity-feed system of circulation) submerged in still brine is about 4 B.t.u. per hour per degree per lineal foot. Accordingly

$$\frac{300{,}000\ B.t.u.}{12\ hr. \times 13.45 \times 4} = 465\ ft.$$

of pipe is necessary, or 78 ft. per tank Continuous electric-welded oval 8-ft. coils can be used, wound 8 in. on centers, 5 pipes high, 5 in. on centers vertically.

The remaining 200,000 B.t.u. of day work can be done by exposed direct-expansion ammonia pipe. Taking the average temperature difference at only 30 deg., and the heat transmission at 1.1 B.t.u. per lineal foot of 1¼-in. pipe per hour per degree difference, the amount of pipe needed is

$$\frac{200{,}000\ B.t.u.}{12\ hr. \times 30\ deg. \times 1.1\ B.t.u.} = 505\ ft.$$

As the frost on this piping can thaw off every night, no additional piping need be reserved for defrosting in day time.

For flexibility in operation a separate circuit should be arranged for the submerged and for the exposed set of piping, each set fed by one expansion valve.

It is to be hoped that the above rational method of arriving at the amount of piping will ultimately supersede the prevailing rule-of-thumb custom of using "ratios." By factors expressing ratios are meant arbitrary allowances, such as cubic feet of brine, or cubic feet of space per foot of pipe, feet of pipe per ton of refrigeration, etc. Such methods are unscientific, because no set of conditions can be considered as standard. The transfer of heat to a pipe is influenced only by k, the coefficient of heat transmission, and by the mean difference in temperatures.

Congealing Tank System

As already stated more refrigeration can be stored up per cubic foot of brine volume in tanks if a portion of the brine is frozen. The tanks can be thinner, which is an advantage if they must be hung along the walls on account of low head room.

To operate the machine with the same economy as with the brine-tank system first calculated, the suction pressure must be not less than 15 lb. gage, giving an evaporating temperature of — 1 deg. F. in the expansion coils. When determining the dimensions of tanks in which ice is to be formed, the following experience with the standard can system of making ice should be kept in mind:

In the fall or winter, when the demand for ice is less than the daily capacity of the factory, the machine is sometimes shut down after all cans are frozen solid, supposing the ice-storage room to be already full. If meantime the brine is allowed to warm to 32 deg., the blocks in the cans will commence to melt. Ice being lighter than water in the proportion of 57.4 lb. to 62.4 lb. per cubic foot, the ice will soon float, so that 6 in. of water might be under the melting block, and an average of 1 in. of water around its four sides.

If, now, the brine is cooled to save the ice, the upper part of the blocks will freeze to the can before the water in the bottom has solidified. The water during freezing expands, and being confined the cans will first be bulged out of shape, causing the seams to leak, and when freezing is continued, rupture must result. Cases are known where several hundreds of cans have been burst in this manner—to the discomfort of the engineer, for a good steel galvanized can for 300-lb. blocks costs about $3.50. The remedy is to keep the brine agitated and not let it get over 28 deg. F., or not

to freeze up the cans in the first place.

To guard against the trouble, the congealing tanks must be built so that the requisite quantity of brine-ice will grow on the cooling coils at a safe distance from the sides. For example, suppose the tank to be 5 in. wide inside, 10 ft. long and the brine 3 ft. deep. These tanks are usually of galvanized steel $\frac{1}{16}$ to $\frac{1}{32}$ in. thick, surrounded by pairs of angle irons every 2 ft. to keep the tanks in shape. Into such a tank a direct-expansion cooling coil of 1¼-in. ammonia pipe can be placed, the bends at the ends being made to a radius of 6 in. to the center of the pipe. As it is necessary to crowd in as many loops as possible the pipes are not run parallel, but zigzag like a steel band folded up. Thus a coil can be used at least 8 pipes high and 9 ft. 6 in. long, the total submerged length being about 80 ft. The brine volume of one tank is 12.5 cu.ft. The four walls of each tank represent 62.5 sq.ft. of effective heat-absorbing surface; six tanks have 375 sq.ft. The duty of each square foot during the night is

$$\frac{150{,}000\ B.t.u.}{375 \times 12\ hr.} = 33.3\ B.t.u.\ per\ hour$$

Taking for k the value already applied to brine-storage tanks, the temperature difference necessary is

$$\frac{33.3}{1.674} = 19.89\ deg.$$

which is about 1 deg. more than with unfrozen brine. To satisfy this condition, the brine temperature may be 8 deg. in the evening and rise to 19.4 deg. by morning. Each of the six tanks must then absorb

$$\frac{150{,}000\ B.t.u.}{6} = 25{,}000\ B.t.u.$$

As 11.58 cu.ft. of salt brine taking up

$$19.4 - 8 = 11.4\ deg.$$

abstract only

$$11.58 \times 11.4 \times 58.4 = 7709\ B.t.u.$$

the deficiency of

$$25{,}000 - 7709 = 17{,}291\ B.t.u.$$

must be supplied in the form of latent heat when the brine-ice melts at 8 deg. F. The writer knows of no tests made to ascertain this quantity. The latent heat of fusion of a pound of brine-ice is certainly less than 144 B.t.u., the latent heat of pure water ice, because less than a pound of water is present; on the other hand, probably salt crystals are expelled in the process of freezing. In the absence of exact data, the following calculation is probably close enough for the purpose:

For an 8-deg. F. freezing point use a 17.5 per cent. solution of salt, not chloride of calcium because the heat capacity would be 8 per cent. less. A cubic foot of this solution will weigh 70.56 lb., 12.35 lb. of which is dry salt, leaving 58.21 lb. of water. The latent heat of this is

$$58.21 \times 144 = 8382\ B.t.u.$$

per cubic foot of solution; but when frozen, the volume is about 8½ per cent. larger, thus the latent heat becomes

$$\frac{8382}{1.085} = 7725\ B.t.u.$$

per cubic foot of brine-ice. For 17,291 B.t.u., 2.237 cu.ft. must be accumulated on the 80 ft. of 1¼-in. pipe. This means 48.25 cu.in. per lineal foot, equivalent to a coating of ice uniformly 0.564 or $\frac{9}{16}$ in. thick.

The next step is to ascertain whether the 80 ft. of pipe is adequate to do both the day and night work. After the 12-hour shutdown all the brine-ice will have melted and warmed to 19.4 deg. F. This brine must first be cooled again to 8 deg.; requiring 7709 B.t.u. per tank. But while this heat is being abstracted, more heat is being supplied to the brine by the frosted tank surface, which extends the cooling period to about three hours, as will be proved.

In three hours the tank surface absorbs

$$62.5\ sq.ft. \times \left(32 - \frac{19.4 + 8}{2}\right) \times 1.674 \times 3 = 5744\ B.t.u.$$

Adding this to the 7709 B.t.u. brine cooling makes the total 13,453 B.t.u. The submerged piping can absorb heat at the rate of

$$80\ ft. \times \frac{19.4 + 8}{2} \times 4\ B.t.u. = 4384\ B.t.u.$$

per hour, and

$$\frac{13{,}453}{4384} = 3.07\ hr.$$

As during the remaining nine hours of running the brine is at 8 deg. and the air at 32 deg., the tank surface may be credited with absorbing

$$62.5\ sq.ft. \times (32 - 8) \times 1.674 \times 9 = 22{,}300\ B.t.u.$$

Operating now with only 9-deg. temperature difference, the capacity of the 80 ft. of pipe is reduced to

$$80 \times 4 \times [8\ deg. - (-1\ deg.)] = 2880\ B.t.u.\ per\ hour$$

$$\frac{22{,}600}{2880} = 7.85\ hr.$$

of work. This leaves only 1.1 hours for congealing the brine, which alone was found to require 17,291 B.t.u. per tank. Now the heat conductivity through the wall of an extra-heavy 1¼-in. iron pipe covered with 0.564 in. of brine-ice, not water-ice, may be calculated at

$$1 \div \left(\frac{0.194}{476} + \frac{0.564}{12}\right) = 21.1\ B.t.u.\ per\ hour$$

and that through the same pipe wall, but covered with 0.564 in. of salt brine of 1.131 specific gravity at 60 deg. F.

$$1 \div \left(\frac{0.194}{476} + \frac{0.564}{3}\right) = 5.3\ B.t.u.\ per\ hour$$

The comparative conductivity in the two cases is therefore as 3.98 is to 1, and since k for 1¼-in. ammonia pipe to brine is 4 B.t.u. per hour per lineal foot, k for ammonia to brine-ice should be 15.92. To prove this estimate correct this value may be compared with the heat transmission secured in the process of can-ice making:

With brine 9 deg. below 32 or at 23 deg. F., 0.564 in. of ice is formed in

$$\frac{28 \times 0.564^2}{9 - 0} = 0.99\ hr.$$

The heat abstracted per square foot of ice 0.564 in. thick is about

$$\frac{144 \times 0.564}{30.1} = 2.7\ lb.$$

$$2.7\ [(68 - 32) \times 1 + 144 + (9 \times 0.5)] = 498\ B.t.u.$$

The mean temperature during the process is

$$\frac{\left(2.7\ [(50 - 32) \times (68 - 32) + (9 \times 144) + (4.5 \times 4.5)]\right)}{498} = 10.65\ deg.$$

Consequently

$$k = \frac{498}{10.65} = 46.8\ B.t.u.$$

per square foot. This is for 1 sq.ft. of freezing surface per square foot of ice sheet. The surface of one lineal foot of 1¼-in. extra-heavy pipe based upon its mean diameter is only 0.377 sq.ft., it should therefore absorb

$$0.377 \times 46.8 = 17.64\ B.t.u.$$

per degree difference, which is 1.72 B.t.u. better than 15.92, the figure assumed, and is unquestionably due to the influence of agitation in the freezing tank, which increases the rate of heat transfer.

Each lineal foot can therefore abstract

$$15.92 \times 9\ deg. = 143.28\ B.t.u.$$

per hour, or 157.6 B.t.u. in the 1.1 hours available for freezing. The 80 ft. will thus store up

$$157.6 \times 80 = 12{,}608\ B.t.u.$$

of latent heat, leaving a shortage of

$$17{,}291 - 12{,}608 = 4683\ B.t.u.$$

It will therefore be advisable to add a ninth pipe to the coil of each tank, which will do freezing for a total period of, theoretically,

$$\frac{4683}{9.5\ ft. \times 143.28} = 3.44\ hr.$$

The day work has been figured at 350,000 B.t.u. During the first three hours each tank absorbs 5744 B.t.u., and during the remaining nine hours, 22,600 B.t.u., total 28,344, or 170,040 B.t.u. is the work of the six tanks, leaving 179,960 to be absorbed by exposed direct-expansion ammonia piping. Using the same values as with the brine-tank plant, the length of the 1¼-in. piping required is

$$\frac{179{,}960}{12\ hr. \times 30\ deg. \times 1.1} = 454\ feet$$

The tabular comparison shows plainly that, contrary to usual expectations, there

is practically no difference in first cost and operating expense between the two systems. With the storage tanks it is easy to make the brine dense enough not to freeze, and when the brine has been cooled to 6 deg. F. in the above example the engineer knows that he can stop. But with the congealing tanks it is a trick to keep the brine always just dense enough to freeze at 8 deg. No one can tell whether sufficient brine has been frozen, or whether the thin tanks will be bulged out of shape, due to uneven freezing.

A COMPARISON OF BRINE STORAGE AND BRINE CONGEALING TANKS

	Brine Storage Tank System	Brine Congealing Tank System
Operating period, hours daily	12	12
Ammonia suction pressure, lb. gage	15	15
Number of tanks required	6	6
Size of tanks	10x3.5x1 ft.	10x3.5 ft.x5 in.
Effective tank surface, sq.ft	396	375
1¼ inch piping, submerged, ft	465	537
1¼ inch piping, exposed, ft	505	450
1¼ inch piping, total, ft	970	987

The calculations herein, which are more intricate with congealing tanks than with brine-storage tanks, prove that the limitations imposed by the low coefficient of heat transfer prevent a coating of more than ¾ in. uniform thickness of brine-ice to be formed on the pipes. The amount of refrigeration that can be accumulated is therefore much less than that corresponding to a tank full of ice; on the other hand, the tanks, if preferred, can be decreased in width to 3 in. As the brine is constantly absorbing moisture from the air and from frosted branch pipes, it becomes diluted and freezes at a higher temperature than desired, it being too troublesome to maintain the solution at a fixed strength.

It has been suggested to fit covers on top of these tanks to prevent the admission of moisture, but it is impossible to prevent this entirely; moreover, the removal of such moisture helps to improve the preserving quality of the cold air, for which purpose direct-drying agents, such as lumps of chloride of calcium, are often used. It is advisable, however, to prevent foreign moisture, such as comes from thawing pipes, from dripping into the brine. For these good reasons brine-storage tanks are seldom operated as congealing tanks. In drinking-water systems ice is frequently allowed to collect on the submerged cooling pipes, 144 B.t.u. being then stored up for every pound of ice so formed.

Liquor in the Cooler

Some textbooks state that there should not be any liquid in the cooler, but the orders here are to keep just enough liquid in the receiver so that the outgoing line to the expansion valve is sealed, and at the same time maintain as much liquid in the cooler as possible. I would like to have this subject explained thoroughly.

Is there any cheap substitute for mercury that can be used in thermometer wells? These questions were recently asked by a correspondent and the following answers may be of general interest:

The best practice in operating brine coolers or refrigerating apparatus of any kind, is to flood it with liquid ammonia as much as the construction of the plant will allow. For instance, with a shell-type brine cooler the maximum efficiency will be obtained if the liquid ammonia in the cooler is carried at as high a level as possible consistent with the proper operation of the ammonia compressor. The presence of too much or too little liquid in the brine cooler to fulfill the above requirements can be determined from the temperature of the discharge from the ammonia compressor. For a suction pressure of about 16 lb. per sq.in. gage, and a condenser pressure of 185 lb. gage, the discharge temperature should be about 250 deg. F. With too little liquid in the cooler this temperature will rise and with too much liquid, will fall. Different suction and condenser pressures will give other discharge temperatures, but the proper figure can be approximated by remembering that for a constant condenser pressure the discharge temperature increases with a decrease in the suction pressure and decreases with an increase in the suction pressure. For a variable condenser pressure and constant suction pressure the discharge temperature varies with the condenser pressure. For example, for a 20-lb. back pressure and 185-lb. condenser pressure the proper discharge temperature is about 235 deg. F. For a 10-lb. suction pressure and 185-lb. discharge pressure, the proper discharge temperature is about 270 deg. F. For a 20-lb. suction pressure and 165-lb. condenser pressure the proper discharge temperature is about 215 deg. F. and so on.

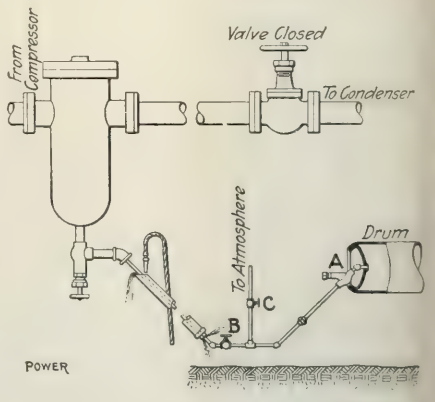
A cheap substitute for mercury in thermometer wells is oil. The oil must be selected so that the temperature range in the mercury well under consideration is not above or below the flash point and the congealing point of the oil respectively, that is to say, for measuring low temperature, cold test oil must be used, and for measuring high temperatures, high flash point oil.

CORRESPONDENCE

Home Made Ammonia Condenser

Last fall I had occasion to withdraw the ammonia from a 10-ton refrigerating system, the condenser of which was out of repair. The lessee of the property did not care to go to the expense of repairing it, so I made the condenser shown in the sketch and succeeded in saving nearly all of the ammonia.

I blew out the oil separator in the discharge line from the compressor, and put a 45-deg. elbow on the 1¼-in. horizontal pipe leading from the angle blow-off valve. At this point I reduced to a ⅜-in. pipe and ran it diagonally to the ground, a distance of about 9 ft. I placed the small pipe inside a piece of 1½-in. pipe, capped on the lower end, with a hole drilled in the cap large enough to allow the ⅜-in. pipe to pass through and leave about 3/32-in. clearance all around. The ⅜-in. pipe was then connected to an empty ammonia drum and a tee and riser to atmosphere so placed that when the drum was full I could shut off the ammonia by the valves *A* and *B* and let that contained in the pipe between the two valves escape through valve *C*. This prevented any ammonia coming in contact with the hands when the union was broken. The drum was placed on scales so that the amount entering could be easily weighed.



HOME-MADE AMMONIA CONDENSER

A hose was placed at the top of the jacket and allowed to discharge just fast enough to overflow. The compressor was a single-acting two-cylinder machine and by cutting out one side and running very slow I was able to obtain good results. It being late fall I had the advantage of good cool water for condensation.

G. B. LONGSTREET.

Somerville, Mass.

"A cubic foot of water heated to the temperature of steam under 70 lb. pressure contains as much energy as a pound of gunpowder," says a contemporary. Those operating old and corroded boilers might well take this to heart.

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POWER

New York City's Smoke

"Why should it be necessary to exhort the Board of Health to do its duty in the matter of quelling the smoke nuisance?" asks the New York *Times*. Architects of the city, according to the *World*, state that "New York will become as black as London unless a stop is put to the soft-coal smoke nuisance." The "increasing and practically unrestricted use of bituminous coal in the public-service power plants and large factories, apartment houses and Municipal, State and Federal buildings" is vigorously condemned.

The real reason for this is that the city has no sane system of looking after the cause of its affliction. It does not use, as far as we can ascertain, a chart such as the Ringelmann to determine the density of smoke that an offending chimney emits. This is left to the discretion of inspectors of the Department of Health, who, by the way, are not strictly smoke inspectors, as such duty is only incidental to others that they perform. Think of it; the world's second largest city smoke-inspectorless and wondering why it is smoky!

Considering that some of the public buildings are the greatest offenders it is no wonder that the smoke nuisance is general and on the increase. And no one is to blame but the city.

Most soft coals, except the high-grade semibituminous sorts, are cheaper than anthracite. Besides, due to its higher quality, the price of anthracite is kept high by its slow supply into the city. The introduction of mechanical stokers continues to increase and so, too, must the use of soft coal.

The smoke-inspection department of Chicago has made elaborate experiments in the smokeless combustion of many grades of coal in many kinds of furnaces. It also coöperates with the plant owners, giving expert advice as to the kind of coal and furnace best suited for the work. As told editorially (Nov. 7, 1911), Boston's smoke-inspection department does the same thing.

Heretofore in Chicago a chimney emitting smoke of No. 3 density (Ringelmann chart) for 7 min. during an hour was considered a nuisance. The total quantity of smoke was not considered; hence some owners were likely to be unjustly punished. Now observation will be made on a given stack every 15 min. throughout an entire day and the average density thus determined. If smoke of more than 2 per cent. density is emitted, the owner is appealed to by being shown that other plants with similar equipment are doing considerably better.

The smoke-inspection department of Chicago is there to advise and assist owners to obtain smokeless combustion and not to hale them before the court until these measures have failed.

New York may well take a few lessons from Chicago, and then demonstrate that it knows how to handle the smoke nuisance.

Who is an Engineer?

The old question as to who should be called an engineer has, like the proverbial bad cent, again "turned up." The daily press has voiced its opinion and still we are as far from a satisfactory conclusion as we were when the British military authorities in 1802 changed the title of "chief engineer" to "inspector general" of fortifications.

A correspondent in *Engineering News* suggests giving the name "energier" to those who hold degrees in mechanical and other lines of engineering, to distinguish them from those "who can use a monkey wrench and keep an engine running." In defense of the term "energier" the correspondent states: "The word already has a prototype in 'financier' (also a man of brains) and would not require much 'accustoming.'"

The parenthetical inference is that a man who can "keep an engine running" has little or no brains. If those engaged in modern power-plant practice are alluded to, the inference is the most effective boomerang that has come to our notice in some time. As for the term

"energier," it could as correctly be applied to a prize-fighter as any grade of engineer. The fighter skillfully directs and controls much energy, but there is little of the Greek "gennao," to beget or produce; or the Latin "inginium" or the English "ingenuity," about his work.

There was a time, before the advent of the Corliss engine, the problems of extensive high-tension electrical generations, steam-turbine practice and the chemistry of the boiler room, in a word, efficient power-plant management, when such engineering consisted chiefly in shoveling coal and "keeping an engine running" regardless of how it was running. That day has gone, and with it has gone that kind of an engineer—the engine-man, as our English cousins term him. It is ludicrous to even imagine a man in charge of modern power-plant equipment who can do little more than handle a monkey wrench, or who is, according to the correspondent's intimation, a man without brains.

There are men in plants who are mere starters and stoppers. They are known as such in the engine room and elsewhere. They are becoming more and more extinct and must continue to become so in the light of power-plant engineering progress.

The "Lusitania's" Turbine Troubles

A few hours after the "Lusitania" left Liverpool on a recent Western trip, one of her forward port turbines gave considerable trouble. This, according to the daily press, was due to a hot bearing. A passenger, in a position to know, states that the accident wakened nearly all on board and that the noise was caused by the tearing to pieces of the turbine blades.

This sounds more reasonable. A hot bearing would not disturb the whole ship or noncommission the turbine until the vessel had again reached Liverpool to enable the Parsons Company to make the necessary repairs.

Early in June this same turbine, we understand, stuck fast shortly after the vessel left the English port. The exigencies of the service do not allow the ship's crew or others time enough to make substantial repairs, and the recent accident may have been the direct result of inadequate correction of the last.

The trouble is probably due either to insufficient clearance between the periphery of the wheel and casing, or is the result of erosion and breaking of a blade or blades, which obviously work havoc in any turbine. The wheel in a Parsons turbine revolves in a medium of high density at the admission end. This means very high friction of the wheel against the medium and also leakage of steam which increases with the clearance between the rotating and stationary parts. This calls for a minimum clearance, and any appreciable reduction of it as caused by an excessively worn bearing will, if not remedied in time, permit contact.

The total number of vanes is also very high and the rotating and fixed blades increase in radial length and also in width as the exhaust end is approached, to pass the increased volume of steam.

Assume a common wooden lath mounted on a spindle as a turbine blade is mounted, except that the outer end of the lath is not held by shrouding. Let the lath be subjected to a high wind velocity. It will vibrate considerably and, if the wind velocity is high enough, the lath generally will break near the spindle. If the length of the lath is increased the danger of breakage is proportionately increased for the same wind velocity. The same action takes place in a turbine and, while the shrouding helps protect the blades against breakage, the vibration tending to break or loosen them from the spindle is nevertheless present.

It may be policy to delude the public about such happenings, but to endeavor to "pull the wool" over the eyes of engineers is not commendable. The Parsons is a good turbine, but confidence in it is not encouraged by concealing from engineers the troubles encountered in its operation.

False Economy

The superintendent of a large concern became interested in the efficiency of the power plant and decided to search into and stop the wastes. The wastes were more imaginary than real, however, as the engineer was thoroughly competent and obtained admirable economy. He belonged to engineering organizations from which he borrowed indicators, Orsats, etc., and he made good use of them in improving his plant.

This was all unknown to the superintendent, so he ordered the engineer to take indicator cards from all the engines every day. Later he learned a little about the diagrams and complained about poor compression, late cutoff and a variety of other defects that never existed. Naturally the engineer tired of this uncalled for criticism, knowing the engines were operating most economically. To put an end to it, he made tin templates of the diagrams that escaped criticism and thereafter all diagrams submitted to the superintendent were drawn from these templates. They were satisfactory, of course, and everybody was pleased.

The superintendent's attitude was an insult to the engineer, either way it may be taken, for it implied no confidence in his ability, or else presupposed that he would not do his best unless watched. The interference, amounting almost to persecution sometimes, that goes with this attitude, discourages good men and inclines them to resort to just such trickery as has been cited, to satisfy some misguided official.

Another poor policy is to hold firemen to a standard percentage of CO_2 which they must obtain or be discharged. A little better, but also having its disadvantages, is the bonus system. It is a temptation for the men to think of their own gain first, and strive for a record by illegitimate means. In their effort to obtain a high percentage of CO_2 they "jockey" the dampers and fires, which very often results in reduced steam pressure at a critical time, and nearly always means an excessive coal consumption. If bonuses and standards must be used, equip the boilers with recording thermometers and draft gages that will reveal fraudulent manipulation.

A man's job means much to him and he will hold it longer and do better work if not provoked to deception.

In the article on the Baumé hydrometer, page 150 of the July 30 issue, the figures were transposed in the composition given for one of the calibration solutions. The mistake occurred at the bottom of the second column and is very obvious. The solution should have been given as 10 parts of salt to 90 parts of water.

When a marine engineer tells you that "too high a vacuum does not pay," you can make up your mind that there's nothing in it (meaning the vacuum).

Readers with Something to Say

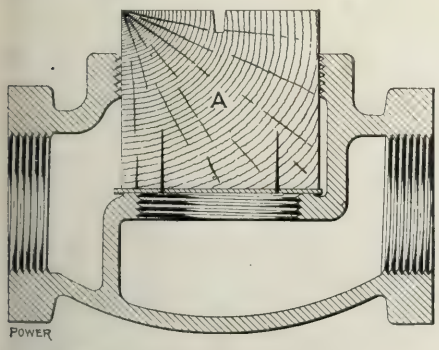
A letter good enough to print will be paid for. Ideas, not mere words, wanted

Simple Valve Reseating Device

For engineers of small plants that do not have a valve-reseating machine, the substitute illustrated will be found satisfactory.

Make a wooden plug *A* that will go loosely through the bonnet opening in the valve and bear squarely on the valve seat. Next, saw a slot in the top of the plug to engage a bit-brace screwdriver. Cut several pieces of emery cloth the size of the plug and tack them on the end of the plug, putting the tacks in the middle. The device is now ready for use.

Put the plug in the valve body and turn it with the brace. When the emery cloth fills with brass, tear it off and another clean surface is exposed. The plug can best be made in a lathe, but one can be



RESEATING BLOCK IN POSITION

made with a knife and hacksaw in a few minutes, that will give a perfectly true and highly polished surface to a valve seat.

F. L. PERKINS.

Waltham, Mass.

Broken Stud Buckles Piston Rod

Our engine, which is located in the basement some distance from the boiler room, commenced to knock, and upon arriving in the engine room, I found steam escaping around the piston rod and a knock at the head end at every revolution. I stopped the engine and examined the marks on the crosshead and guides for clearance, and on the crosshead and rod to see that the rod had not backed out of the crosshead. I found the clearance equal and the rod in place.

After further examining the engine I again started and finished the day at noon as we only run a half day on Saturday. Upon removing the flange and false head, I found the cause of the trouble. One of the studs had broken off in the piston spider and, in working out, had been caught between the piston and head. The clearance being $\frac{1}{16}$ in. and the nut on the stud $1\frac{1}{8}$ in. something had to give which happened to be the rod for it bent in two places, so that the piston was $\frac{1}{4}$ in. out of square with the rod. The follower plate had an imprint of the nut on one side and cracks running in all directions on the other side. The bridge dividing the foot of the head end also cracked through. As the cylinder was considerably out of round, I had it rebored and fitted with a new rod, junk ring, packing rings, centering bolts and a set of studs.

From the appearance of the metal at the break, it had been broken for some time, but, fitting tightly in the hole in the spider, it was prevented from falling immediately. As the stud that failed was the upper one on the side near the port it fell when the piston commenced the next stroke and was out of reach in the port when I found it.

Since being repaired the engine runs very smoothly and considerable less coal is consumed, owing to the tight piston and true cylinder.

A. C. SIDINGER.

Salem, Ohio.

Direction and Amount of Strain on Pins

It is sometimes desirable to show the direction and strain on the two pins that support a lever of the form illustrated. A mathematician can easily give a calculation intelligible to other mathematicians, but the mechanic needs a simpler solution.

Assume the weight to be 100 lb., and it is desired to determine the direction and extent of the strain on the pin *B*. The weight is trying to rotate the lever around the pin *A*, and is prevented from so doing by the pin *B*. The strain on *B* is in a direction at right angles to the line *AB* and is as much more than 100 lb. as the horizontal line *C* is longer than the line *AB*. That is, assume those two lines were the arms of a bent lever, the direction of the strain and its extent would be self-evident.

To determine the direction and extent of the strain on the pin *A*, assume that the weight is tending to rotate the lever around the pin *B*. The direction of the strain on *A* must be at right angles to the line *AB* at *D*, and the amount of strain must be as much greater than 100 lb. as the line *E* is longer than the line *AB*.

Why the strain is as the length of the line *E*, instead of the line *BF*, is because the pull of the weight is evidently straight down and at an angle with the line *BF*. If it was pulling at right angles, that is,

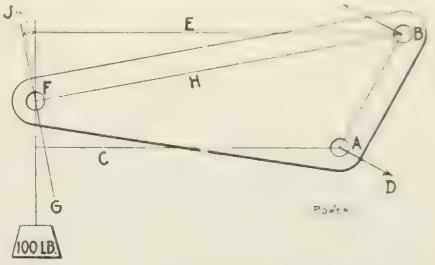


DIAGRAM TO SHOW STRAIN ON PINS

toward *G*, the lever arm would be the length of the line *H;* but pulling at an angle either less or greater than at right angles, the force exerted grows less and the decrease is shown by the difference between the length of the lever and the line at right angles to the pull, as at *J*.

JOHN E. SWEET.

Syracuse, N. Y.

Isolated Plant Management

Owners of isolated plants very often are misinformed by central-station agents, who tell them it is cheaper and more reliable to buy current than to produce it in their own plants. In some plants, where little power is generated, it might be cheaper to buy current, but how about steam heat in the winter? And do not the engineer's services count for something? Who is the first man called upon when things go wrong in a building? How about alterations in the wiring, plumbing and general repairs? In most plants it is "up to" the engineers.

I have encountered this central-station solicitation and had to show my employers that I was making good; otherwise I might now be looking for a job. I have charge of an amusement building and park combined. The capacity of the plant is about 200 kw. The current is used for about five months in the year, and the cost is below 2c. per kw.-hr. The other seven months the load is about 100 kw. The building is heated by exhaust steam only, at $1\frac{1}{2}$ lb. back pressure. The heating surface is 8000 sq.ft. and the

circulation is excellent, there being no trouble in heating the building adequately in zero weather.

The plant consists of two Corliss engines directly connected to two 100-kw. direct-current generators, two water-tube boilers, rated at 150 hp., one vertical slide-valve engine directly connected to a 3-kw. generator, one 16-ton ice machine connected to a Corliss engine, one brine pump 6 and 5¾x6 in. and three 6 and 4 by 6-in. boiler-feed and condenser pumps. I erected the plant personally, and being induced by the management to stay after it was completed, have now had charge about five years.

The manager has received numerous calls from the central-station agents, who talk cheap kilowatt-hours, low-pressure boilers, no-licensed engineers, the porters being able to heat and care for the building, etc. It seemed to him a profitable proposition, so he obtained estimates from the central-station people. The amount which was asked for just lighting our building now pays for heating, lighting and refrigerating; besides my services and the fireman's thrown in.

To win the respect and confidence of our employers we must be absolutely "on the level," save a dollar when we can, stop crying for unnecessary help, and get out of the chair and do some of the work ourselves. Should there be room for improvement which would save dollars, we should not fail to convince them of it.

For instance, our building is used for amusement purposes only. Very often as many as 5000 people congregate; therefore the amount of water used for toilet purposes is very great. Our water bill was at one time $1200 per year. I finally induced my employers to drive our own well at an expense of $350. Our water bill now is just the price of pumping. We use well water for all purposes.

GEORGE ZIMMER.

New York, N. Y.

Engineers' Examination Questions

The following questions were asked an applicant for a first-class license and are given word for word as near as they can be remembered:

1. How much babbitt would you put into a set of crankpin boxes? How would you find out how much you should put in?

2. Tell all the good and bad qualities of both open and closed feed-water heaters.

3. What is the height of a Bulkley condenser? (a) Can it be worked without a pump? (b) Is there any more danger to the cylinder heads with this type of condenser than with others?

4. Where is the weakest part of a Stirling boiler?

5. Suppose that you take charge of a 2000-hp. cross-compound engine, and the superintendent tells you after shutting down the first night that the engine is out of line and he wants it lined up that night and ready to run the next morning and you are not to strip the engine, how would you do it?

6. What kind of metal are pistons made of? What kind of metal are piston rings made of?

7. If you ordered rings and they came too large, how would you fit them so that you would not have to make more than one cut?

8. What have you got to keep the water from going over into the engine if an open heater is used?

9. If the head of a firm should ask you the necessity of an open drip between the cylinder and the open heater, how would you explain that waste of steam to him?

10. How can you balance the load between the cylinder of a cross-compound engine without using an indicator?

The following questions were asked an applicant for a second-class license:

1. Would 1-in. plate be all right to use for a high-pressure boiler? Give reasons for answer.

2. What are the usual sizes of boiler heads?

3. How many rows of rivets are there on the girth seam of a boiler?

4. Why not more?

5. A safety valve set to blow at 100 lb. blows back to 90 lb. How can you remedy it?

6. What is the latest cutoff possible on a single-eccentric Corliss engine? Why not later?

7. How do you get a later cutoff with a double-eccentric engine?

8. On a compound engine, why are two eccentrics used on the exhaust instead of one? Why would not one do for both cylinders?

9. On a duplex pump the piston on one side hits the head. How do you remedy it?

10. If you open the throttle on a single pump that makes one stroke toward the head and stops there, what is wrong?

11. Why will not a compound engine work with the governor working just the high-pressure side alone?

12. When running a condensing plant the vacuum drops suddenly to 10 in. and from that point keeps uniform. What is wrong and how remedied? How would you favor the engine until you got the vacuum back again?

13. Will a slide-valve engine cutoff at ¼ stroke?

14. How do you find the safe working pressure on a boiler?

L. W. CHADWICK.

Bridgewater, Mass.

International Standard for Flanged Fittings

Now that the main objection to changing the present standard of flanged fittings (the cost of new patterns) has been eliminated by the adoption of a new schedule by the manufacturers, it would seem fitting that the representatives of the manufacturers and buyers join forces and agree on one schedule acceptable to both.

The present condition appears to be the result of injured feeling on the part of the manufacturers, who say to the consumer, "You didn't consult us when you made your schedule so we won't use it, but will get up one of our own." As the purchaser usually is in a position to do the dictating, it hardly behooves the manufacturers to maintain too lofty an independence; still it is no more than right that they should have a voice in the schedule to be adopted.

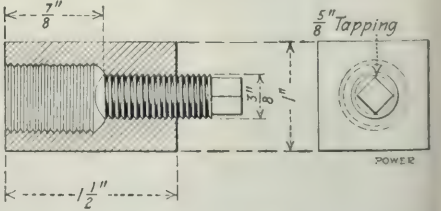
Now that there is to be a new standard, would it not be desirable if England could be included and an international standard drawn up that would be used throughout both countries? Take, for instance, the large amount of engineering work now being done in Canada. In almost all cases part of the material and machinery is purchasd in England and part in the United States. If the same standard were used much trouble would be avoided.

W. S. DURAND.

Brooklyn, N. Y.

A Handy Stud Driver

In setting in studs common practice is to put two nuts on the stud and screw them until they are well locked together. This frequently injures the thread, but it can be avoided by making a stud driver similar to the one shown. For a ⅝-in. stud a piece of 1-in. square iron is suitable. This is drilled and tapped out ¼ in. deeper than the diameter of the stud.



STUD DRIVER

A ⅜-in. hole is drilled and tapped through from the top also. This small hole allows a pin to be screwed through and comes down on top of the stud, and thus prevents injury to any threads on the stud when screwing it in.

J. A. CAMPBELL.

Providence, R. I.

Locomotives using superheated steam run 30,000 miles with air-furnace iron piston rings; 1000 miles with gray iron rings is considered good.

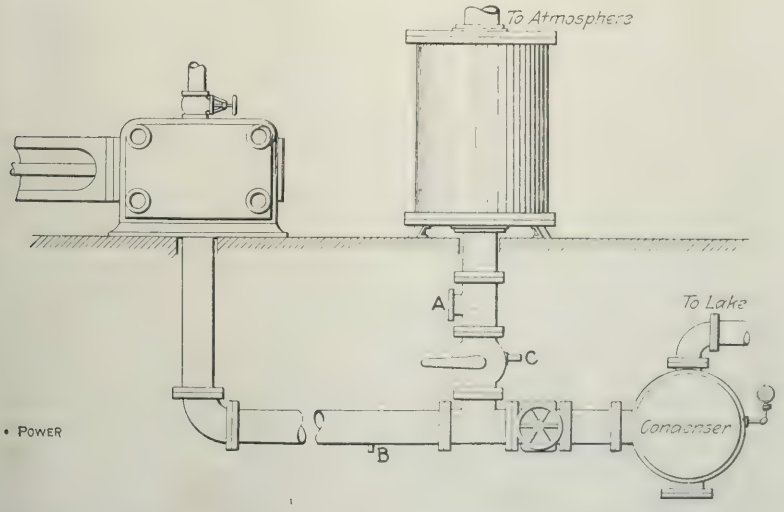
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Mr. Jacobs' Condenser

If B. A. Jacobs, who writes of condenser troubles in the Apr. 30 issue, will examine the pipe connections from the engine to the condenser and from the condenser to the lake, he will probably find numerous small air leaks, or if the piping is of cast iron, even a cracked piece of pipe. The best way to find the leaks would be to put a water pressure of from 15 or 20 lb. on the system from the engine to the foot valve on the suction pipe. If he cannot get the condenser to maintain a vacuum of at least 26 in., when the engine is fully loaded, he will have to decrease the suction lift to 18 or 20 ft. as suggested by C. E. Aldrich in the June 18 issue.

I believe that the plan of piping, shown herewith, has several advantages over Mr. Aldrich's. It would be less work to make the change and the chances for air leakage are fewer. By reducing the number of right-angle turns by four, and doing away with about 20 ft. of pipe, the vacuum would be increased, 1 or even 2 in.



PROPOSED ADDITIONAL CHANGES IN MR. JACOBS' CONDENSER LAYOUT

By using the exhaust steam from the boiler feed, condenser and air pumps in the heater, as shown by the connection at *A*, the temperature of the feed water would be increased 60 deg. or more. If for any reason the pump must be stopped and the engine run noncondensing, its exhaust would then pass through the heater, still maintaining a higher feedwater temperature.

An automatic atmospheric valve should be installed to prevent serious accident. A drip and vacuum breaker should be provided, as shown at *B*, and also a drip about 2 or 3 in. above the seat of the atmospheric valve, as shown at *C*. This would serve the double purpose of keeping the exhaust pipe from the atmospheric valve to the heater free from the condensation from the pump exhaust and also serve to form a water seal on the top of the atmospheric valve.

W. A. FIELD.

Anoka, Minn.

Crank Position on Shafts

In the June 4 issue, C. E. Aldrich writes as to the positions of the cranks of a three-cylinder plunger pump. In the construction of the engine the unusual water end (as he describes it), and the movement of the plungers is the main factor to be considered. As each set of plungers is driven directly from the crossheads it would be necessary to have the cranks placed at 120 deg., to have the plungers advanced the same distance apart to effect a continuous flow of water. Another point to be considered is the operation of the valves. With the cranks at 120 deg. the valves will operate at regular intervals, which is necessary for a smooth running pump.

If the cranks were placed at any other position than 120 deg., the plungers would be unequally spaced, thus causing an unequal and irregular flow of water and other undesirable results which would have to be overcome by additional appliances.

C. E. FLETCHER.

Hanford, Calif.

Submerged Flotation

The discussion of "Submerged Flotation" in the issues of June 4 and July 9 brings to mind an inquiry the writer often heard soon after the disaster to the "Titanic," "Why don't they raise her?" When the practical difficulties are made clear, such questions have arisen as, "What is there left to raise?", all based more or less upon the question, "How hard did she hit bottom?"

So many things are involved that an estimate can only be approximate. Starting with the vessel's weight acting against the water resistance, exact calculation is complicated by its submerged weight varying with depth. Neglecting this let us use weight as the propelling force downward, and, consider a straightforward plunge. At high velocities, when a partial vacuum would be formed behind the vessel (similar to the vacuum formed by test propellers revolving at enormous speeds and low efficiencies), although the formulas for marine resistance may not be expected to hold with entire reliability, the approximation will be of interest. Considering the cross-sectional area, or the resistance of the entire vessel, to be, say three times that of the hull below the water line, we must triple the resistance per ton of weight which is indicated by the formula.

The various formulas for the resistance of water to vessels passing through it, all depend upon the exact shape, size and proportion of hull. A common approximation of this sort is

$$R = D^{\frac{2}{3}} \times V^{2} \times C$$

where

R = Resistance;
D = Displacement;
V = Speed;
c = Constant depending upon the lines of the vessel.

Another method, which amounts to the same thing but has an actual rather than a theoretical basis, is to take the horsepower required to propel some large vessel at a given speed, reduce it to the corresponding propeller thrust (or pull in a tow line) and calculate the force necessary to drive the vessel at other speeds, from the well known laws of variation in

horsepower, or in resistance with speed. This variation in horsepower is approximately as the cube of the speed, or, the variation in resistance is as the square of the speed.

For ready reference, we will use a figure from Kent, where, under "Results of Progressive Speed Trials in Typical Vessels," we find the indicated horsepower of a steamship of 11,550 tons displacement given as 14,500. Assuming a combined engine and propeller efficiency of 65 per cent., the corresponding actual propelling horsepower becomes 9425. Dividing this by the tonnage, we have 0.81 hp. per ton, or $33{,}000 \times 0.81 = 26{,}730$ ft.-lb. per min. A knot or nautical mile being 6080 ft., 20 knots per hour is 121,600 ft. per hour, or 2026 ft. per min. Then 26,730 ft.-lb. through 2026 ft. corresponds to $26{,}730 \div 2026$, or 13.2 lb. pull per ton at 20 knots per hour.

At 40 knots per hour, since the square of 40 is four times the square of 20, the force required will be four times the force required for 20 knots per hour, or 52.8 lb. Similarly, at 80 knots the force is $52.8 \times 4 = 211.2$ lb., and at 160 knots, 844 lb. This is based upon submergence of only that part of the hull below the water line. Since we are to triple the resistance because of the entire hull and upper works being submerged, we have 3×844 or 2532 lb.

The marine ton in the air weigh 2240 lb., but when submerged steel is considered weighs around 1960 lb., or, say 1500 lb. The vessel with its contents is not all steel, but contains porous material thoroughly saturated under enormous pressure.

Starting with 20 knots per hour, and a resistance of 13.2 lb. pull, for any other speed V the resistance R is found by the formula

$$\frac{V^2}{20^2} = \frac{R}{13.2}$$

or

$$R = \frac{13.2\ V^2}{20^2} = 0.033\ V^2$$

for regular propulsion, or with three times the resistance per ton when entirely submerged,

$R = 0.099\ V^2$, or $0.1\ V^2$ (approximately)

Using our estimate of the weight of a submerged ton of vessel, $R = 1500$ lb., $1500 = 0.1\ V^2$, or $V^2 = 15{,}000$, or $V = 123$ knots per hour, approximately, the combined speed of two passenger trains meeting at 65 miles per hour, or 208 ft. per sec., $\frac{1}{10}$ the velocity of the projectile from a high-power rifle, or the theoretical velocity of a body which has fallen from a height of 700 ft.

Did the "Titanic" have this velocity when she struck the bottom? Because of the variables involved, especially the variation in specific gravity with depth, any attempt to calculate the acceleration in water, due to gravity, would involve considerable guess work. Our figures are of interest merely as indicating the velocity which may have been obtained, and which probably was not exceeded.

E. G. MEDART.

Quincy, Ill.

[The calculation is interesting, but the premises and conclusion exceedingly doubtful.—EDITOR.]

The College Graduate and Practical Experience

The scheme suggested by Julius G. Berger, under the above heading in the June 25 number, would work if it could be carried out successfully. Will Mr. Berger please name some factories where a man can get such a training and at the same time be earning a little coin. To most men who are working and who expect work in power plants, "earning as you learn" is often necessary. Most of them are not rich and cannot afford to pay a premium to a firm to teach them.

Mr. Berger mentions the General Electric Co. as a sample of industrial concern to learn with; perhaps he is right, but will it work out in practice? Has he had the practical experience or is he just giving us theory? I worked for some years with the Canadian General Electric, and also in some other large industrial concerns, and I have learned some things.

A man might get employment, but he could put in a lifetime with any one of them, and fail to get a chance to learn very much of value about the different apparatus in use, because he will not be allowed to work at what he wishes, but will have to work at what the foreman puts him on. In large industrial factories, the rule is generally to place a man at some particular machine or in some department and keep him there, the man may become an expert in that one branch, but he will get little chance to get practical instruction or experience about anything else. If he can afford to pay the company a premium and work for "nothing in the pay envelope," they may take him as a student, and if he is a pusher he can learn, but even in this case he may have to fight for his rights, for in some cases an endeavor will be made to keep him at certain things as long as possible so that the firm may make something from his work.

In one factory where I was employed, the student system was in force and little or no endeavor was made to teach them what they were paying for, with the result that two-thirds of the students' time was spent in loitering—time which was being paid for in most cases, at a good stiff figure—by the parents of the students. Even a young man who enters such a factory determined to learn, will lose interest and become indifferent, when he sees that the ones being paid to look after his education are indifferent to his success or failure.

About the only way a man could get such an education as Mr. Berger suggests would be to work with one firm in some department for a short time, then if he could not get transferred to another, he should leave and try to get what he wants with another firm and so on if necessary. Even using this scheme he might have to work for very small wages as his services would be of little or no use to these firms at the beginning of his work.

If he can get employment in a good power plant and will use his spare time in study, making a change to different plants when opportunity offers, he will ultimately become as good a man as the best and would understand human nature just as well as in the former case.

JOHN THORN.

London Junction, Ont.

All Mr. Berger says is true; and I agree with him fully,, but he looks only on one side of the question. How about the college student who has to work to pay his way through college? Can he afford to spend two years more as an apprentice in some large manufacturing establishment?

In the first place, the pay these large concerns offer apprentices is just about enough for them to live upon if careful not to indulge in pleasures of any kind, and after four years of hard work at college without any pecuniary reward, it is hard for the student to go back to it for two years more. He wants to be independent, and enjoy life. Besides, he may have had to borrow money to help pay his way through college, and he is anxious to repay it.

To my mind the right thing is for the graduate to pick out his field of work, and get a job in that field. His first job may be at the lowest rung of the ladder; but any student who has had to work his way through college will not be afraid to tackle almost any sort of a job. The next thing is to stick to his job; if he shows the right spirit he will soon be rewarded with a better position and better pay. At the end of two years he will be better off, perhaps, than his classmate who worked as an apprentice for some large manufacturing concern. The former will be holding a position with some responsibility attached to it, while the latter will yet have to make up his mind as to what branch of engineering he will follow, and go out and get a job.

A technical education only opens the various engineering fields to the student and gives him a good foundation for deeper study in any particular field. While at college a student should make up his mind to what field he will devote himself. If he has had to work his way through college he is more likely to get

much more out of his course than if he is fortunate—or, I should say, unfortunate—enough, to have everything he needs without knowing the value of a dollar, and having all his own time taken up with pleasures. This student needs at least two years of apprentice work to make him wake up and realize that earning one's own living is a serious thing.

J. E. POCHÉ.

New Orleans, La.

Failure of Boiler Tubes

The July 9 issue contains a letter on this subject by W. A. Cooil, which is a comment on Mr. Maxwell's complaint in the issue of May 28.

Mr. Cooil's solution is correct, so far as it goes, but he does not explain the real cause, which can be nothing but the scale deposit on the tube. A perfectly clean tube submerged in water or filled with water cannot possibly become overheated, no matter how intense the heat is in or around the tube. If the tube is clean the water readily absorbs the heat, and the tube temperature cannot exceed that of the water itself, but if the tube is coated with a thin film of scale or even of oil, the tube temperature may be raised far above that of the water.

At 500 deg. the tensile strength of an average boiler tube is about 66,000 lb. At 1200 deg. it drops to about 10,000 lb. Under such conditions it is not surprising that tubes sometimes fail.

When the tube becomes overheated it naturally stretches or expands. If that expansion is gradual and uniform the ordinary boiler is so constructed as to take care of it. In a water-tube boiler, when the tubes expand, the bending is always upward; in a return-tubular boiler, the bending is always downward. This is due to the location of the scale in or on the tube. In a water-tube boiler the scale usually forms on the inside and bottom of the tube. The scale does not expand or contract and holds that part of the tube in a vise-like grip. It is therefore the upper portion of the tube which expands, and as the expansion follows the lines of least resistance, the tube bends upward. For the same reason the tubes in a return-tubular boiler either bend downward or loosen at the header. In 99 cases out of 100 a loose or buckled tube is proof positive of overheating and scale.

Mr. Maxwell states that tubes sometimes bag when perfectly clean, or when the amount of scale is so small that it alone is not the direct cause of the rupture. He little appreciates how small an amount of scale causes overheating and rupture of the tube, and if he investigates he will find the only possible cause is the scale or incrustation deposited on or in the tube. Boiler tubes should last as long as the boiler shell. If clean they are subjected to no extraordinary wear or tear or no unusual strain. It is the strains which destroy the life of the tube and their principal cause is overheating due to scale deposit. This is well illustrated in the arch tubes now being used on locomotives. These tubes hold up the brick arches and, where they are not kept clean, the weight of the arches quickly cause them to sag. When the latter condition prevails, the life of the tube is only two or three months.

C. S. DAVIS.

Buffalo, N. Y.

[If the circulation is poor, a clean tube may bag with an exceedingly hot fire. The water would not be supplied to the tube fast enough to avoid the formation of steam pockets or bubbles, which would allow the tube to become hot enough to bag under pressure.—EDITOR.]

Things Commonly Wrong

The advice in your foreword, "To the College Graduate," May 28, was timely and to the point, especially for the man who expects later to design machinery. He will find valves made with a yoke and if it becomes necessary to pack them he will find that the stuffing-box gland will barely clear the box; if the valve is in a hot place he will wish that the designer had made the yoke ½ in. higher so that the gland would slip far enough back for him to get the packing in handily.

He will find pumps that have threaded stuffing-box glands and if he fills the box with packing, just even full, the gland will not start on the thread. He will find the same trouble on the stud bolt kind where, if the studs were made ½ in. longer and the threaded gland ½ in. deeper, the box could be filled even full and the gland put on without trouble.

He will also find large pumps with the top cover so close to the valve decks that if you happen to break a valve stem off in the hole, he will either have to make a special ratchet to drill it out or break joints on piping and lift the top off. He will also find 3x4-in. handholes for a 4x8-in. hand, then he will wish the designer had made a little more room by having the top a couple of inches higher and the handholes a little larger, as anyone who has worked with pumps will agree. He will find engine beds where the holes come so close to the frame that the jaw of a wrench cannot be got around the nut of the anchor bolt and have the wrench strong enough to turn and tighten the anchor bolts properly.

This same fault is found in all kinds of machinery; I suppose it is done to save iron. On a pump in my plant we had to chip away the casting to get the nuts to turn, and they are hexagon nuts at that.

He will also find valves with bonnets threaded, that have the squared part so narrow that it is almost impossible to make a monkey wrench hold on them, and recourse is had to a Stillson pipe wrench. If he does not put the new valve in the vise to remove the bonnet and replace it with graphite on the threads and shoulder, he will probably not be able to remove it at all, should he put it on the pipe line without the above precaution. I could go on and give other instances, but I think these are enough so that, if the graduate goes to designing after he gets some practical knowledge, as pointed out by you, and happens to see this article, he will give us a little more metal and a little more room where it is most needed.

ISAAC VIGGERS.

Everett, Wash.

Hoarding Knowledge Hinders Progress

In POWER, June 11, the foreword "Hoarding Knowledge Hinders Progress" seems to advise engineers to tell all they know.

No one believes in or enjoys telling or helping any, especially brother engineers, better than I. But let us look at it in a business way. Our technical schools, mechanical or electrical engineers, experts or our trade papers do not give us knowledge free. If we get into any trouble and have to ask expert help, we pay for it.

Then why should we, who have subscribed for many of their expensive books and papers, perhaps taken a course in a correspondence school, exert ourselves to benefit those who will neither buy books nor papers nor help themselves.

We all know men very different from the one pictured in the foreword with a locked safe of knowledge on his head, having instead an exceedingly large and empty cracker box and depending on others to fill it.

Many employers will advise our envesting in books, papers and schools to increase our knowledge, but when we have done so are not so quick to increase our wages. The manufacturer or the agent does not share information with his competitor, but uses it for his own interests and keeps the other fellow guessing. It is his business.

I always like and do help the other fellow and enjoy so doing, but think it is often misused. I believe in education; the more the better, and helping one another and bettering our profession, but it is a matter of honest competition with us as in other lines of business. Nor should we depend too much on the other fellow or his advice. Always help those who will and do help themselves, but if you do not care for "number one," who will?

ASA P. HYDE.

Binghamton, N. Y.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Lubricator Troubles

A standard type of lubricator on the low-pressure cylinder of a compound engine, which has its steam connection taken from the high-pressure end of the receiver, stops feeding a few minutes after the engine is started. What may be the cause and remedy?

W. T. H.

Probably the connection to the receiver is so made that steam sweeps past the lubricator connection and reduces the pressure at the connection below the receiver pressure. This could be remedied by connecting to a bent pipe placed within the receiver with its opening presented to the direction of flow. An easier and surer way of getting sufficient pressure is to lead the lubricator steam connection from the high-pressure steam pipe in place of from the present receiver connection.

Flash Points of Cylinder Oils

What are the usual working flash points for cylinder lubricating oils used for air compressors, steam engines, turbines, gas engines and oil engines?

J. E. M.

For cylinders of air compressors it is desirable to use oils having a flash point of 550 to 600 deg. F. Cylinder oil for use with superheated steam may have a flash point of over 600 deg. F. For steam cylinders with ordinary steam, about 550 deg. F. No oil is used in the steam space of a turbine. For internal-combustion engines, cylinder oil having a flash point of about 450 deg. F. is used, as the heavier oil is likely to leave a residue which will cause back-firing and other troubles.

Submergence of Air Lift in Pumping Deep Wells

For raising water from a deep well with an air lift, where the free surface of water outside of the water pipe stands 210 ft. below the level of the point of discharge, what should be the depth of submergence of the air and water pipes?

J. M. C.

It is found that the highest efficiency is obtained when the air and water pipes extend below the free surface of the water a distance equal to two-thirds of the difference of levels of the free surface of water outside the discharge pipe while lifting, and the point of discharge. The difference of levels being 210 ft., then for best efficiency the submergence should be 140 ft.

Location of Engine Belt Tightener

Should a belt tightener be used on an engine pulley or on the jackshaft receiving pulley?

A. L. J.

Whenever a belt tightener is required it can be used with best advantage to the belt if employed for holding the slack side of the belt to the face of the pulley on which there is the greater amount of slippage. This ordinarily occurs on the face of the smaller pulley and, on that account, tighteners for engine belts are usually employed to greater advantage for holding belts to receiving pulleys, though a tightener should not be on an engine pulley if it is avoidable, as it tends to throw the engine shaft out of line in a direction which is different for each adjustment of the tightener.

Water Horsepower of Pump

What is meant by the "water horsepower" of a centrifugal pump, and what would be the water horsepower and the efficiency of a centrifugal pump having a capacity of pumping 5000 gal. per min. against a total head of 140 ft.?

B. J. T.

The water horsepower would be the number of horsepower which would be developed theoretically by the same quantity of water falling per minute from the same height. The 5000 gal. per min. raised against 140 ft. head would absorb energy at the same rate that energy would be generated by the fall of the same quantity of water from the same height in the same time. Taking the weight of 1 gal. of water at 8⅓ lb., the work performed in either instance would be at the rate of $5000 \times 8\frac{1}{3} \times 140 = 5{,}833{,}333$ ft.-lb. per min. or

$$\frac{5{,}833{,}333 +}{33{,}000} = 176.76$$

water horsepower.

The efficiency would be the ratio of the water horsepower output to the brake horsepower input at the coupling or pulley which receives the power required for driving the centrifugal pump. The efficiencies of well designed centrifugal pumps of the capacity stated, range from 75 down to 70 per cent. The net or brake horsepower required therefore would be from $\frac{176.76}{0.75}$ hp. to $\frac{176.76}{0.70}$ hp., or from about 236 to 253 hp.

What is an Interpole Motor?

What is an interpole motor?

R. C. D.

One which has small poles located between the main poles, their windings being in series with the main circuit. The magnetizing effect of these poles counteracts that of the armature reaction; hence shifting of the neutral plane is prevented, and the brushes need not be shifted when change of load occurs.

Efficiency of Double Riveted Lap Joint

What would be the efficiency of a longitudinal double-riveted lap joint of a boiler made of ⅜-in. plate of 55,000 lb. tensile strength and 2¾-in. pitch of $\frac{13}{16}$-in. rivets having a shearing strength of 42,000 lb. per sq.in.?

C. P.

The cross-sectional area of the solid plate for the pitch distance, 2¾ in., would be

$$2.75 \times 0.375 = 1.03125 \textit{ sq.in.}$$

and the strength would be

$$1.03125 \times 55{,}000 = 56{,}718.75 \textit{ lb.}$$

As the distance between the rivet holes would be

$$2\tfrac{3}{4} - \tfrac{13}{16} = 1.9375 \textit{ in.}$$

the cross-sectional area of plate between rivet holes would be

$$1.9375 \times 0.375 = 0.72656 \textit{ sq.in.}$$

and the strength between rivet holes would be

$$0.72656 \times 55{,}000 = 39{,}960.8 \textit{ lb.}$$

For each pitch distance, there would be the resistance of two rivets in single shear. The diameter of rivets being $\frac{13}{16}$ in., the area of each rivet would be 0.518487 sq.in. and the strength of rivets would be $0.518487 \times 2 \times 42{,}000 = 43{,}552.+$ lb.

The strength of a joint of the dimensions given would therefore depend on the strength of metal between the rivet holes, and the efficiency would be,

$$\frac{39{,}960.8}{56{,}718.75} = 0.7045$$

of the strength of the solid plate, or 70.45 per cent.

Study Questions

This Week's Questions
Last Week's Answers

(76) A 1½-in. wrought-iron bolt with a head ¾ in. thick is under a tension of 12,000 lb. What is the stress per square inch tending to shear the head off?

(77) An eliptical manhole has a major axis of 24 in. and a minor axis of 16 in. What is the area of the opening?

(78) A man wishing to gage the height of a stack holds a 1-ft. rule vertically in front of him so that the top of the rule is in line from his eye with the top of the stack and the bottom of the rule in line with the bottom of the stack. He then finds the horizontal distance of the rule from his eye to be 18 in. and his own distance from the base of the stack 162 ft. How high is the stack?

(79) Given that steam at 15 lb. pressure occupies a volume 1670 times that of the same weight of water, how many pounds of water must be evaporated to fill a cylinder 74 in. in diameter by 7½ ft. stroke with steam at 15 lb. pressure?

(80) How much more heat is realized in burning a pound of carbon to carbon dioxide (CO_2) than in burning it to carbon monoxide (CO)?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(71) This problem illustrates the use of the barrel calorimeter.

Let x = Quality of the steam.

Neglecting radiation and the effect of the barrel in absorbing heat, the heat taken up by the water must exactly equal that given up by the steam. The heat taken up by the water is its initial weight 360 lb. multiplied by the rise in temperature (109.6 — 52.8 = 56.8 deg.) or

$$360 \times 56.8 = 20{,}448 \text{ B.t.u.}$$

The heat given up by each pound of steam condensed is equal to its quality x times the latent heat of evaporation (891.3 B.t.u. for steam at 79.7 lb. gage) plus the difference between its temperature (323.3 deg. F.) and the final temperature of the water (109.6); that is, $x \cdot 891.3 + (323.3 - 109.6) = x \cdot 891.3 + 213.7$

The total heat given up by the steam is this quantity multiplied by its weight (final weight of water minus initial weight, or 379.1 — 360 = 19.1); that is, $19.1\ (x\ 891.3 + 213.7) = 17{,}023.83\ x + 4081.67$

Equating the heat given up by the steam to that taken up by the water

$$17{,}023.83\ x + 4081.67 = 20{,}448 \text{ B.t.u.}$$

$$x = \frac{20448 - 4081.67}{17023.83} = 0.9613$$

or

= 96.13 per cent. quality.

Therefore, the percentage of moisture is

$$100 - 96.13 = 3.87 \text{ per cent.}$$

(72) From

$$\text{Area} = \frac{\pi D^2}{4} = 0.7854\ D^2$$

the area of the high-pressure cylinder is 615.75 sq.in.; of the intermediate 2123.72 sq.in.; of the high 4071.51.

Multiplying the area of each cylinder by the mean effective pressure in that cylinder and adding the three gives

$$\begin{aligned} 60 \times 615.75 &= 36{,}945 \text{ lb.} \\ 38 \times 2123.72 &= 80{,}701.36 \text{ lb.} \\ 18 \times 4071.51 &= 73{,}287.18 \text{ lb.} \\ &\overline{190{,}933.54 \text{ lb.}} \end{aligned}$$

as the total pressure.

Multiplying this by the length of stroke in feet and by twice the rotative speed and dividing by 33,000 gives

$$\frac{190{,}933.54 \times 4.5 \times 2 \times 100}{33{,}000} = 5{,}207.28 \text{ i.hp.}$$

(73) With a factor of safety of 4½ and an ultimate shearing strength of 40,000 lb. per sq.in., the safe working stress will be

$$\frac{40{,}000}{4.5} = 8888.88 \text{ lb. per sq.in.}$$

There will then be required

$$\frac{20{,}000}{8888.88} = 2.25 \text{ sq.in.}$$

of area in the rivets and as each ⅞-in. rivet has a cross-sectional area of 0.60132 sq.in.

$$\frac{2.25}{0.60132} = 3.74 \text{ rivets}$$

making four rivets the smallest allowable number to be used.

(74) All data must be reduced to the foot-pound-second system. If the body had covered 315 yd. in 179 sec., its initial velocity was

$$c = \frac{315 \text{ yd.} \times 3 \text{ ft.}}{179 \text{ sec.}} = 5.279 \text{ ft. per sec.}$$

Its final velocity when covering 10,800 in. in 1/50 hr. was

$$v = \frac{\frac{10{,}800 \text{ in.}}{12 \text{ in.}}}{\frac{1}{50} \text{ hr.} \times 60 \text{ min.} \times 60 \text{ sec.}} = 12.5 \text{ ft. per sec.}$$

The time the force acted was

$$\tfrac{2}{3} \text{ min.} \times 60 \text{ sec.} = 40 \text{ sec.}$$

Then from the formula

$$v = c + pt$$

where

v = Final velocity;
c = Initial velocity;
p = Acceleration;
t = Time;

Then

$$12.5 = 5.279 + 40\ p;$$
$$p = 0.1805.$$

The total force acting was

$$P = Mp$$

where

$$M = \text{Mass} = \frac{G}{g}$$

where

G = Weight in lb.;
g = Acceleration of gravity = 32.2;
G = 2½ tons = 5000 lb.

then

$$M = \frac{5000}{32.2} = 155.28 \text{ lb.}$$

and

$$P = 155.28 \times 0.1805 = 28.028 \text{ lb.}$$

(75) The volume of the room would be

$$40 \times 80 \times 12 = 38{,}400 \text{ cu.ft.}$$

To change the air five times an hour would require this quantity of air every 12 min. The fan discharge should be

$$\frac{38{,}400}{12} = 3200 \text{ cu.ft. per minute.}$$

The South's Hydro-Electric Progress

The remarkable progress being made by the South in hydro-electric development is an example to the rest of the country. Plans are being carried forward rapidly for the formation of the Little River Power Co., which will supply energy throughout the northern part of Alabama.

Three water-power developments, having a total rated output of 150,000 hp., are contemplated, the first being on the Little River, near Fort Payne, la., which will have a rating of 50,000 hp. under a head of 590 ft. Energy will be transmitted from the plant at 100,000 volts to Anniston, Gadsden, Huntsville and neighboring localities.

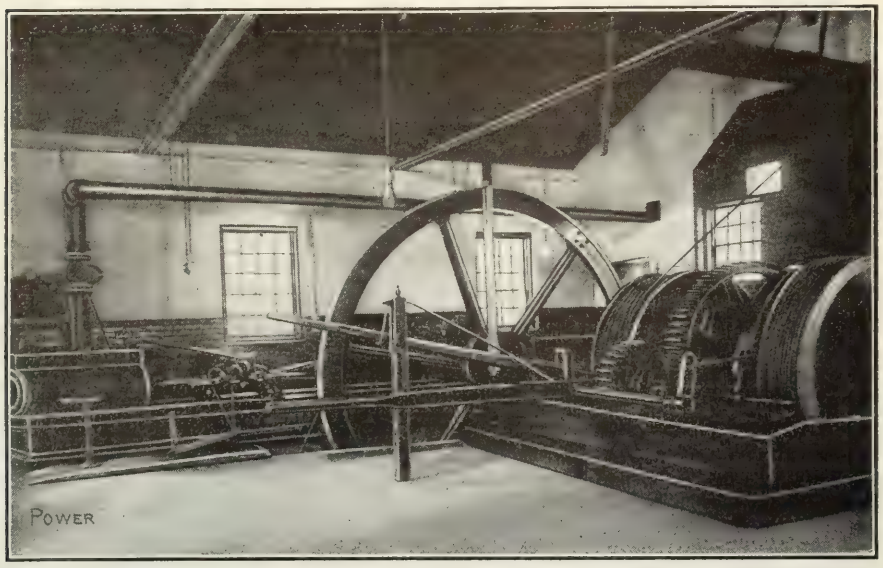
The Dix River Power Co. will build a large hydro-electric plant on the Dix River, near its confluence with the Kentucky River. It expects to spend $3,000,000 on its 24,000-hp. plant, and in constructing transmission lines to Louisville and Lexington, Ky., and Cincinnati, Ohio, as well as to smaller towns nearby, such as Stanford, Harrodsburg, Lancaster, Nicholasville and Danville. A feature will be a 200-ft. concrete dam across the river near Danville.

The formation of the Tennessee Ry., Light & Power Co., with a capital and bonds of $120,000,000, points to a huge hydro-electric development to serve large portions of Tennessee and Georgia. The Tennessee Power Co. will own the 20,000-hp. hydro-electric plant on the Ocoee River at Parksville, Tenn., now in operation; the transmission lines now in use from this plant, by way of Cleveland, to Chattanooga; the transmission line from Cleveland to Knoxville, Tenn., which is nearly completed; the transmission line from Cleveland southward to the Georgia state line; the Great Falls power site and reservoir at the junction of the Caney Fork and Collins rivers, near Rock Island, Tenn., capable of developing 80,000 hp., and two additional sites on the Ocoee River, one capable of developing 16,000 and the other 30,000 hp.

Old Time Hoisting Engine

The new design of hoisting engines so commonly seen about collieries has, in most instances, supplanted the earlier make of engine which was used for hoisting purposes, but a few of the old-timers may still be seen at work on shafts where miners and supplies are handled. These engines were of the single type and were known as "hook" engines, because the eccentric reach rods hooked onto a rocker-arm much as the carrier arm of a Corliss engine hooks onto the wristpin of the wristplate.

The eccentric rods were each operated in and out of gear by a foot lever which was hung on an iron pivot stand at the center and connected with a hooked rod at the inner end; the other end was made with a flat footplate. As the rotation of the engine was reversed, one of the eccentric rods was lifted from the pin of the rocker-arm and secured in that position by a lever which was held in an off or unlocked position when the eccentric rod was not hooked onto the rocker-arm pin.



OLD HOISTING ENGINE OF THE HOOK VALVE-GEAR TYPE

The rocker-arms hung from a rocker shaft which ran across the crosshead guides and operated the valve stem as the rocker-arm was actuated by the eccentric to which it was connected. If the valve had lead at one end of the cylinder with one eccentric hooked up, the lead would be charged to the other end of the cylinder when the other eccentric was hooked up. The rocker-arm was fitted with a starting-bar. In starting the engine the engineer handled the throttle-valve lever with one hand, the starting-bar with the other and operated the foot lever which controlled the eccentric rod with the foot.

The illustration shows a box-frame engine of this type which is still in operation at the Barnum colliery of the Pennsylvania Coal Co., Pittston, Penn. It has but one 12x24-in. cylinder. No date appeared on it anywhere, but it is supposed to have been built about 1849. The main bearings are adjusted by wedges and keys, the latter being held in place by setscrews.

The engine shaft is fitted with a spur gear which meshes with a large gear on the drum shaft. The engine shaft also supports a 14-ft. flywheel with a 3¾-in. face. The two drums are 7 ft. in diameter and have flat faces.

Naturally the band brakes are of the old type. There are two long wooden brake levers, one of which is loosely secured to the engine shaft by an iron band which passes around the shaft and is bolted to the end of the brake lever at the top and bottom. The brake band is secured to the drum frame on one side and to the top of the brake lever on the top side near the end by an eye-bolt connection. Shoving the lever down tightens the brake band. The second brake lever operates in the same manner, with the exception that it is attached to a cross shaft which tightens the end brake band by a series of link connections. The levers are held up by a hook when the brake bands are loose and by a saw-tooth piece of iron when tight.

The Expert and the Engineer

The engineer sat in his easy chair, puffing an occasional ring from his favorite pipe toward the dirty ceiling as he leaned back and reflected on the possible chances of the various aspirants for local political offices. After about half an hour he made the rounds of the engine room, looked at the steam gage, filled the oil cups and cylinder lubricator, felt of one or two bearings and resumed his chair. These were his duties, varied only by an occasional gasket to be replaced, a pump to pack or a glass to put in; they had been the same for several years.

Employed by a manufacturing concern, his place was in an obscure corner of the building and his existence was called to the attention of the management at times by fuel bills to be paid, also little items for repairs for his department and in addition to some salesmen who asked to see him; moreover, the boiler inspector when he called often referred to him by name. He made out no reports, and kept no log of any description, and as long as he was let alone he was perfectly satisfied except on payday. Being master of his own profession, with nothing more to be attained or learned, he often found time to consider and imagine improvements which might be made in other departments, but always resented any suggestions relative to his own.

Recent increases in machinery had made necessary the installation of a new and larger engine which when purchased was of another make than he had recommended. This engine had never run as it should, in his estimation, and he had broken his silence to the management to report that fact, greatly to their surprise. As a word from him was so unusual, they thought something certainly must be wrong and notified the engine company that the call of one of their experts would be appreciated.

The expert, upon his arrival, was escorted to the engine room by the superintendent and introduced to the engineer, with whom he did not "take" at all. First, because he wore "dude" clothes and even had on kid gloves; second, his manner and speech were what the engineer thought belonged at pink teas or missionary-society meetings. He was very agreeable, polite and precise in his language and the engineer made no secret of his disapproval.

The expert looked over the place; told the engineer he would be on hand early the next morning and bid a friendly good day.

Upon the engineer's arrival the next morning he was surprised to find the expert already on the scene, but somewhat changed in appearance; he now wore overalls and did not seem to feel out of place in them. As the engineer opened the feeds on the oil cups, the expert opened the cylinder drips. This act, the engineer believed, furnished the proper opportunity for letting the expert know his position; he accordingly turned fiercely upon the latter with that purpose in mind, but met a clear, penetrating gaze from a pair of gray eyes which rather changed his mind and caused him to defer the outburst. At the same time the expert remarked that time was valuable when one of the engines was to be out of service, and consequently part of the plant.

After the other engines had been started, the expert proceeded to the business in hand with a despatch which took the engineer by surprise. He asked questions which were to the point and asked them in a way which the engineer could not evade. He asked by name and size for the tools wanted, asked for them all before he began and did not omit any which were needed afterward. As he picked one up he told the engineer what to do with another, and the engineer was mortified to note how much quicker the expert could handle them than he could. It apparently came natural to do what the former told him to do.

After the valve-chest cover had been taken off, the expert inquired who had tried to set the valves since the erecting man left, and whether the engineer owned and used an indicator. The engineer replied that he had changed the valves and did not have an indicator; furthermore, he had never felt the need of one. The expert said he would set the valves again and then use his indicator, which had been brought along. This he did and after the valves had been adjusted the expert smilingly told the engineer that there was nothing wrong with the engine and that the makers would agree to any tests which the purchasers wished, but that he had a few friendly words for the engineer personally if the latter cared to listen.

By this time the engineer had changed his opinion of the expert and had come to regard him as an "engineer," agreeing to listen attentively.

"I will be forced to stay here until 4 p.m. and would like to call your attention to a few things," said the expert. "First, there are a few things which you can have and use which will call to the attention of the people you work for, the fact that you are alive: An indicator, for instance; get one and learn to use it. Then some good reading matter—not printed on pink paper or dealing with ball scores, but something that will help you in your work. Then keep a record of what your plant is doing. I notice the smoke coming from your stack is about as black as the border of the unused surface around the fires. Every blowoff valve on the boilers is wasting coal into the sewer and the traps are doing the same."

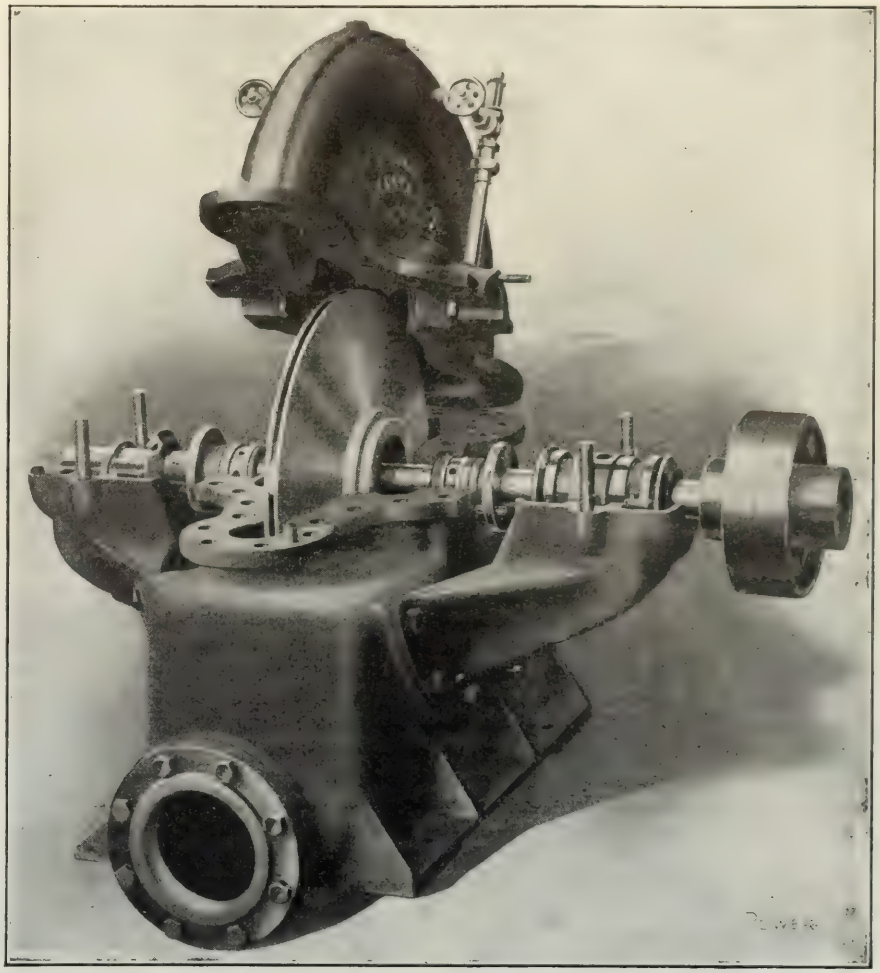
After going over the steam lines into the factory which furnished steam for purposes other than power, and pointing out leaks, the expert got the engineer and his firemen together and described briefly the "A B C" of combustion; then spent the balance of his time instructing the engineer how the proper procedure in getting up cost comparisons and how to figure them.

After he left, the engineer reported that he was thoroughly satisfied with the new engine. Two months later he told his wife of his raise in salary.

Goulds Centrifugal Pump

The illustration herewith shows the general construction of a new single-stage, double-suction centrifugal pump recently developed by the Goulds Manufacturing Co., Seneca Falls, N. Y. This pump is adapted for general water supply, hot-water circulating in heating systems and all services where the total head does not exceed 150 ft.

As shown, the casing of this pump is divided horizontally to give access to the interior of the pump. The two parts are bolted together and doweled to give exact alignment.



Design of the Goulds Centrifugal Pump

The machined impeller is of the inclosed, double-suction type, the water being taken in axially from both sides through openings surrounding the shaft. The two streams are directly opposite, thus giving hydraulic balance and preventing end thrust. The impeller and other parts of the pump are such that serious overloading of the driving mechanism is impossible. The shaft stuffing-boxes are provided with water-sealing rings.

When used for pumping liquids which attack iron, the pump is furnished with a bronze impeller and all parts of the shaft coming in contact with the liquid are covered with a bronze sleeve.

Cylinder Lagging and Economy

By James H. Cannell

There is a great loss of heat from engine cylinders due to radiation and conduction. This loss must have a considerable effect upon the economy of an engine, and in order to find the improvement in the steam consumption of a triple-expansion high-speed engine, due to a more complete lagging of the cylinders, the following tests were taken.

The engine to be tested was coupled direct to a 330-kw. dynamo and this was the means of loading the engine. Three series of tests were taken, the first with the standard engine, the second with the special cylinder lagging, and the third was a repetition of the first series in order to find if there had been any improvement in the steam consumption due to the prolonged running. If during the tests there had been an improvement in the economy of the engine, due to the piston and valve rings becoming tighter, or to decreased friction of the engine, then this improvement would have to be taken from that due to the more complete lagging.

The arrangement of cylinders and lagging during the first series of tests is

shown in Fig. 1. The engine is of the usual design of this type, the throttle valve being at the side of the high-pressure cylinder and the receivers and exhaust pipes at the back of the cylinders.

The cylinders are quite independent, the only connection between them being the receiver pipes; each cylinder is separately lagged with composition and sheet steel. Owing to the necessity of keeping the centers of the cylinders as close as practicable, the amount of lagging on the sides of the cylinders is reduced to a minimum; the cylinder and valve covers are also lagged, as shown, in the manner usual with this kind of engine.

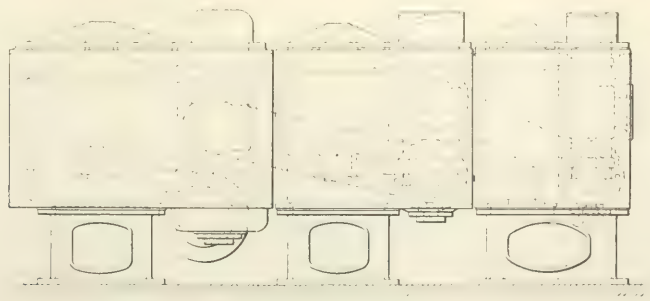


FIG. 1. CYLINDER LAGGING USED DURING FIRST AND THIRD SERIES OF TESTS

Fig. 3 shows the result of this series of tests, the curves of the total steam per hour, and the steam per indicated horsepower per hour being given.

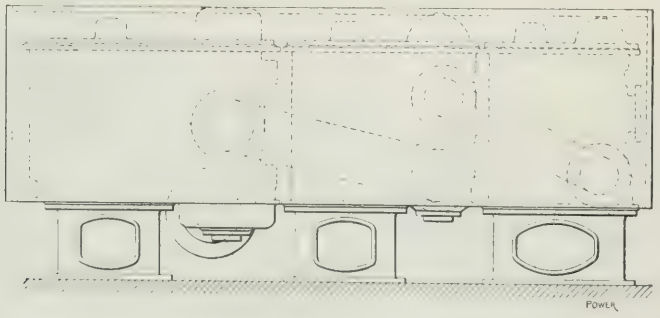


FIG. 2. LAGGING USED DURING SECOND SERIES

By referring to Fig. 1, it will be seen that the flanges of the cylinder and valve covers and part of the top of the cylinder are not lagged, and that there is no air space between the composition and the sheet steel lagging. It is well known that a body of stagnant air is a most efficient lagging, and these tests were taken for the purpose of finding the effect of this and the covered cylinder tops on the economy of the engine.

The lagging during the second series of tests is shown in Fig. 2, in which the sheet steel is continuous around the cylinders and the cylinder tops are covered. In this case there is an adequate air space between the composition on the cylinders and the sheet steel, the steel being fixed on angle irons supported from the cylinder flanges.

The results of these tests are also shown in Fig. 3, in similar manner to the result of the first series of tests, so that the gain due to the more complete lagging is readily seen.

After the second series of tests had been completed the cylinders were again lagged as during the first series, and further tests were taken to find whether any improvement in the economy of the engine, due to the prolonged running, had taken place.

The result of the third series of tests was found to be in close agreement with the result of the first series, proving that the difference between the first and second series was due entirely to the more complete lagging.

By referring to Fig. 3 it will be seen that the percentage of gain due to the improved lagging is practically constant at all loads and that this gain is approximately 6 per cent. From these tests it can be realized how important the efficient lagging of a cylinder is toward attaining the maximum economy from an engine, and this can usually be attained with very little expense.

During the second series the temperature of the air inclosed in the space above the cylinder was taken, and it varied during the various loads from 180 to 200 deg. F. The space between the bottom of the cylinder and the lagging was not inclosed during the tests, but in the subsequent adoption of this type of lagging this was done as it insured a more air-tight lagging.

The curves in Fig. 3 are interesting from another point of view, as they show the usual steam consumption obtained with the best designs of this type of engine. Throughout the tests the speed of the engine was kept constant at 350 r.p.m., the steam pressure at the stop valve was 150 lb. and the vacuum in the exhaust pipe was 25 in.

To obtain reliable results, the steam at the stop valve was superheated from

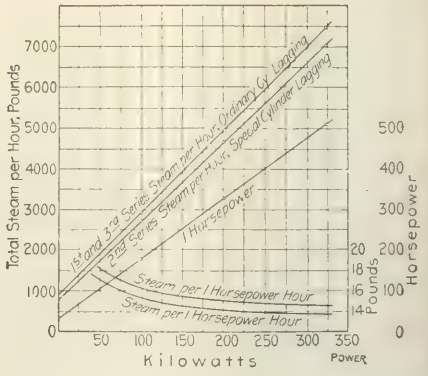


FIG. 3. SHOWING RESULTS OF TESTS

5 to 10 deg. F. and the steam consumption obtained during the tests was corrected to that which would have been obtained with dry steam, by adding the correction found by previous tests, so that the curves are obtained from tests under the same conditions and can be readily compared.

Third International Rubber Conference

During the week beginning Sept. 24, the third International Rubber Conference will be held at the Grand Central Palace, New York City, in connection with the Rubber Exposition. These meetings will be attended by many foreign and American manufacturers of mechanical rubber goods.

The chief topics for discussion will be specifications for steam packings; specifications for steam hose and other hose used by engineers in industrial plants; specifications for textile fabrics used in the manufacture of mechanical rubber goods; efficiency methods in the rubber industry.

Engineers and chemists interested in these topics from the consumer's standpoint are invited to attend the sessions of the conference. Persons desiring to take part in the discussions or present an original paper on a related topic should write to the honorary secretary, Dr. Frederic Dannerth, Third International Rubber Conference, New Grand Central Palace, New York City.

Interesting Diagrams

By M. R. Jonas

In hunting the trouble with one of the engines in our plant, the other day, I obtained some indicator diagrams which may interest some readers.

The engine is a vertical, cross-compound condensing engine having two steam and two exhaust valves on each cylinder. The steam valves are double-slide, one sliding over the other, and the exhaust valves are simple slide. It

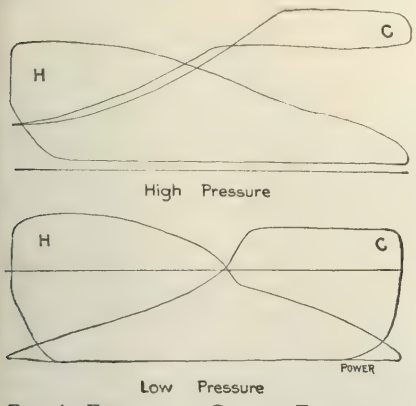


Fig. 1. Effect of Closed Exhaust Valve

is direct-connected to a direct-current generator.

The first indication of trouble was when the engine refused to pull the load, and would only develop half load, and the receiver pressure, normally 20 lb., dropped to 6 lb. per square inch. Upon indicating, the diagrams shown in Fig. 1 were obtained.

Then we saw at once that the trouble was with the crank-end exhaust valve. Upon taking off the exhaust steam-chest covering we found that the nuts holding the valve stem to the yoke had been

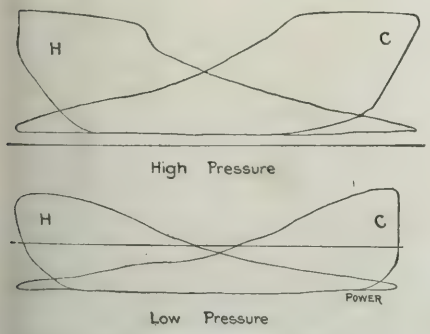


Fig. 2. Showing Normal Operation

stripped off, the stem moving in and out while the valve remained shut. Upon remedying this, the engine pulled full load all right; and the indicator diagrams shown in Fig. 2 were obtained.

The engine exhausts into a condenser together with several rolling-mill engines. A few days each week the mill is down and on these days the engine is run noncondensing. While running under these conditions the indicator diagrams shown in Fig. 3 were taken. It will be seen that the expansion in the low-pressure cylinder goes below atmospheric pressure. Then, when the exhaust valve opens, the pressure increases to a little above atmospheric and stays there during the exhaust. This accounts for the

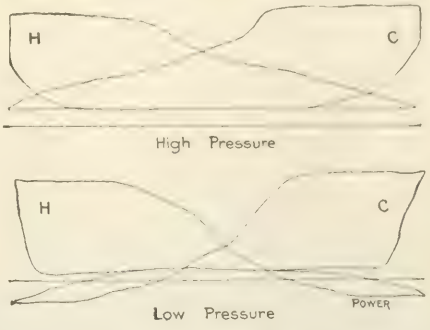
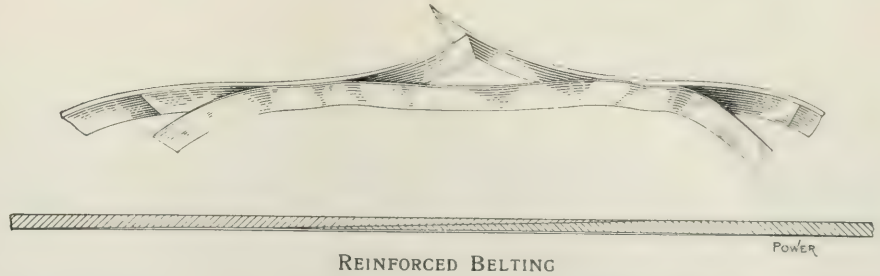


Fig. 3. Showing Conditions When Engine Runs Noncondensing

negative loop and gives a fairly good idea of what a condensing engine will do when run noncondensing.

Reinforced Belting

The accompanying sketches show the method used by the Bronner & Barnewall Co., 111 Hudson St., New York City,



Reinforced Belting

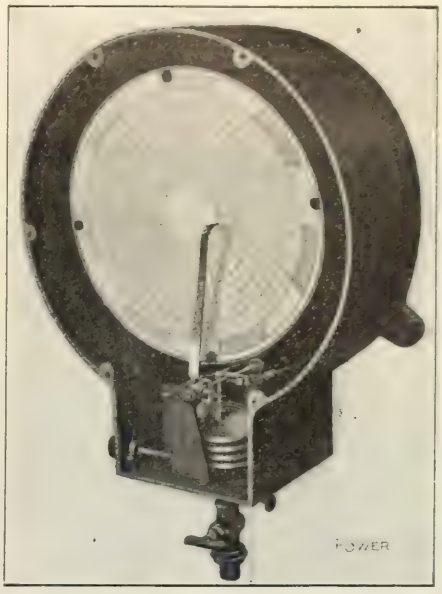
for making the joints of belting. The sections to be joined are skived to a thin edge and reinforced by a strip inserted beneath the joint upon the under side of the belt, as shown. The object is to eliminate uneven stretching, should strips of uneven quality come together in the joint.

Precision Dead Beat Recorder

This instrument, illustrated with the cover plate removed, is manufactured by the Precision Instrument Co., 49 Larned St., Detroit, Mich. The principle and construction of the lever arrangement by which any inequalities of movements of the diaphragm are compensated are shown, making the reading from zero to full stroke.

Variations in pressure or vacuum are transmitted by the diaphragm through a vertical, adjustable plunger attached to a horizontal shaft by an adjustable link. This shaft is supported by a bracket and is normal to the plane of the shaft disk and converts the reciprocating motion of the diaphragm into a rotary motion. When the shaft revolves, the pen-arm moves parallel to the chart, thus recording the pressure.

A screw adjustment projects through the side of the case, by which the operator may bring the pen to zero of the chart. The gage is mounted in either



Dead Beat Recorder

round or standard cases and is suitable for steam, gas, water or other liquids or air. The clock movement used to drive a shaft is an imported jeweled chronometer having cut gears throughout.

Fluorescence in Oils

Practically everyone is familiar with the beautiful purple fluorescence which certain oils show. This phenomenon is much stronger under illumination by the inclosed arc than that of any other light, and is usually a property of mineral oils only. It has consequently been proposed that as a test for the purity of animal or vegetable oils, they be exposed to the light of an inclosed arc, when if pure, they do not fluoresce, while if adulterated with mineral oil, they will.

This test is usually conclusive; it is said that some of the lighter distillation products of petroleum can be bleached by prolonged exposure to sunlight until the fluorescence is no longer perceptible, while certain resin oils will fluoresce slightly under the inclosed arc. However, the test suggested will at least serve as a rough one for any who are suspicious of any alleged "pure" animal or vegetable oil.

Care of Governors

By C. R. McGahey

All governors require constant care if they are to perform their functions successfully. Improper oiling, dust, etc., will make them stick, and uneven tension of the springs, especially if they are rusted between the leaves, will cause

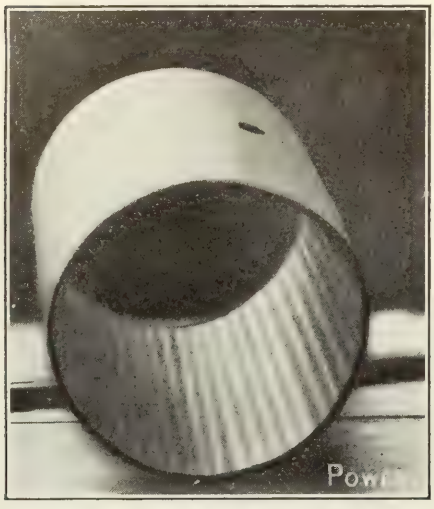


Fig. 1. Fluted Governor Bushing

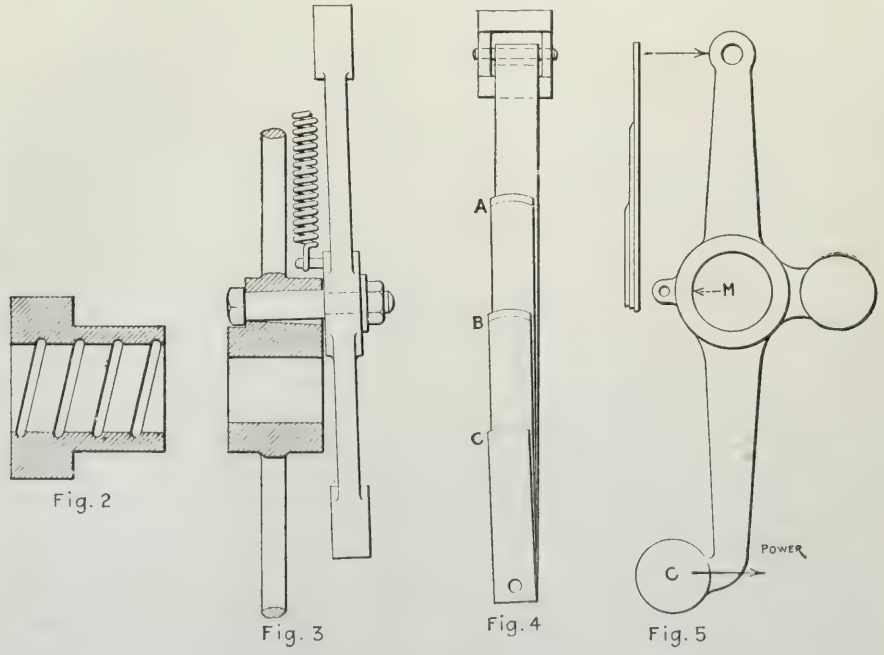
very poor regulation and racing of the engine.

Fig. 1 shows a bushing taken from a standard shaft governor employing a roller bearing. The bushings, though made of hardened steel, were considerably worn and fluted, making poor regulation, as the governor arm hung in its movements and the roller could not operate freely.

In many shaft governors, the design of the main pivot is a troublesome factor to both designer and operator. Inadequate lubrication and dirt accumulation are other sources of trouble which can be eliminated if the bushings are grooved as in Fig. 2. The grooves should be made deep to allow a free movement for the grease or other lubricant.

Fig. 3 shows another case of wear common to governors of the single-bar type. The spring tension and weight wear the pivot at one side, thus causing the governor to jump, and produce poor regulation and much noise. The writer was called upon recently to correct this fault in a governor. After striking the springs a few blows with a hammer and setting them in line, as at *A*, *B*, *C*, Fig. 4, the engine showed excellent regulation.

When a governor is undergoing repair the springs should be taken apart and a liberal application of oil and graphite put between the leaves. The point *M* of the governor arm, Fig. 5, shows where trouble usually begins. Friction and irregularity of action will greatly increase as the wear at *M* becomes greater.



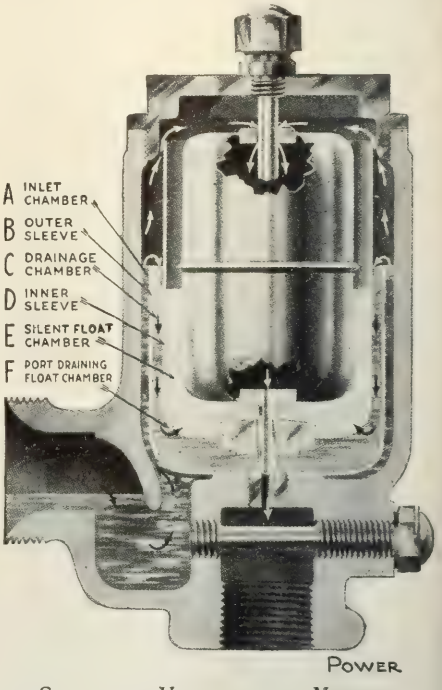
Showing the Governor Parts

Improved Monash Noiseless Radifier

In this device the water of condensation enters the inlet chamber over the outer sleeve into the drainage chamber, through the ports in the bottom of the inner sleeve, and enters the silent float chamber. A separate passage provides for the free and constant escape of air into the return through the center of the float.

When the water in the silent float chamber is at the line of flotation the float rises and the valve opens, emptying the drainage chamber. With the outrush of water the water level in the silent float chamber is gradually lowered by the escape of a little water through the ports, when the float sinks, closing the valve.

The float and drainage chamber are separate, each performing its individual function. Drainage from the drainage chamber into the return is direct and



Sectional View of the Monash Radifier Valve

rapid. The body of water in the silent float chamber is always still, with a slight and gradual lowering of the line of flotation, thereby insuring silent action of the float. The separate drainage chamber insures a direct discharge of oil, grease and dirt into the return pipe, instead of into the silent float chamber.

This device is for use on vacuum-heating systems. It is manufactured by the Monash-Younkers Co., 1407 West Jackson Boulevard, Chicago, Ill.

Kerr Economy Steam Turbine

A new type of steam turbine with vanes and blades producing parallel flow in the direction of the turbine shaft is being manufactured by the Kerr Turbine Co., Wellsville, N. Y. It is known as the Economy steam turbine and is built for condensing and noncondensing service in sizes of from 2 to 750 hp. capacity and in sizes up to 450 hp. for exhaust steam. A decrease of steam consumption of from 10 to 15 per cent. in noncondensing service and in some instances as great as 25 per cent. on condensing turbines is claimed by the manufacturers for this turbine over their former type.

The curves, Fig. 1, indicate performances under identical conditions of an

old machine rated at 250 b.hp. and an Economy turbine having the same number and diameter of wheels. With both turbines taking steam at 100 lb. at the throttle at rated load, the water rate under 26 in. vacuum was 24 lb., for the old as compared to 18.7 lb. for the new construction. Similarly, under 28 in. vacuum the water rate is cut from 23 lb. to 17.2 lb. The relative size of the two types for equal duty is illustrated by the projections which are drawn to the same scale.

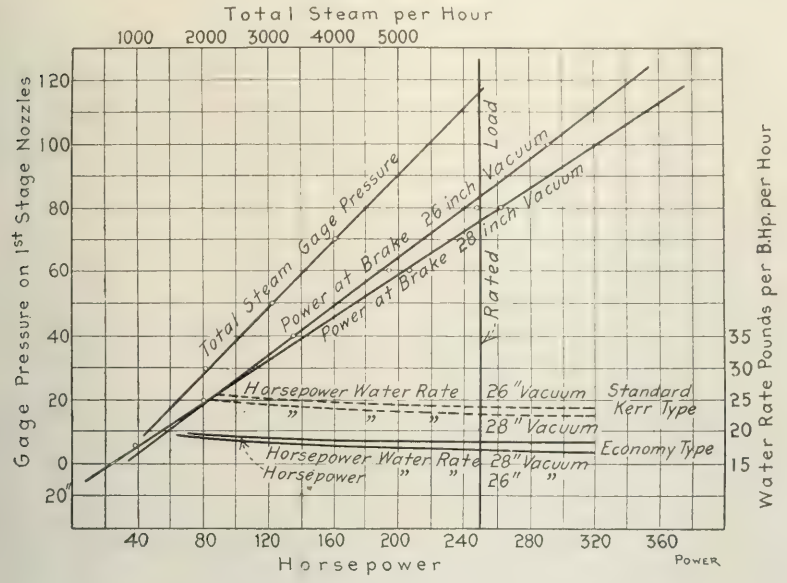

Total Steam per Hour
1000 2000 3000 4000 5000
Gage Pressure on 1st Stage Nozzles
120 100 80 60 40 20 0 20"
Water Rate Pounds per B.Hp. per Hour
35 30 25 20 15
Total Steam Gage Pressure
Power at Brake 26 inch Vacuum
Power at Brake 28 inch Vacuum
Rated Load
Horsepower Water Rate 26" Vacuum Standard Kerr Type
" " " 28" Vacuum
Horsepower Water Rate 28" Vacuum Economy Type
Horsepower " " 26" "
40 80 120 160 200 240 280 320 360
Horsepower
POWER

FIG. 1. COMPARATIVE CURVES OF OLD TURBINE

The departure from former design has been accompanied by an increase in strength of all the working parts. The jet velocity is kept low by expanding the steam in a succession of stages, usually from two to ten as may be desirable for different conditions of pressure and vacuum, and for desired steam-consumption rates at varying speeds.

Each set of nozzles discharges against a separate bucket wheel, steam entering the bucket at the side of the wheel and making the usual reversal. One set of nozzles and one wheel constitute a stage. Steam issues from the nozzle of the first stage, impinges upon the first-stage buckets and leaves within the first-stage casing a pressure to be expanded through the second-stage nozzles and imparted to the second-stage buckets and so on through the machine. Thus the total pressure drop is divided and the energy abstracted from the steam at each stage is comparatively small, the nozzles of the stages being so proportioned in number and size that the steam velocity is equal at each set.

The cylinder of the turbine is divided into separate stages by separate circular diaphragm castings between the steam- and exhaust-end castings. Each diaphragm casting contains one set of nozzles, provides for one bucket wheel and is arched at the center to withstand the steam pressure to the best advantage. These diaphragm castings are accurately centered with each other and with the two end castings by turned and bored tongue and grooved joints; the weight of the turbine is carried by feet on the end castings. The bearing cases also are centered by machined faces on the end castings and fall into alignment the same as the diaphragms.

Steam tightness at the joints and accuracy of alignment of the cylindrical parts of the shell are effected by annular machined tongues and grooves, the latter being packed with fibrous packing laid in graphite and oil. The sections of the turbine are bound together by continuous stay-bolts passing through drilled holes in the flange of one end casting and screwing into the flange of the other. Heat radiation is checked by filling in between the stay-bolts, and surrounding the cylinder and packing the turbine ends with lagging, held by an annular jacket and end plates of russia iron.

Leakage of steam about the shaft at the steam and exhaust end and between stages is prevented by floating bronze bushings in contact with ground metal seats and held to place on the shaft by the difference in steam pressures in the stages.

Fig. 2 shows, in addition to those bushings at the steam and exhaust ends of the turbine, packing glands for soft ring packing. Between the packing glands and the metal glands is a chamber which receives the leakage from the metal glands. A soft packing is used to confine the leakage within these chambers and prevent its escape along the shaft. The leakage through the metal gland on the steam end discharges through a pipe connection to one of the low-pressure stages of the turbine.

Leakage of air into the vacuum, where the turbine is run condensing, is prevented by supplying the chamber at the exhaust end with sealing steam.

The nozzles formed by walls within the diaphragms and thin metal vanes, which are die-pressed into shape and cast into the diaphragms, are adjacent to the bucket wheels with axial clearance between them and the buckets. The nozzles are of a form that has given maximum efficiency with the new type of bucket. The vanes are of a metal not subject to corrosion or erosion under service conditions.

The rotor, Fig. 3, is made up of bucket wheels, machined from flange steel of 60,000 lb. per square inch tensile strength, mounted upon the steel shaft. Each bucket wheel is bolted on a three-piece split steel or iron hub fitted to the shaft and kept from endwise or rotary movement by pin keys. A split mitered ring is forced by a locknut into the bore of the disk and against a turned taper, rendering the wheel on the hub quickly removable from the shaft when necessary. The buckets are drop-forged steel with a hard oxidized surface that are not subject to erosion or rust.

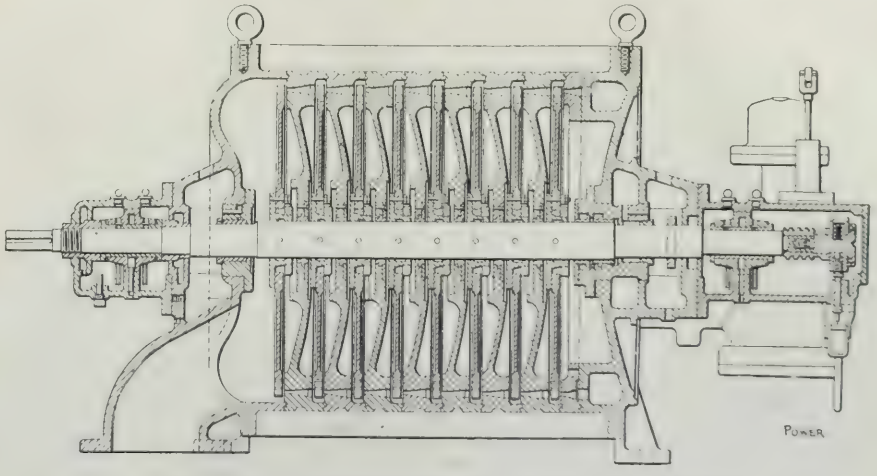


FIG. 2. SECTION THROUGH EIGHT-STAGE TURBINE

Steam is admitted to the turbine through

a steam chest having ports to the first-stage nozzles; a double-poppet valve is operated by the governor, the construction of which for average service consists

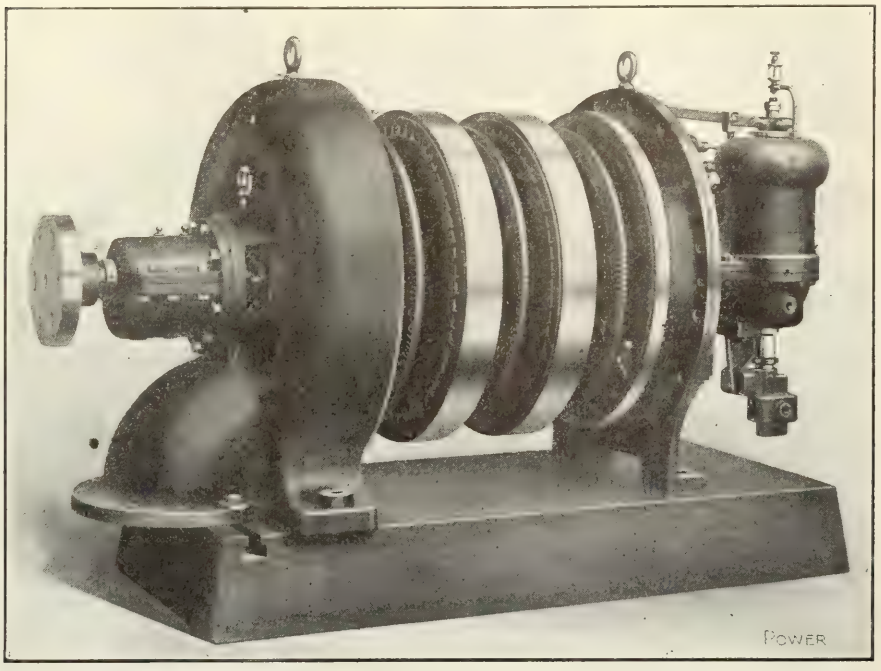


FIG. 3. INTERNAL VIEW OF THE ECONOMY TURBINE

of semiannular weights mounted directly upon the turbine shaft and acting through lever connections to throttle the steam valve. On the larger units, especially those for driving generators where close regulation is desired, the governor consists of spherical weights driven through a spiral gear on the turbine shaft and acting upon the valve stem through a relay pilot valve. An emergency governor is also employed to close a valve in the steam line when the turbine overspeeds, thus keeping the steam valve shut until reopened by hand.

The main bearings are ring-oiling, self-aligning and split in halves for removal from the bearing case by removing the bearing-case cap.

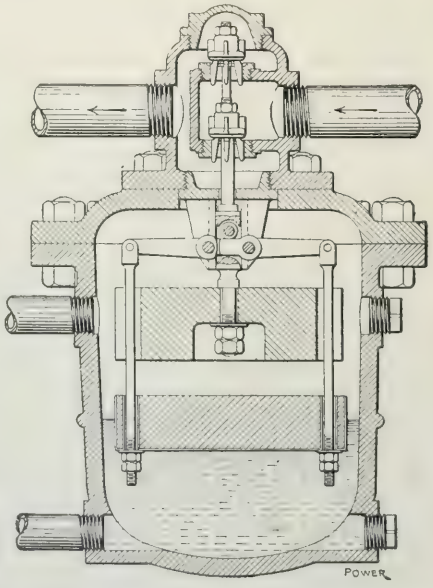
On high-speed machines, oil is forced into the bearings at about 3 lb. pressure by a rotary pump on the end of the governor spindle from the reservoir that supplies the governor.

A thrust bearing and locating collar are provided in connection with the exhaust-end bearing, which insure correct position of the bucket wheels with reference to the nozzles.

Fogarty-Thoens Boiler Feed Water Regulator

The feed-water regulator illustrated consists of a vessel connected with the boiler above and below the water level and contains a balanced valve through which the feed water enters the boiler. The valve is opened or closed, more or

in the boiler, by a weight which is counterbalanced when half submerged, thus acting the same as a float. The regulator operates by gravity only, and eliminates all unreliable parts. It is positive in action and has no parts to stick, or get out of order.



SECTIONAL VIEW OF THE FOGARTY-THOENS BOILER FEED-WATER REGULATOR

This feed-water regulator is manufactured by James H. Fogarty, 126 Liberty St., New York City.

Correction—On page 193, upper part of first column, in the Aug. 6 issue, the delivery velocity of the air into the room should be 600 rather than 6000 ft. per min. The additional cipher was a typographical error.

Pump Valve Reseating Machine

The accompanying cut, Fig. 1, illustrates the improved Dexter pump-valve reseating machine attached to a pump ready for

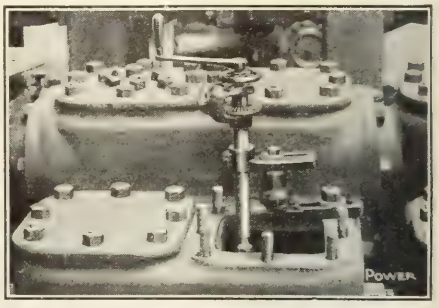


FIG. 1. MACHINE ATTACHED TO WATER END OF PUMP

reseating the lower valve seat. The seats in the upper valve plate are refaced in a similar manner. The machine is attached to the water end of the pump by

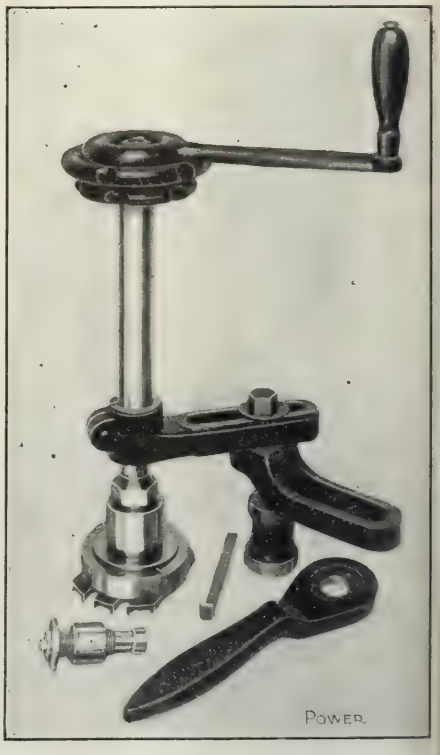


FIG. 2. MACHINE ASSEMBLED

bolting the yoke to any stud on the valve chamber. A few turns of the handle, while turning the hand feed wheel slow-

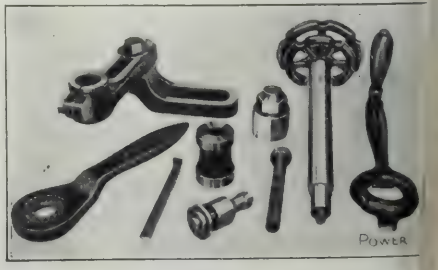


FIG. 3. PARTS OF THE MACHINE

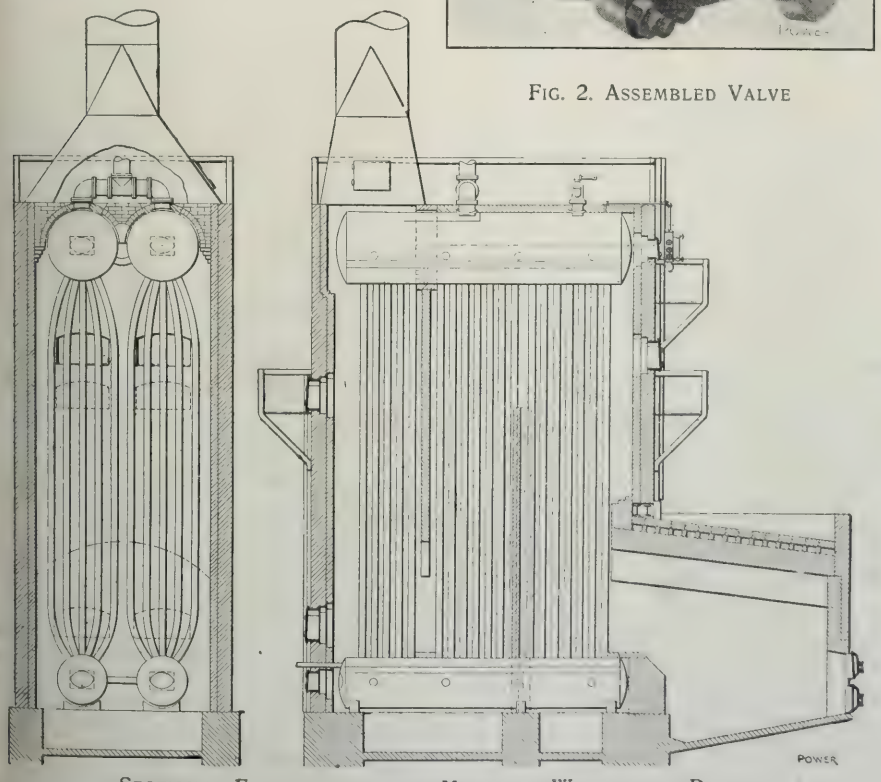
ly, refaces the valve seats without removing them from the pump.

Fig. 2 shows a machine for valve seats 2 to 4 in. in diameter. A view of the several parts is shown in Fig. 3. This device is manufactured by the Leavitt Machine Co., Orange, Mass.

Morrison Water Tube Boiler

This boiler is being manufactured in the United States by the Sharon Boiler Works, Sharon, Penn. It comprises two bottom and two top drums connected by tubes so separated that the entire arrangement is conducive to uniform and thorough combustion. Any tube in the boiler can be removed without injuring any other tube and also without displacing baffles or brickwork. This is accomplished by staggering the two intermediate rows of tubes in each bank with the center row, at the same time arranging the two outer rows in a series of short and wide spaces. The wide spaces in one outer row alternate with those in the opposite outer row. The boiler is built in single-, double- or triple-pass types and in single, double or multiple sections, end or side fired, depending on the location of the grate at the end or side of the drums.

The illustration shows an end and a side elevation of the boiler and dutch-oven furnace, and also the arrangement of the baffle walls. All surfaces are cylindrical in form and there are no bolted joints or gaskets to blow out. The longitudinal seams have butt joints with inside and outside straps. All parts are easy of access for cleaning out scale or soot.



SECTIONAL ELEVATIONS OF THE MORRISON WATER-TUBE BOILER

Forged Steel Valve

A new design of forged-steel valve, manufactured by the Patterson-Allen Engineering Co., 2 Rector St., New York City, is illustrated in section in Fig 1.

The body of the valve is made in three pieces from forged-steel boiler plate, bolted together, the center piece containing the valve seat. The valve stem yoke is fitted to the end section of the valve body, as shown in Fig. 2. The disk and seat are made from monel, the stem of nickel steel.

The forged-steel body is very strong and eliminates possibility of one side being thinner than the other. Each valve is tested to 1500 lb. per square inch hydrostatic pressure.

FIG. 1. SECTIONS OF VALVE

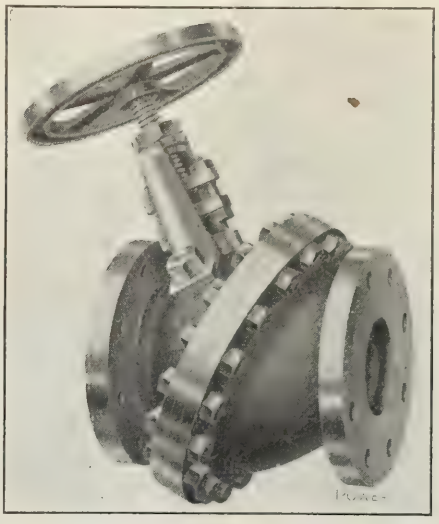
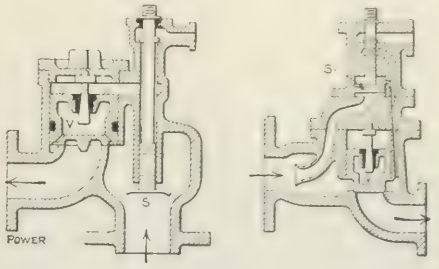


FIG. 2. ASSEMBLED VALVE

Differential Safety Valve

The accompanying engravings, reproduced from our German namesake, Kraft, illustrate safety valves in which the steam operates upon a piston or valve having differential areas. The lifting of this valve removes the boiler pressure from the top of the main valve, which is thus forced suddenly open to its full extent and so remains until the pressure is released sufficiently to allow the pilot or primary valve to seat. The seating of this valve shuts off the escape of steam from the back of the main valve, which, under the balanced pressure thus established, is seated by its own weight. In the engravings, the differential piston, or valve, is indicated by *s*, the main valve by *v*.



DIFFERENTIAL SAFETY VALVE

"Power Means Population"

When Augusta, Ga., needs an added facility in a business way, it immediately makes its desires known. And now it is more power. The Augusta *Chronicle* says:

"Augusta's crying need is more power, for power means population. For many years the water power of the canal has been exhausted. For many years there has been a demand for more power than the power company in the city could furnish.

"It is proposed to develop additional power above Augusta and bring it to the city. Ten thousand more horsepower is promised shortly; 24,000 hp. is declared available at this point.

"Here is what horsepower means. It takes 12 men to equal one horsepower; 10,000 more horsepower means the working energy of 12,000 men. If these men worked in eight-hour shifts, it would require the services of 36,000 men every 24 hours to produce the working energy of 10,000 more horsepower in Augusta."

NEW PUBLICATION

TWELVE PRINCIPLES OF EFFICIENCY. By Harrington Emerson. Published by "The Engineering Magazine," New York, 1912. Size, 5½x7 in.; 416 pp.; cloth. Price, $2.

This book is an enlargement of the author's previous work "Efficiency," and while it treats of the philosophy of efficiency, it is more specific on the fundamental principles upon which a practical efficiency system must be based. Read in conjunction with the former volume, it should prove of much assistance to those seeking a general conception of efficiency systems and also serve as a guide to efficiency engineers.

The first and second chapters are devoted to types of organizations which are wasteful because their progress is hampered by traditions and custom. As an excellent example of what may be accomplished by the execution of plans devised by a keen perception, the achievements of the German field marshal Von Moltke are cited.

Ideals, the first of the 12 efficiency principles, are considered in Chapter III, Mr. Emerson contending that the individual and aggregate excellence of an industrial organization depends upon its ideals.

Chapter IV is devoted chiefly to the lack of common sense exhibited by Americans in industrial and commercial operations, especially in the matter of imports and exports, and their mania for over-equipment. It is pointed out that the imports consist mainly of articles of luxury and delicate machinery, while the exports are of products which by their great volume tend to rapidly deplete the country's natural resources. This chapter is interesting and alive with valuable thought.

Competent counsel is the subject of Chapter V and deals with possibilities for increased economy by securing the services of a competent efficiency engineer, and Chapter VI treats of the admirable results obtainable in a business organization where a high order of discipline prevails.

Chapter VII, "The Fair Deal," contains among other subjects a common-sense treatment of the momentous question of wages, the number of hours in the working day and the highly injurious effects of overtime work.

Chapters VIII and IX, on records and dispatching, are also interesting. The remainder of the work is more specific in its treatment of the various efficiency principles.

A Small Engine Wreck

The Chambersburg Ice & Cold Storage Co., of Chambersburg, Penn., had an accident, Sunday, May 12, at 12:15 a.m. No one was killed or injured, but about $300 damage resulted.

The damaged piece of machinery was a Ridgeway high-speed slide-valve engine built by the Ridgeway Dynamo & Engine Co., of Ridgeway, Penn. It was rated at 28 hp. when running at 375 r.p.m., and directly connected to a Crocker-Wheeler 14-kw. direct-current generator and belted to a line shaft to which the following were attached: A George S. Comstock 4x4x4-in. power air compressor, a Goulds 2x4-in. triplex power pump for boiler-feed purposes, a Goulds 7x8-in. triplex power pump for condensing purposes and a countershaft to which two screw blade agitators were attached.

At the time of the accident this engine was running at 350 r.p.m. The accident was caused by the piston rod unscrewing from the crosshead and this broke the crosshead, crank bearing, crankshaft and buckled the connecting-rod, besides snapping the oiling pipes on one side of the oiling system and the gage on the reservoir and snapping the piston rod off at the piston head.

At the time of the accident only the tankman and engineer were on duty, who constituted the night crew and both were in the boiler room eating dinner. The Goulds 7x8-in. triplex pump stood not more than 2½ ft. from this engine and, although broken, parts were lying all around it, it was not even scratched.

When the engine was wrecked the lights went out. After procuring a torch, the men closed the throttle and exhaust valves and hooked the line shaft up on another small engine and ran all but the air compressor. The entire lighting system besides one of the elevators was supplied by the generator, but within 12 hr. after the accident purchased current was being used for the elevator and the lighting system switch was thrown on the "central service" one minute after the accident. No other damage than this was done and very fortunately no one was hurt for the engineer was at the engine only about five minutes before the accident happened putting oil in the reservoir.

The Bureau of Mines Recent Bulletins

The Bureau of Mines has copies of these publications for free distribution, one copy to one person. Order by number and title, addressing the Director of the Bureau of Mines, Washington, D. C.:

Bulletin 18. "The Transmission of Heat into Steam Boilers," by Henry Kreisinger and W. T. Ray, 1912, 180 pages.

Bulletin 40. "The Smokeless Combustion of Coal in Boiler Furnaces," with a chapter on central heating plants, by D. T. Randall and H. W. Weeks; 188 pages. A reprint of the United State Geological Survey Bulletin 373, revised by Henry Kreisinger.

Electric Power for Panama Canal

Electric power will be used almost exclusively in operating the Panama Canal. The hydro-electric station will be adjacent to the north wall of the Gatun spillway, and the installation will have a capacity of 6000 kw.

The average hydraulic head throughout the year will be about 75 ft. The maximum quantity of water diverted for hydro-electric development will be approximately 7 per cent. of the minimum water supply, and will be the excess which is not required for lockages, evaporation and leakage.

Locomotive Firemen's Capacity

At the recent Atlantic City convention of the American Railway Master Mechanics' Association, the firing capacity of a locomotive fireman was discussed. This capacity depends so much on the individual characteristics of the man and the peculiarities of the engine, the train and the road, that it is difficult to get any average figure, but it was brought out that the ordinary fireman working on a six- to eight-hour run generally does not fire more than 3000 lb. of coal per hour. It was also stated that on the passenger engines from 4500 to 5000 lb. of coal per hour might reasonably be expected.

An example of exceptionally high endurance and strength was noted by C. D. Young, of the Pennsylvania R.R., who stated that last summer in some capacity tests between Ft. Wayne and Valparaiso, Ind., a distance of 105 miles, a specially recommended fireman handled fuel for three hours at speeds greater than 60 miles per hour at an average of 8400 lb. of coal per hour.

This work was done with a No. 5 shovel. Mr. Young believes that had the man used a larger shovel he could have exceeded these figures. He also stated that in the locomotive test plant there were men who could fire as high as 9700 lb. of coal per hour.

St. Louis Steam Engineers Club

Organized less than two years ago, the St. Louis Steam Engineers Club now has an accredited membership of 300. Its homelike and comfortable quarters are at 810 Olive St., to which all visiting engineers and their friends are most cordially invited when in the city.

The board of directors meets every Monday evening, and on the first Wednesday evening of each month there is a general business meeting, followed by a concert provided by the entertainment committee.

During the winter months, one evening monthly is devoted to entertaining the ladies. An annual ball is given each winter and a picnic during the summer. Many delegates and visitors at the recent convention in St. Louis of the Universal Craftsmen, through the courtesy of E. N. Hunt, a well known member, were hospitably entertained at the club during their stay in St. Louis.

The officers are President Charles T. Parkinson, Secretary Price Simms, and the following board of directors: A. H. Herron, M. G. Skinner, H. C. Tulley, H. F. Rosenow, L. M. Sperry, O. P. Thompson, Price Simms, R. L. Brannon, F. C. Laufketter.

OBITUARY

F. J. Hunt, an engineer at the Milwaukee County poor farm at Wauwatosa, Wis., was electrocuted in the engine room of that institution on the night of Aug. 12. It is believed that he was fixing some wires when the accident occurred. Mr. Hunt was 36 years old and a member of Wisconsin No. 1 of the National Association of Stationary Engineers.

SOCIETY NOTES

For its joint meeting with the Verein deutscher Ingenieure, Leipzig, June 23 to 25, 1913, the American Society of Mechanical Engineers has appointed through its Council a committee consisting of Col. E. D. Meier, chairman, John W. Lieb, Jr., vice-chairman, Dr. W. F. M. Goss, Charles Whiting Baker, Major William H. Wiley and Dr. Alexander C. Humphries, president, and Calvin W. Rice, secretary, *ex-officio*. A very active interest has been manifested in the meeting and it is hoped that a sufficient number will signify their intention to attend to warrant the chartering of a special steamer. Besides Leipzig, where the joint meeting takes place, many other places in Germany have expressed their desire to give the American engineers an opportunity to see the advances made in the profession. Places it is tentatively proposed to visit are Hamburg, to visit the greatest shipyards in Germany; Munich, with a reception by the Prince of Bavaria; Berlin, visiting the industrial plants, and by special arrangement the castle of the Kaiser at Pottsdam; Essen, where there will be a reception by Krupp von Bohlen, Stuttgart, Düsseldorf, Cologne, Frankfort, Mannheim and Heidelberg. In Leipzig it is also expected that there will be, during the visit of the society, a patriotic gathering in commemoration of the Battle of Leipzig, and it is hoped that the party will be received by the Kaiser.

The second annual meeting of the Institute of Operating Engineers will be held on Friday and Saturday, Sept. 6 and 7, at the Engineering Societies Building, 29 West Thirty-ninth St., New York City. The committee on constitution and by-laws, which has been working for a year, will present its findings, and the changes and additions will be submitted for discussion.

The Sixth Congress of the International Association for Testing Materials will be held at the Engineering Societies Building, New York City, Sept. 2 to 7. About 170 foreign and about 55 American papers will be considered in the proceedings of the various sections. The entertainment program includes an informal reception, Monday evening, under the joint auspices of the American Society for Testing Materials and the American Institute of Electrical Engineers, the American Society of Mechanical Engineers and the American Institute of Mining Engineers; on Tuesday evening a reception by the city officials in the Public Library Building; on Wednesday afternoon and evening an excursion to West Point; on Thursday a reception by the American Society of Civil Engineers at its home on West Fifty-seventh St.; on Friday a harbor excursion, and on Saturday receptions by the Museum of Natural History and the Metropolitan Museum of Art.

The August meeting of the American Society of Engineer Draftsmen, held in the Engineering Societies Building on Aug. 15, was well attended.

W. T. Walters, a Chicago member, read a paper on "Safety Devices—Their Application and Design" which was followed by some graphic instances of the need of safety devices. Prof. Charles W. Weick, of Columbia University, gave an interesting lecture on "Practical Perspective" which aroused much interest and drew many subsequent questions. A question as to which is the better way to place figures on a drawing, perpendicular to the bottom of the drawing or perpendicular to the dimension line, created considerable discussion.

The American Museum of Safety, adjoining the meeting room, was thrown open for the evening to members and visitors to inspect the varied collection.

The sundry civil appropriation bill passed by the House of Representatives on June 21, and by the Senate July 24, contains an item appropriating $15,000 for the use of the Bureau of Standards in determining standard units of refrigeration, the physical constants of the materials used in the refrigerating industries and the thermal conductivity of materials. The securing of this appropriation is due to the efforts of the American Association of Refrigeration.

PERSONALS

Dr. William Cawthorne Unwin, F. R. S., president of the Institution of Civil Engineers of Great Britain, honorary member of the Institution of Mechanical Engineers of Great Britain and of the American Society of Mechanical Engineers, formerly professor of engineering of the Royal School of Naval Architecture and Marine Design and a well known authority upon hydraulic and machine design, will visit America in September as delegate of the Institution of Civil Engineers to the sixth congress of the International Association for Testing Materials.

Prof. John E. Sweet will attain his eightieth birthday this year, and it is planned to tender him a reception about the time of the annual meeting of the American Society of Mechanical Engineers of which he was one of the founders and is an honorary member and past president.

Herbert E. Stone is the New England agent for the Pittsburgh Piping & Equipment Co., contractors and engineers, with an office in the Equitable Building, Boston, Mass. Mr. Stone has for many years sold power accessories in New England, and was previously connected with the Chapman Valve Co. and the Elliott Manufacturing Co. He is a past national president of the National Association of Stationary Engineers, treasurer of the Commercial Engineers, and a member of the Engineers Blue Room Club.

Clemens Herschel, a consulting engineer, of New York City, has been appointed assistant to Lieut.-Col. W. C. Langfitt, Corps of Engineers, U. S. A., in connection with the hydro-electric development to be undertaken by the United States Government at Great Falls on the Potomac River above Washington, D. C.

E. P. Gould, for several years in the sales department of the Lunkenheimer Co., has acquired an interest in and will act as sales manager of the Steam Appliance Co., Milwaukee, Wis., manufacturers of steam and oil separators and steam traps. He resigned from the Lunkenheimer Co. on Aug. 1.

Frank H. Evans, for seven years New England manager for the Strong, Carlisle & Hammond Co., Cleveland, Ohio, has resigned and will take a much needed rest.

A business increase of 83 per cent. in the year in the revenues of the municipal electric plant at Riverside, Calif., is claimed by that city's government. This gain was principally made in increased industrial business.

A mail-tube line, 7½ ft. in diameter, is to be constructed in London by the British Post Office. Each vehicle will be operated by an individual electric motor.

Moments with the Ad. Editor

An old and valued correspondent of Power is Jas. E. Noble of London Junction, Ont., Canada.

Mr. Noble is an example of the kind of peaches the ad. editor started out to raise several years ago when this page was started.

This page was and is devoted to subscribers, not advertisers. Its whole purpose is to create for the selling section the interest that it deserves.

For, advertising generally is one of the most interesting, yes, even *romantic*, things in the world.

Forty years ago it was next to nothing. Now a billion a year goes into it in this country alone. Back of that startling progress are some of the best brains in the world, some of the most strenuous work and much of typical American energy and "get-there."

"Goods advertised in Power are made by reliable concerns," argued the ad-editor, "the subscriber who buys 'em is sure of buying satisfaction, advertising is now out of the 'bunk' class—and on a truthful basis—why not tell these things to those subscribers who, maybe, are still indifferent to the value of Power's Selling Section?"

Now read Mr. Noble's letter and you'll see an excellent instance of how it developed.

In the past I have told you about my experience with Power advertisers but a good thing will stand a second installment.

At one time I seldom read ads, but when Power started printing "Moments with the Ad Editor," I at first read the page from mere curiosity which rapidly developed into interest, not only in the page, but in the ads themselves. The third stage was the climax. I purchased some of the goods advertised and recommended the purchase of others with very satisfactory results to the buyers and probably also to the sellers.

I have recommended these goods to engineers (outside of my employer's plant) in numerous cases.

A short time ago I removed a number of pages from the Selling Section of Power, with ads of the makers of large power pumps, and sent them to a gentleman who was in want of just such information. I believe he sent to the advertisers in question for their literature and no doubt a sale will result in the near future.

I have used goods from most of the dealers mentioned in the July 30th letter and from several others who carry their ads in Power. In *every* case the goods were satisfactory and fully up to the standard claimed for them.

C. G. Scott of Wales, Wis., is another enthusiastic rooter for the advertising pages and if you doubt that just read his letter.

I read your page 'Moments with the Ad. Editor' the first time I looked the July 30th. issue through, and the next time I picked up Power I read it again.

I read the last paragraph several times before I realized that I had gone into a brown study and I found the main question in that study that I was trying to answer was, not what I have bought but what I have *not* bought through Power's ads.

Why, man, when I come to think, and your talk on page 174 set me to thinking, I find that I am using Power as a power plant appliance cyclopedia!

I've bought pump valves of Jenkins Bros., gaskets of McCord, lubricators from Greene Tweed, packing from Garlock, ventilators of Burt, steam pumps from The American and look which way I will in my plant I find something that I have either bought directly through advertisements in Power or through knowledge gained from advertisements read.

I read my Power from cover to cover (outside to outside) and I presume I spend as much if not more time over the advertising matter than I do over the reading matter. More than many times have arguments been settled through reference to advertisements in Power.

Let me say to conclude with Mr. Ad. Editor, if Power ever fires you, they might just as well fire me also for I will quit. As much as I think of Power, it would not be worth the money if they took away the advertising.

* * *

Any way you look at it, men, you'll always come back to the starting point, viz:

It pays to read advertisements and buy advertised goods.

POWER

Vol. 36 NEW YORK, SEPTEMBER 3, 1912 No. 10

In Convention Assembled

THE National Association of Stationary Engineers will convene in Kansas City and the International Union of Steam Engineers at St. Paul on September 10. These are the two largest organizations of operating engineers in America. Their avowed objects are highly commendable and their possibilities tremendous. Opportunity was never better for advancing the interests of the operating engineer.

On every side is evidence of central-station encroachments on the isolated-plant field. Now is the time for the organizations to launch a vigorous campaign of education for both the engineer and the plant owner. If it were effectively developed, it would be the biggest single thing that could be accomplished.

It is growing more and more difficult to become a thoroughly good engineer because of the increasing complications of plants and the consequent need for greater theoretical knowledge and because the plant owner is continually demanding better service and efficiency.

Developing an adequate educational course for such organizations is not easy, due to the diverse needs of the membership.

First, make the course systematic. Then, arrange it so that each individual's mental equipment, experience, opportunities and aims may be taken into account in modifying the line of study to suit his requirements. The mentally alert and ambitious young man, whose youth allows his spending several years acquiring a thorough fundamental grounding, needs a more elaborate course than the older man. Moral encouragement should be used to the limit of its possibilities. Many a man has given up in despair, not appreciating that he was making progress or not realizing that by perseverance and consistent systematic effort he would win out.

A successful educational campaign can be developed, but it is no easy task. It requires the efforts of men not only versed in the sciences to be taught, but those having a knowledge of human nature as well.

Apparently, then, the educational committee should be selected with especial care and be empowered to develop and put into effect a broad, comprehensive plan of action from which all who can may *Learn More to Earn More!*

Now turn to the "Moments with the Ad Editor," at the end of the text pages. It deserves the careful consideration of every man in the organizations.

If you agree with the doctrines set forth, dig in, tooth and nail, for their accomplishment. It's new thought, and we believe you will be glad you read it.

Impulse and Reaction Turbines

By F. R. Low

While the difference between an impulse and a reaction turbine is very apparent in the elementary machine, it is not as easily seen in the complex turbines as now made. The article points out the essential difference, with a simple explanation of the application of the velocity diagram to each.

What is the difference between an impulse and a reaction turbine?

This difference is easily seen when the elementary machines like the single-rotor DeLaval, Fig. 1, and the Hero or Avery turbine, Fig. 2, are considered, but is not so apparent in the multi-stage form which the commercial turbine has taken.

Compare the Rateau and Parsons types for example, the first an impulse, the second a reaction turbine. Each consists of alternate rings of stationary and movable blades, Figs. 3 and 4. In each the steam is expanded in the stationary blade passages, enters the passages between the moving blades, and leaves these blades with its velocity absorbed, or greatly reduced, having given up to the revolving member the energy lost by the reduction in its velocity. By further expansion in the next row of stationary blade passages, the velocity is again generated, to be again absorbed in the next row of rotating blades.

Where, then, is the difference, and wherein does it consist that the action is in one case that of impulse and in the other of reaction? Perhaps the difference can be best shown by tracing the action in each case.

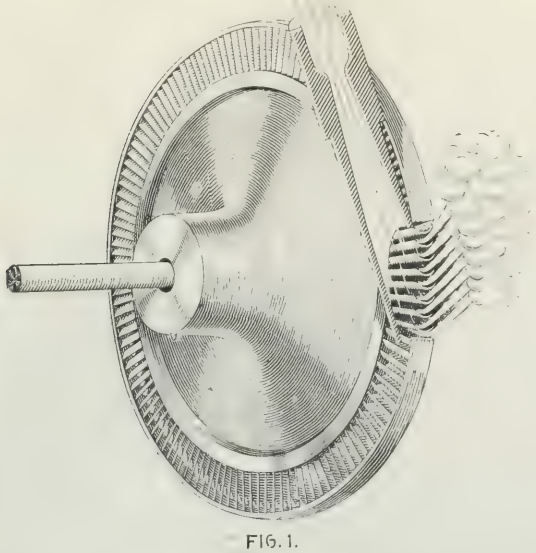


FIG. 1.

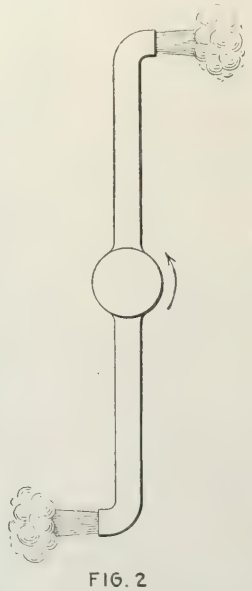
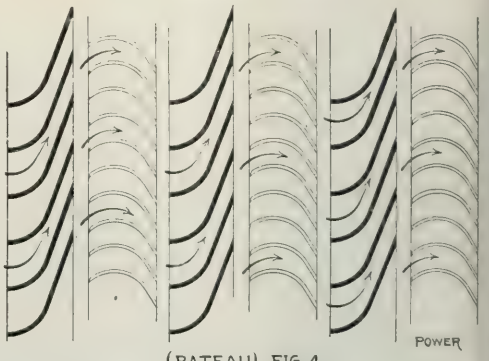


FIG. 2

(PARSONS) FIG. 3.



(RATEAU) FIG. 4.

In Fig. 5, steam is expanded in the nozzle so that the jet issues with a velocity of 1000 ft. per second. To get out of this jet the energy which has been expended in setting it in motion it must be brought to rest relatively to the nozzle. If the blade is moving with a velocity equal to that of the steam, the latter will move along with it and lose none of its velocity. If the blade is stationary, the jet will simply be turned around upon it and, friction neglected, pass off with the same velocity at which it entered the blade, but in the reverse direction. But if the blade is moving with half the velocity of the jet the jet will approach and impinge upon the blade with a velocity of 1000 — 500 = 500 ft. per second, and will flow backward from the discharge edge of the blade with the same velocity, *relatively to the blade*. But, since the blade is going forward at the rate of 500 ft. per second, and the steam is flowing backward at the same velocity relatively to it, the velocity of the steam, relatively to the nozzle, will be zero. It will have been brought to rest, and the energy due to its velocity transferred to the blade.

If the line AC, Fig. 5, represents the velocity of the jet as it leaves the nozzle and the line BC the velocity of the bucket, then the difference, AB, will represent the relative velocity. On the discharge side the jet comes off with the same velocity, $A'B'$, backward, and the difference, or algebraic sum of its velocity, $A'B'$, in this direction, and its forward velocity, $B'C'$, in the blade, is zero.

In the turbine it is difficult to completely reverse the direction, and the nozzle, or guide blade passages, usually lead in at an angle. Suppose the velocity and direction of the steam as it leaves the nozzle to be represented by AC, Fig. 6. Its velocity and direction relatively to the blade, will be represented by AB where, as before, BC represents the direction and velocity of the blade. The blade is made of such a shape that the steam will enter smoothly at this angle, and if the blade is symmetrical, the jet will issue at the same angle relatively to the blade, and (still neglecting friction) with the same relative velocity, $A'B'$. Laying out the blade velocity $B'C'$, it is seen that the only velocity left is that proportional to $A'C'$ in the direction of the axis of the turbine. By varying the speed of the jet, that of the blade, and the angles of entry and exit, these component velocities may be varied all sorts of ways in magnitude and direction.

A turbine constructed in this way with no expansion of the steam in the moving blade passages, acting simply by the impulse of the steam upon those blades, by the effort which the steam exerts upon them when its direction of flow is deflected, is an impulse turbine.

When one wants to board a moving car he runs until he is moving as fast as the car. His velocity relatively to the car is then zero and he can board without shock, just as though both he and the car were standing still.

In the impulse turbine the steam is expanded in the stationary nozzles, or blade passages, until it has a velocity more than twice that of the blades. In the reaction turbine of the Parsons type it is expanded in the stationary blade passages (for the abstract case) until it has a velocity equal to that of the moving blades, just so that it can get aboard

comfortably. In the passages of the moving blades, however, and here is the essential difference, it is expanded until it has a velocity in the blade equal to the forward velocity of the blade itself.

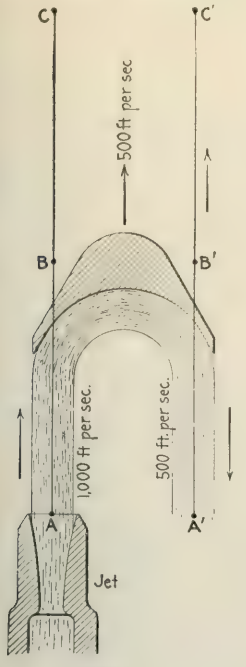


FIG. 5

In Fig. 7 a car is paying out rope which simply lies quietly upon the ground as it is left by the car. The rope is issuing from the eyelet with the *same* velocity that the car moves forward *relatively to the car,* but that which has issued with this velocity has no motion relatively to the ground. In the same way the steam jet issuing backward from the moving blades with a velocity equal to their forward motion, is brought to rest with reference to the casing, and has given up its energy to the revolving member.

In Fig. 8, *AB* represents the velocity and direction with which the jet issues from the stationary guide passages, *CB* the velocity and direction of the blades. The direction and velocity of the jet,

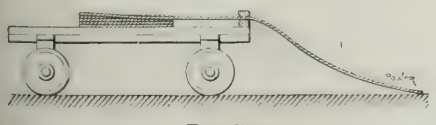


FIG. 7

relatively to the moving blades, will then be represented by *AC*. The steam simply flows, without impact or shock, into the passages as they go by, having the same amount of forward motion or velocity as the blades. It would now simply stay in the wheel and be carried around with it indefinitely, but the pressure on the delivery side of the blades is less than that on the entry side, and the steam expands in the passages between the moving blades enough so that the velocity, *A'B'*, with which it issues, is such that its component, *B'C'*, in the direction of the blade's motion, shall be equal to the blade velocity. The resultant relative motion is represented by A_1C', with a velocity proportional to which the steam flows into the stationary nozzles to be expanded again until its velocity,

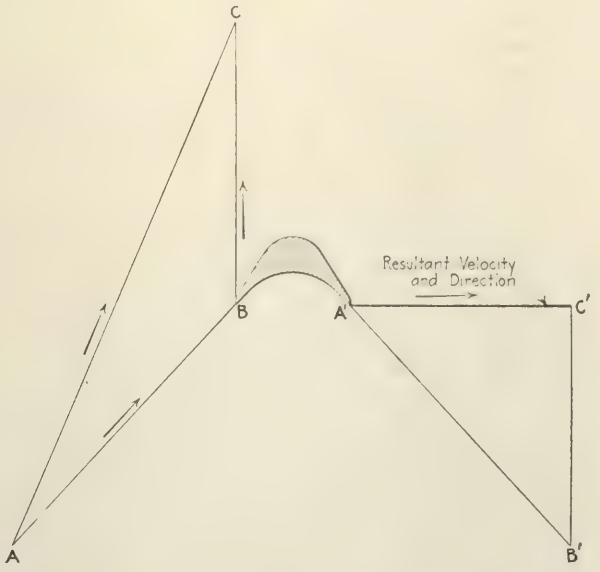


FIG. 6

in the direction in which they are moving, equals that of the blades.

The passages between the blades, notwithstanding the steam is expanded in them, grow smaller in the direction of flow, that is, they form converging nozzles. This is because the increased velocity allows the steam to pass through a

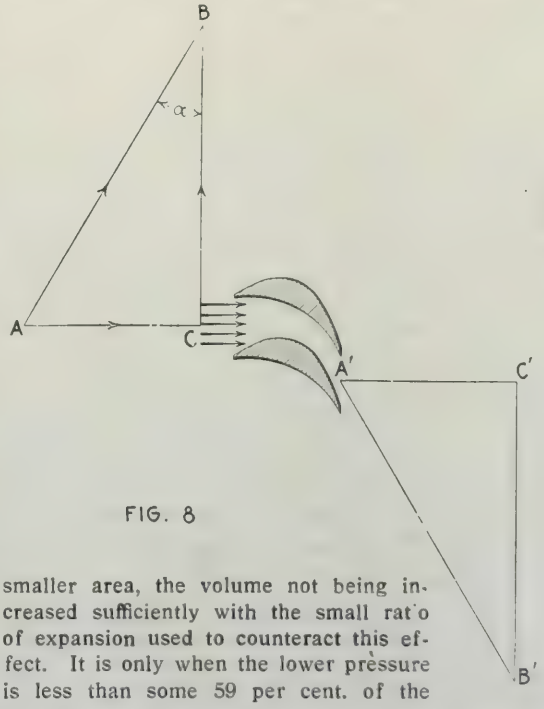


FIG. 8

smaller area, the volume not being increased sufficiently with the small ratio of expansion used to counteract this effect. It is only when the lower pressure is less than some 59 per cent. of the initial that a divergent nozzle like that of DeLaval is needed, and this is convergent at the entering end.

In the ideal case above considered, the blade velocity *CB*, Fig. 8, must be pretty nearly that of the jet *AB* (0.94 for an angle at α of 20 deg.) This means either a very high blade velocity, or a low steam velocity. The blade velocity is fixed by the speed at which the turbine is to run and a practical diameter. Only a certain amount of steam passage can be allowed in the initial stationary row to admit the quantity of steam required to run the load for which the turbine is designed, and if the diameter is made too large, the blades of the first rows will have to be made so short that the clearance will be excessive in proportion to their projected area. There is therefore a limit to blade velocity. On the other hand, keeping the jet velocity down means a small expansion at each stage, and hence a large increase in the number of stages. The designer is therefore compelled by constructional and commercial considerations to use steam velocities, the components of which, in the direction of the blade movement, exceed the blade velocity, which produces the condition illustrated in Fig. 9. The resultant direction of flow is not axial. The steam enters the blade passage with a velocity proportional to, and in the direction represented by, *AC* and thus exerts an impulse in the direction of rotation. The steam then expands in the moving passages and operates by reaction, as before explained. This impulse effect is usually, if not always, present in the Parsons machine, and when it is involved, the machine is to that extent a combined impulse and reaction turbine, the reaction feature, however, predominating. With high-speed turbines as they are being

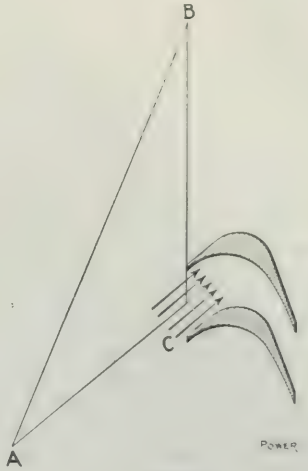


FIG. 9

built today the ideal condition is closely approximated.

The moving blades of the impulse turbine run in a chamber the pressure in which is essentially the same throughout. When the blades are carried upon disks

the latter usually are perforated to maintain equal pressure on the two sides of the disk. With the reaction type, the pressure at the entry side of the blades is greater than at the delivery side, and this difference for each stage, produces in the aggregate, a heavy end thrust which must be taken up by balancing, or dummy pistons.

The Parsons is a "full-admission" turbine, that is, the first row of stationary, or guide blades, occupies the full circumference, and the whole circle, except such part of it as is filled by the blades themselves, is available for admitting steam. The first rows of blades are mounted upon a drum of small diameter so that the blades may have a fair height without making this opening too large, and a large number of rows are used so that the pressure fall may be small in each stage, and the velocity kept down. As the volume of steam gets larger, larger drums and fewer stages are used. In the impulse type the diameters of the rotors are the same throughout.

Flue Gas Analysis for Beginners—I

By J. E. Terman

In simple language, the author explains the object of flue-gas analysis and the principle of operation, describes the apparatus necessary and tells how to prepare chemicals.

Many writers on flue-gas analysis for beginners commence by impressing the reader with its extreme simplicity. They often imply that the manipulator of the analyzing apparatus need not be familiar with chemistry to make successful volumetric determinations of the four principal gases in the products of combustion, viz., nitrogen, carbon dioxide, oxygen and carbon monoxide. Perhaps this attitude is due to their being so expert themselves that to them it is simple. Therein they are at a disadvantage in instructing the beginner; they have forgotten the little perplexities that arose when they were beginners themselves and fail to mention them.

In what follows the writer is relating his experience and describing methods adopted to overcome difficulties encountered after a comparatively short time in dealing with the subject; hence there may be many better ways to do certain things than here related.

Familiarity with the manipulation of the Orsat apparatus and the mixing of reagents was obtained by studying an excellent little book entitled "Gas and Fuel Analysis for Engineers," by Prof. Augustus H. Gill; this book is recommended to all beginners. The only analyzing apparatus used is the ordinary three-pipette Orsat, with the reagents described by Prof. Gill for determining carbon dioxide, oxygen and carbon monoxide. Hereafter for brevity, the four gases nitrogen (N), carbon dioxide (CO_2), oxygen (O) and carbon monoxide (CO), will be referred to by their chemical symbols.

Object of Flue Gas Analysis

Most engineers interested will understand the object in making flue-gas analysis and what the results obtained indicate. Knowing the flue-gas composition shows how the coal was burned, because in good boiler practice all of the coal, except that remaining as ash or clinker, passes up the stack as gas, fine ashes, or soot, mixed with about 18 times its weight of air. The burning of coal is the combination of the larger part of its constituents with oxygen from the air.

A low flue-gas temperature does not prove that all the heat has been extracted from the fuel for two reasons: First, the air taken in under the grate to support combustion carries away a certain amount of heat depending on the temperature difference between the air and the stack. Under best conditions from 12 to 15 per cent. of the heat of the fuel is so lost; hence if more air is passed through the furnace than is required for complete combustion (and frequently two or three times this amount is used) considerable heat is wasted, even though it gives a lower stack temperature. Second, the carbon of the coal, the chief heat producer, can combine with oxygen either as CO or CO_2. With plenty of oxygen and a suitable temperature CO_2 will be formed; otherwise CO. Carbon burned to CO gives only about one-third the heat produced when burned to CO_2, so low stack temperature if due to prevalence of CO does not mean economy. Escaping CO takes with it two-thirds of the heat in the coal from which it came.

A thin bed of fuel, although making an intensely hot fire, is not economical, as an Orsat will show, for the heat developed is largely expended in heating excess air instead of evaporating water. An ordinary evaporative test does not separate the furnace performance from the heat-absorbing capacity of the boiler, but a flue-gas analysis is needed to locate the trouble, if the evaporation per pound of fuel is not what it should be. It tells whether the heat in the coal has been fully utilized in the furnace and, by varying the way of firing, will show whether the fireman or the furnace is responsible for the poor results. A boiler should not be charged with the shortcomings of the fireman, or furnace, unless it is such that it cannot be equipped with a suitable furnace to burn the fuel for which it was designed.

Principle of Volumetric Gas Analysis

The volumetric analysis of gas with the Orsat apparatus consists of absorbing successively three of the four principal gases found in the products of combustion. After each is absorbed from a measured sample, the decrease of volume is measured. A strong solution of caustic potash, given sufficient time in contact with the sample, will absorb all of the CO_2, leaving only O, CO and N. This passed through potassium pyrogallate, a caustic potash solution with pyrogallic acid added, loses its O, leaving only CO and N. Finally a solution known as acid cuprous chloride will absorb the CO, leaving only N.

As potassium pyrogallate will absorb CO_2 as readily as O, the CO_2 separation must be made first through the caustic potash solution. Similarly the acid cuprous chloride solution will absorb both CO_2 and O as well as CO, hence this reagent must be used last.

Apparatus

The Orsat apparatus may be obtained from any dealer in chemical supplies and apparatus. The cost complete, including

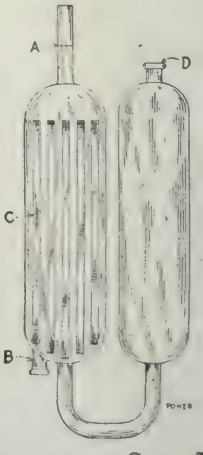


Fig. 1. Pipette with Glass Tubes in Place

chemicals and sampling apparatus, is about $30. Compared with an indicator outfit, it is cheap and, if intelligently used, the chance for saving coal is several times

that for the indicator in the average plant. Before assembling a new apparatus all parts should be thoroughly washed.

Fig. 1 shows a side view of a pipette with the front leg filled with short lengths of glass tubing *C*, the purpose of which will be explained later; there are three of these pipettes to the apparatus, as may be seen in Fig. 2.

In Fig. 1, *A* is the end to be connected with one of the branches from the main connecting pipe which terminates at *H* in Fig. 2, the other end is connected to the measuring burette at *I*. Each of the three branches from this main connecting pipe has a glass stop-cock *N* to control the connection between each pipette and the burette. The glass ends of the pipettes and the branches for the main connecting pipe are connected by 2-in. pieces of thick rubber tubing, as at *A*, Fig. 1.

By moistening the glass ends the tubing will slip on easily with less danger

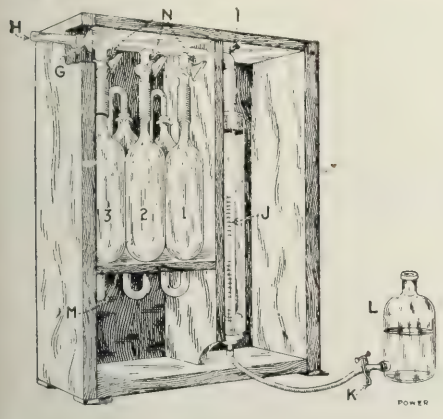


FIG. 2. ORSAT APPARATUS ASSEMBLED

of breaking the apparatus. Prof. Gill and some others recommend pinch cocks on the rubber tubing, instead of the glass stop-cocks, because the latter tend to stick. The writer uses glass cocks without any trouble from sticking by preventing reagents from running through them, and lubricating them with a very little vaseline, spread on each plug with the finger. Too much vaseline is likely to stop up the passages through the connections or the plug.

Enough short lengths of glass tubing to fill the three pipettes come with the apparatus; these should be inserted in the connection *B*, Fig. 1. One set of these tubes will contain small spirals of copper wire; these should be put in the pipette used to determine CO, No. 2 in Fig. 2. The glass tubing gives more surface for the reagent to spread over and breaks up the gas into thin layers so that the chemical reaction may be more rapid and complete.

After inserting the glass tubes in the pipettes and connecting up and placing them in the wooden frame, as in Fig. 2, the burette *J* should be connected to the right end of the main connecting pipe with rubber tubing. The burette should be thoroughly washed out beforehand and the annular space between the outer casing and the measuring tube should be filled with water to prevent, as far as possible, any change in temperature of the sample while being analyzed, as such a change would affect its volume.

The apparatus, completely assembled in Fig. 2, is ready to receive the reagents. The balance of the laboratory equipment, such as the measuring glass and bottles for holding the surplus supply of reagents, is as follows: Two bottles of 500 c.c. capacity each are used to hold a supply of dilute hydrochloric acid, and should be labeled "HCl sp.gr. 1.10," a 1000-c.c. bottle should be labeled "Potassium Hydrate" and a 500-c.c. bottle "Acid Cuprous Chloride." Four bottles, with necks on the side near the bottom, similar to the one in Fig. 4, are used, as described later, for collecting gas samples. A 500-c.c. graduate, for measuring, and a 3-in. glass funnel complete the outfit.

CHEMICALS REQUIRED FOR REAGENTS

The chemicals and materials for making the reagents and the best amount

CO_2	O	CO
11	8	0.1
12	5	1.0
13	7	0.5

POWER

FIG. 3. RECORD FOR DETERMINING CONDITION OF REAGENTS

to purchase of each are: hydrochloric acid (commercially pure), a 6-lb. bottle holding about 5 pints; black copper oxide, 1-lb. bottle; pure copper in the form of wire, about 1 lb. of No. 10 gage; potassium hydrate, which comes in sticks, a 1-lb. bottle (that purified by lime is superior to that purified by alcohol for gas analysis); pyrogallic acid in fine white crystal form which is extensively used in photographic work, a ½-lb. can.

MIXING REAGENTS

The reagent for the first pipette, No. 1 in Fig. 2, for determining CO, is made by dissolving potassium hydrate in water; the entire pound may be dissolved at once in about 900 c.c. of distilled water and kept in the stock bottle for use as required. As considerable heat is evolved in dissolving potassium hydrate in water, the bottle should be placed where, if it breaks, the contents can do no damage.

The potassium pyrogallate for the second pipette should be made up only as needed since its affinity for oxygen makes it spoil quickly by contact with air; it is therefore most satisfactory to make it directly in the pipette by adding 5 grams of pyrogallic acid to 100 c.c. of the potassium-hydrate solution, or like proportion, according to the capacity of the pipette. When the arm of the pipette containing the glass tubing is filled to the line on the neck, the rear leg of the pipette should be almost empty, say, about ½ in. above the level of the frame *M*, Fig. 2, supporting the pipettes.

The pyrogallic acid may be placed in the pipette through the opening *D*, Fig. 1, but this is laborious and the solution will be nearly as well protected from the air by placing the pyrogallic acid in the glass funnel and pouring the solution of potassium hydrate over it. The acid dissolves almost instantly and is carried through the funnel into the pipette in solution. It is not necessary to weigh the acid after becoming accustomed to mixing it, for a quantity can be gaged with sufficient accuracy with the eye. While pouring the solution into the rear leg of the pipette the stop-cock *N*, Fig. 2, controlling the connection from the pipette to the main connecting pipe, should be opened. As soon as the solution has been put in, close the cock and place the rubber bag, which comes with

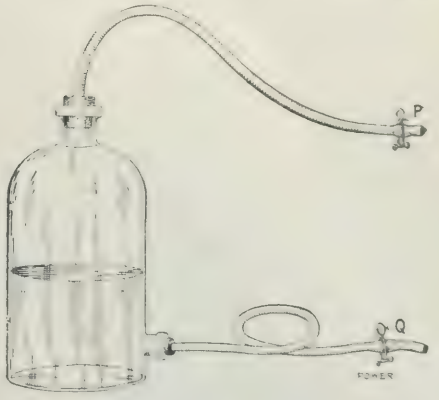


FIG. 4. BOTTLE FOR COLLECTING GAS SAMPLE

the apparatus for this purpose, over the connection *D*, Fig. 1, to prevent access of air to the reagent.

Hydrochloric acid, specific gravity 1.10, is made by diluting the commercially pure acid with an equal volume of distilled water. Enough should be mixed to fill the two 500-c.c. bottles labeled for this purpose. This dilute acid is of frequent use for neutralizing potassium hydrate that is accidently spilled. This acid is also used for making the acid cuprous chloride for determining CO. It is the most difficult reagent to prepare and the least satisfactory in its action. Prof. Gill gives the following directions for making it:

Use the 500-c.c. bottle for mixing and

place enough of the black copper oxide in it to cover the bottom about ½ in. deep; then put in eight or ten pieces of the No. 10 copper wire cut long enough to reach from the bottom of the bottle to the neck. Pour in dilute hydrochloric acid until the bottle is full, then cork tightly with a rubber stopper to keep the air out. The bottle should be well shaken and placed where warm, to hasten the chemical action. When first mixed, this reagent is a very dark brown, but after 10 days, or more, will probably clear up and the bottom of the bottle contain small while crystals with the remaining black copper oxide. The correct quantity of this clear liquid can be poured into pipette No. 3, which contains the tubes with the spirals of copper wire.

The Orsat apparatus is now ready for use. The stock bottle in which the acid cuprous chloride is prepared should be filled at once with the diluted hydrochloric acid, and it is ready for further use when occasion requires. When the acid cuprous chloride does not become clear, it is due to air reaching it, which turns it brown. This is difficult to prevent, but because the solution is dark, does not detract from its quality, if the white crystals are noticed, for their presence indicates that the solution is saturated and ready for use. This reagent is not very satisfactory on account of its weak attraction for carbon monoxide, which it is used to dissolve.

Absorption Capacities of Reagents

Each of the reagents has a limited capacity; that is, becomes worn out after it has dissolved a certain amount of gas. Therefore the different quantities of the various gases analyzed should be noted to know when the reagents should be renewed. For this purpose a small pad may be fastened to the side of the Orsat frame and the number of cubic centimeters of each gas analyzed entered on it. There should be three headings on this pad and the amounts put down, as shown in Fig. 3, so that they may be added up at any time.

Potassium hydrate will absorb 40 times its own volume of gas before becoming too weak for use. A pipette holding 150 c.c. of this solution should absorb 6000 c.c. of CO_2 before needing renewal.

Potassium pyrogallate is not nearly so active, as only twice its volume of oxygen may be absorbed before a new solution is required.

According to Prof. Gill, acid cuprous chloride has an absorption capacity, with the Orsat apparatus, of only one, that is, 150 c.c. of the reagent will absorb only 150 c.c. of CO before renewal is required. Other authorities give it considerably greater capacity than this, but as the reagent is very cheap and, at best, is unsatisfactory, the writer would recommend that no more gas than that equal to the volume of the reagent be dissolved in it before using a new solution. On account of the extremely large capacity of the reagent for CO_2, as compared with the others, it is not necessary to keep a record of the amounts of this gas analyzed, for it is only necessary to renew this solution once to every eight, or ten, renewals of the potassium pyrogallate.

Nordberg Variable Capacity Power Pump

The Nordberg Manufacturing Co., Milwaukee, Wis., has developed a variable-capacity power pump which affords simplicity of control and close regulation of output. The length of stroke is variable and the water end of any type of reciprocating pump may be used.

Power is applied to the shaft *A*, Fig. 1, and is transmitted through the reducing gears *B* and *C* to the shaft *D* and crank *E*, connected to the pin *F* by the rod G. The pin *F* ties the ends of the rods *G*, *H* and *I* together, as shown. The rod *I* is hinged on the pin *J* which is secured to the casting *K*. Rotating the crankpin *L* causes the pin *F* to swing in an arc, with the pin *J* as a center. The crosshead *M* reciprocates horizontally between its guides a distance depending upon the position of the arc described by the pin *F*.

The casting *K* is hinged to the side frames of the machine and rotates about *N* as a center and the pin *J* may be turned to any point in an arc about the center *N*. The radius of the arc is such that by rotating the casting *K* sufficiently in a counter-clockwise direction from its position shown, the center line of the pin *J* will coincide with the center line of the

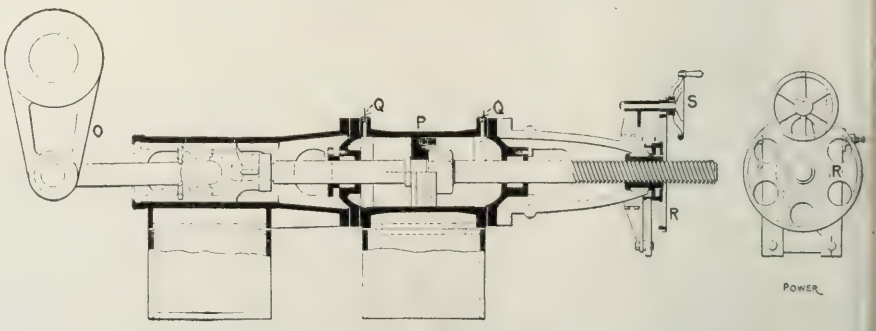


Fig. 2. Details of the Adjusting Mechanism

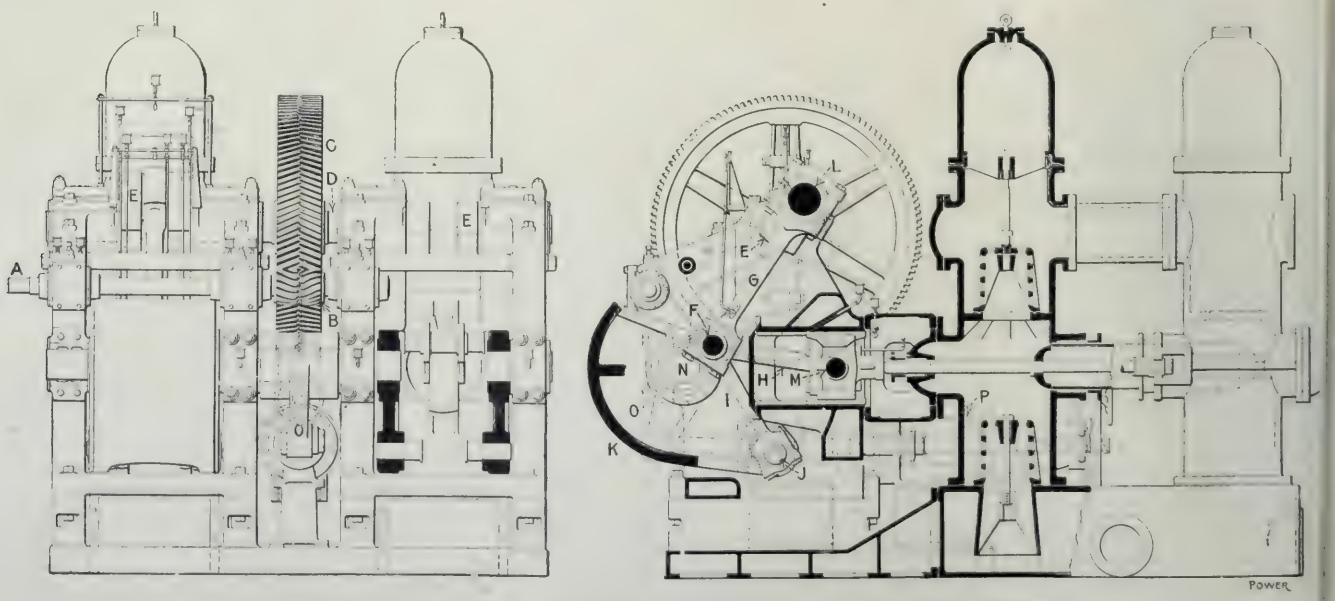


Fig. 1. Showing Design of the Nordberg Variable Capacity Pumping Engine

crosshead pin at *M* when the latter is at the extreme end of the out stroke. The rods *H* and *I* are the same length and when the center lines of *J* and *M* coincide, the pin *F* swinging about *J* as a center produces no motion of the crosshead *M*. By adjusting the position of *J*

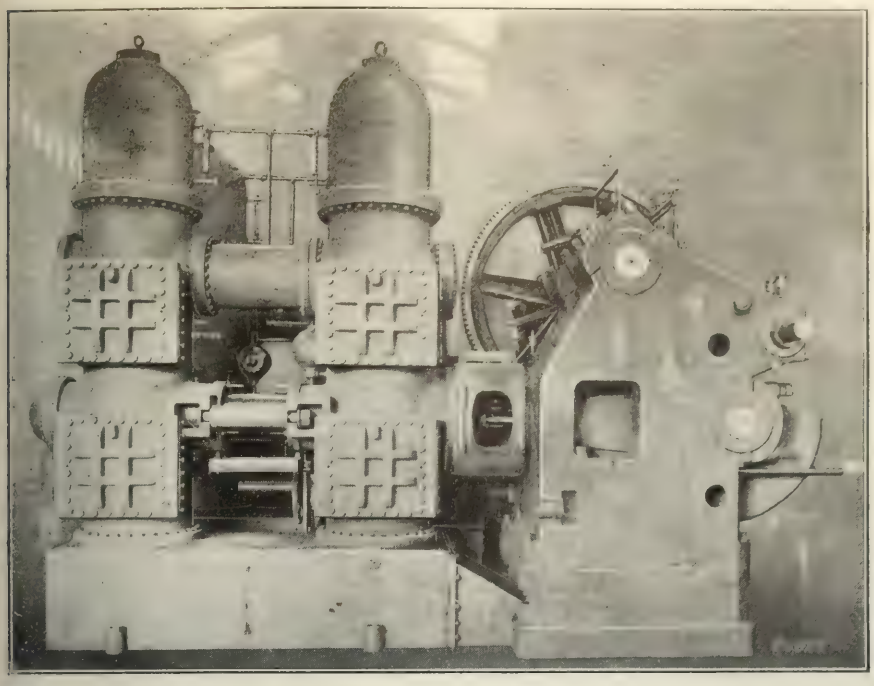


FIG. 3. SIDE VIEW OF ROCHESTER PUMP

any length of stroke from zero to the maximum may be obtained.

The position of the casting *K* is adjusted by the arm *O* which is connected to the piston in the hydraulic cylinder *P*. This piston may be operated automatically, by a float in the discharge reservoir, a pressure-controlled pilot valve, etc., or

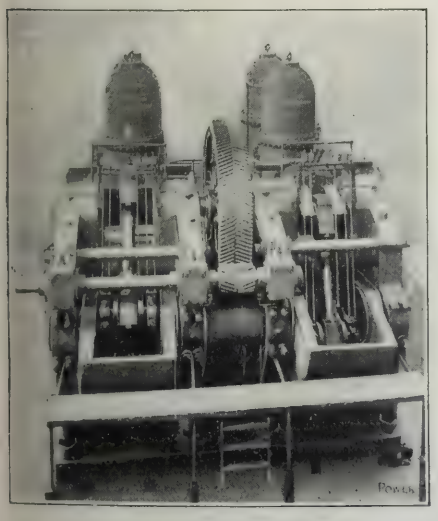


FIG. 4. END VIEW, SHOWING DRIVING MECHANISM

it may be controlled by hand as in the present case. Fig. 2 shows the construction details of the hydraulic cylinder *P* and the connections with the arm *O*.

The pump shown in Figs. 3, 4 and 5 has a capacity of 6,000,000 gal. per 24 hr. when operating at full stroke, 18 in. The plungers are 16⅛ in. in diameter and the speed is 67 r.p.m. The pump works against a static head of 530 ft.

The piston in the hydraulic cylinder *P*, Figs. 1 and 2, is operated by water pressure through a four-way valve communicating with the ports *Q*, Fig. 2. An adjusting rod attached to the piston extends through the end of the cylinder and has high-pitch screw threads cut in its outer end. A wormwheel *R*, con-

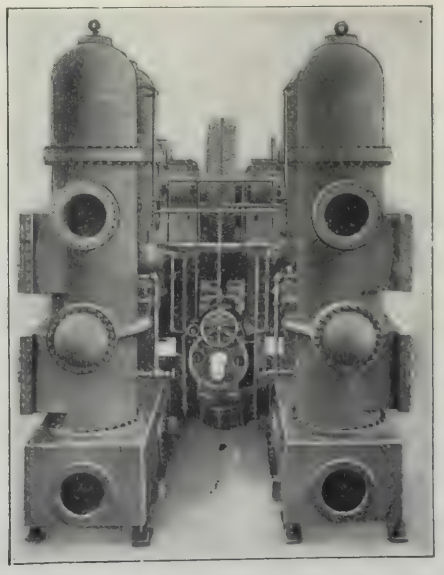


FIG. 5. VIEW SHOWING CONTROL WHEEL

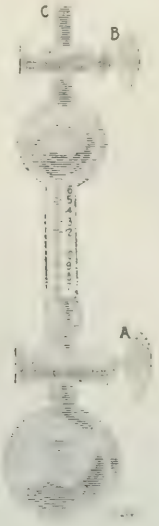
trolled by a handwheel *S*, rides on this screw and acts as a brake to prevent the piston from moving except when operated by the handwheel.

The output of the pump is changed by adjusting the four-way valve to admit water pressure to the proper end of the cylinder and exhaust it from the other; the handwheel is then turned to allow the piston to move in the desired direction. A 600-hp., three-phase, 60-cycle, synchronous motor running at 500 r.p.m., will run the pump. Although directly connected, the motor may be started light and the load applied as slowly as desired.

The reducing gears are of the Wuest staggered herring-bone machine-cut type, and are noiseless.

Simple CO_2 Meter

A simple apparatus for measuring CO_2 is shown in the accompanying engraving, reproduced from our German contemporary, the *Berliner Tageblatts*. The lower globe is filled with the absorbing liquid, 260 grams of caustic soda in one liter of water, and the opening just above it is completely closed with a rubber cork. One filling will suffice for some 200 samplings.



A SIMPLE CO_2 METER

When the apparatus is to be used, the rubber cork is removed and replaced with the cock *A*. The apparatus is then drawn full of the gas to be tested, the upper cock *B* closed, and the under cock *A* turned to place the absorbent in communication with the gas. The apparatus is inverted so that the alkali solution will run into the gas-holding chamber, and, after the gas is thoroughly absorbed, is again placed in the position shown in the engraving so that the liquid will return to the lower globe. The cock *A* is then closed, the whole apparatus placed under water, and the cock *B* opened when water will enter through the opening *C* to replace the gas absorbed by the liquid.

The apparatus should now be held upright with the opening *C* still under water until no more water will run into it,

when the level of the liquid inside is the same as that upon the outside. Then the cock *B* should be closed, the apparatus returned to the position shown in the engraving, the cock *B* opened to allow the water in the neck *C* to pass to the interior, and the level of the water upon the graduated scale will indicate the percentage of CO_2.

Laying Out Power Plant Piping

By J. P. Lisk

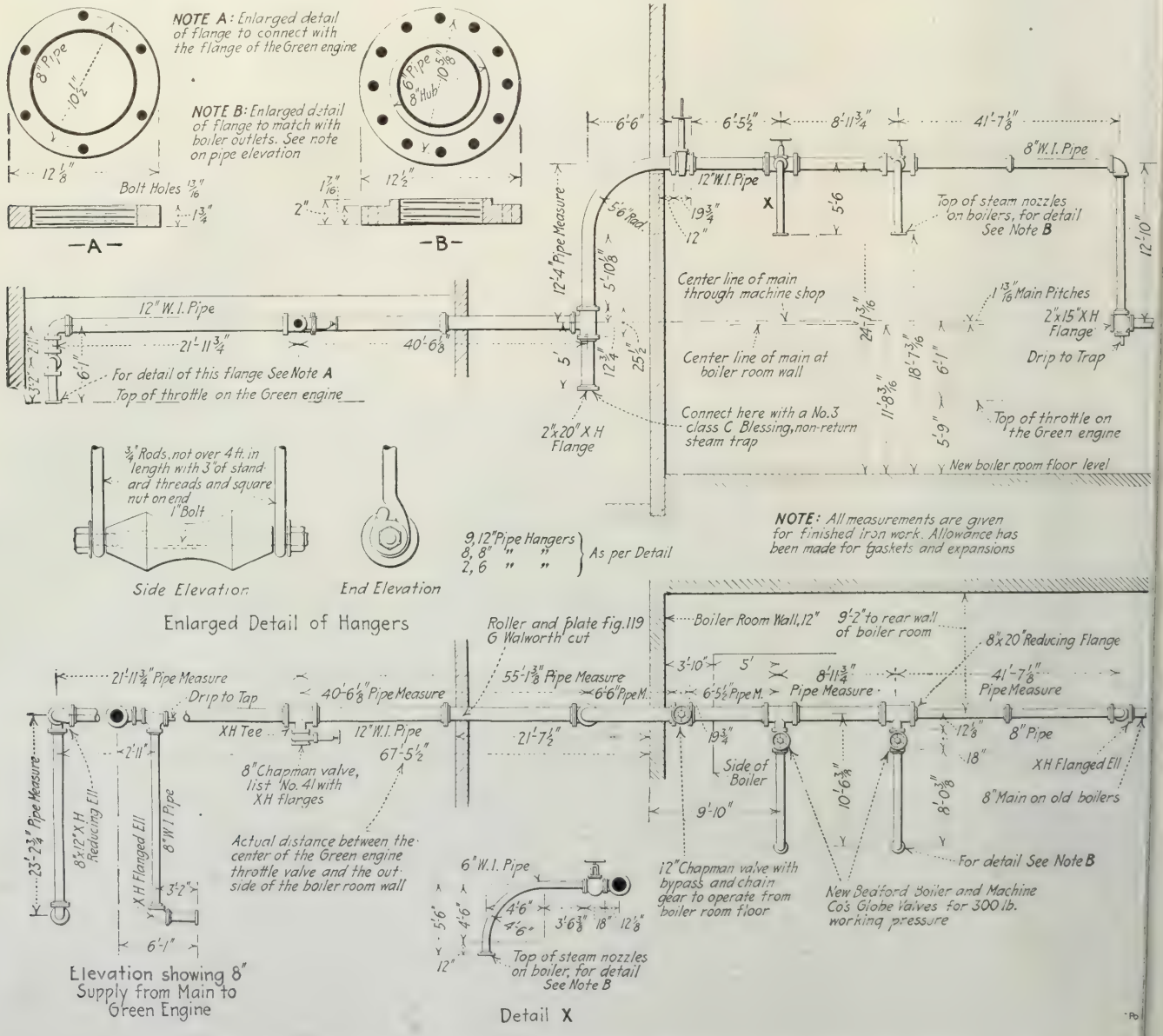
A simple illustration from a practical engineer showing what measurements to take and how an accurate plan can be made intelligible to an ordinary workman.

When an engineer is called upon to lay out the piping system for a power plant he should have in mind, first, safety; second, efficiency; third, durability. Extravagance and excessive economy are to be avoided, as either, if carried to the extreme, paves the way for disastrous results. High fixed charges result in excessive cost of production, and too much economy brings about the same result in excessive repair bills, with the added possibility of wrecking the plant and causing loss of life.

Starting with the maximum pressure, the engineer makes his calculations and specifies plainly just the kind of material and fixtures to be used, giving size, capacity and some make as a standard that shall be accepted, or one of equal make subject to approval. Methods of attachment and support and every part of the work must be so fully described that a mistake in carrying out his intention is practically impossible.

After the specification is written, the plan is made, and with the specification and free-hand sketches made by the engineer when working out his calculations, it is an easy matter for the draftsman.

In making plans it is essential to include and dimension accurately all that part of the building within which the apparatus and piping is to be placed. Everything must be made perfectly clear, even if it does take considerable paper


NOTE A: Enlarged detail of flange to connect with the flange of the Green engine
NOTE B: Enlarged detail of flange to match with boiler outlets. See note on pipe elevation
8" Pipe
10½"
12⅞"
Bolt Holes ⅞"
1¾"
—A—
6" Pipe
8" Hub
10⅜"
12½"
1 7/16"
2"
—B—
6'-6"
6'-5½"
8'-11¾"
41'-7⅛"
8" W.I. Pipe
12" W.I. Pipe
19¾"
12"
5'-6"
Top of steam nozzles on boilers, for detail See Note B
12'-10"
12'-4" Pipe Measure
5'-10⅜"
Center line of main through machine shop
1 13/16" Main Pitches
2"x15" X H Flange
Drip to Trap
Center line of main at boiler room wall
Connect here with a No. 3 class C Blessing, non-return steam trap
2"x20" X H Flange
Top of throttle on the Green engine
New boiler room floor level
12" W.I. Pipe
21'-11¾"
40'-6⅞"
For detail of this flange See Note A
Top of throttle on the Green engine
¾" Rods, not over 4 ft. in length with 3" of standard threads and square nut on end
1" Bolt
Side Elevation
End Elevation
9, 12" Pipe Hangers
8, 8" " "
2, 6 " "
As per Detail
Enlarged Detail of Hangers
NOTE: All measurements are given for finished iron work. Allowance has been made for gaskets and expansions
Roller and plate fig. 119 G Walworth cut
Boiler Room Wall, 12"
9'-2" to rear wall of boiler room
8"x20" Reducing Flange
21'-11¾" Pipe Measure
Drip to Tap
40'-6⅞" Pipe Measure
55'-1⅜" Pipe Measure
6'-6" Pipe M.
6'-5½" Pipe M.
8'-11¾" Pipe Measure
41'-7⅛" Pipe Measure
XH Tee
12" W.I. Pipe
67'-5½"
21'-7½"
8" Chapman valve, list No. 41 with XH flanges
Side of Boiler
9'-10"
12⅞"
8" Pipe
18"
XH Flanged Ell
8" Main on old boilers
10'-6⅜"
8'-0⅜"
For detail See Note B
Actual distance between the center of the Green engine throttle valve and the outside of the boiler room wall
12" Chapman valve with bypass and chain gear to operate from boiler room floor
New Bedford Boiler and Machine Co's Globe Valves for 300 lb. working pressure
23'-2¾" Pipe Measure
8"x12" X H Reducing Ell
8" W.I. Pipe
XH Flanged Ell
3'-2"
6'-1"
Elevation showing 8" Supply from Main to Green Engine
6" W.I. Pipe
4'-6"
3'-6⅜"
18"
12⅛"
5'-6"
12"
Top of steam nozzles on boiler, for detail See Note B
Detail X

WORKING DRAWING FOR SMALL PLANT, GIVING NECESSARY DIMENSIONS AND NOTES

and tracing cloth, and it is important not to make the sheets too large. It is much better to have several that may be conveniently handled than one large sheet that must be folded or laid on the floor to be read. From experience in the field the author can testify that making up a list of material and getting dimensions from a 36x48-in. blueprint is very annoying work. Prints, 24x30 in., outside measure, are the largest that should be made. If the plant is large and complicated, a small scale drawing should be made, showing the location of the various apparatus and piping, and then one-quarter section sheets to a larger scale, and details to a still larger scale. Following this order the writer has made excellent plans of a complete plant of large size, and all on a uniform sheet 18x24 in.

Always should be given the center-line measurements of the apparatus to be installed, the distance from the walls of the building, the distance between machines and the heights to the various outlets.

The piping layout being about the last thing to be put on the plan, is usually made in a hurry, and the draftsman often does not hesitate, when he meets with some apparent obstruction, to dodge it with a bend or go around it with fittings and short pieces, where perhaps some slight change further back in the line would give a perfectly straight run.

After the fixtures are all properly located with plenty of room around each to operate, clean and repair, the engineer should run his pipes to and away from each fixture in such a way that all valves and parts of the apparatus may be easily reached, and not conflict with the operation or repair of the apparatus to which they are attached. All of these branch lines should be carried up or down, as the case may be, and then the main lines laid out, which are of comparatively large size, so that all unnecessary turns may be avoided, even at the expense of making the branch lines longer and larger than would be necessary if the main was brought nearer to the fixture. Finally, one should look carefully into the details of support; this should not be left to the judgment of the steamfitter. It is not sufficient to mention in the specification that a suitable hanger shall be placed at intervals of 10 ft. on all lines of pipe, or a pipe chair with roller where such an arrangement is convenient. Where several extra-heavy fittings and valves are located in a 12-in. line, the weight to be supported runs up, and in view of the vibration, in addition to the dead weight, the problem of putting it up to stay deserves attention.

The accompanying drawing is an illustration of a high-class job of piping that filled all requirements of the service for which it was designed, namely, 125 lb. working pressure, without any of those features in its makeup that might be called extravagant or unnecessary. The drawing speaks for itself so that a description would be merely repetition.

A Machinist's Engine Room Experience

By Donald A. Hampson

When the new engineer at the woolen mills shut down the engine on his first night he failed to open the drips and he also closed the exhaust valve. As the throttle valve leaked considerably, a great quantity of condensation lodged in the cylinder. With conditions unchanged, he started up the next morning and things happened on the return stroke of the engine.

Instead of the usual cylinder-head accident, the connecting-rod bent to an obtuse angle. On the forward stroke it failed to straighten out and the movement of the flywheel, weighing several tons, turned the shaft within the crank until the key was crushed into a "Z" shape.

Attempts to remove the key with a sledge-hammer failed utterly, so a brace was rigged up from the wall for ratchet drilling. After a 1½-in. drill had been run 10 in. into the 14-in. long key, it was possible to drive out the piece. The crank was sent to the shop to be dressed and the keyseat was chipped out for a 1½x2¼-in. key and also for a 1½-in. square key which the owner insisted on having put in on the quarter. When assembled and fitted, using a 16-lb. sledge, it was reasonably certain the keys would never fail again.

The engine was a Corliss and had broken the bell-crank on one of the admission valves. When removing the valve a ½-in. nut was found wedged between its edge and the seat. How it got there was a mystery. The plant was 5 miles off in the woods and the machinery was chiefly that discarded from other plants. Naturally the engineer had to be "all hands and the cook," aided by his assistants. The buildings were strung out along the brook, with the boiler room 200 ft. up the valley in another building.

It developed that one of the "assistants" had been reseating the globe valve over one of the boilers a fortnight before the accident to the engine and had dropped the nut holding on the handwheel into the steam pipe, and had then closed the valve and said nothing about it. This same nut had been carried through the 200 ft. of pipe, the throttle valve on the engine, and had failed in its attempt to reach the open air again.

"We want a man to measure our cylinder for a new piston ring," said the call from the glass works. When the writer reached there, the engineer and the "boss" were waiting around the piston and cylinder head which they had taken out. The piston head was of the built-up type with cast-iron spider and follower and a bronze bull ring grooved for four rings and split diagonally. The bull ring was broken into five pieces and, judging from the smoothed edges, it had been so a long time; likewise, the faces of the grooves were worn so that a shoulder near the bottom was quite distinguishable, showing that the expansion rings had had "a little movement all their own" for many days.

When the writer asked to be shown the rings he was told, "Dere vos no odder rings. Der brass ist der ring." Wonderful! He learned also that during the plant's seven years' operation the cylinder had not been opened, and that the amount of exhaust steam had always seemed abnormal required no stretch of the imagination to credit.

The writer was instructed to make a new ring like the broken bronze one, but no cast-iron expansion rings. A plea to do the latter availed nothing—partly due probably to the writer's appearance perhaps suggesting a lack of experience. However, castings were made for a complete set of five rings and turning the bull rings had just been finished when the boss came in the shop. With the air of a man springing a new one, he displayed a piston ring from an 8x14-in. engine loaned him by a friend as a sample of what should be put in the disabled engine. Without informing him that there was a similar blank casting under the lathe, the writer agreed with him that it would be a good thing to do. The next morning the entire set was working in the engine. The owner was square at heart, for a month later he telephoned his thanks for the "wery goot yob on my engine," adding that according to the figures of one month his coal bill would be reduced over $700 a year.

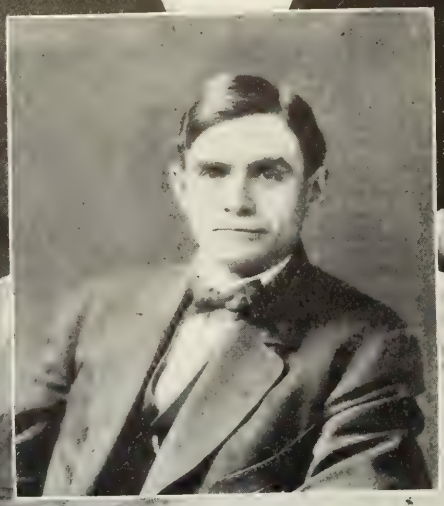
The funny side sometimes makes its appearance in the engine room. One of the engines was being "limbered up" after having its bearings babbitted, and the oil guards had all been replaced. The crankpin was a circle of bright, revolving metal made conspicuous by the nickeled arm of the oiler extending from its stand to the revolving pin.

Enter Mr. Blank, the lawyer receiver of the property during its financial troubles, who stood for a while watching that piece of ½-in. nickel tubing; then he said: "It always seems strange to me, Mr. Jones, that that slender rod could support such a heavy crank."

A floating dock, large enough to take the largest battleship in the British fleet, or the largest merchant vessel trading in the St. Lawrence, left England, Aug. 10, for Montreal. This dock is 600 ft. long and 135 ft. wide over all, and the walls are 59 ft. high.

30 TH ANNUAL N·A·S·E· CONVENTION KANSAS CITY.

GOV. HADLEY OF MISSOURI



MAYOR JOST OF KANSAS CITY

GOV. STUBBS OF KANSAS

CASINO HALL

CONVENTION HALL

JOHN H. ATWOOD

THE EXECUTIVE COMMITTEE:
C.E. EDGAR, E.H. LANE, J.J. WHITE, W.F GROOM, F.B. DOGGETT.

Missouri River Power Station

By A. R. Maujer

Description of plant containing a 15,000-kw. and a 5000-kw. Curtis vertical turbine; a 10,000-kw. Westinghouse-Parsons horizontal turbine, and three 3000-kw. Allis-Chalmers vertical reciprocating engines.

A feature of this plant is the duplicate main steam piping.

The larger and more important of the two generating stations of the Metropolitan Street Railway Co., of Kansas City, Mo., which operates all the street cars in that city, and its neighbor, Kansas City, Kan., is the Missouri River plant, Fig. 1, at Second St. and Grand Ave., Kansas City, Mo. The station was erected in 1904, just before the reputation of the steam turbine as a reliable and efficient machine was established, and hence the first installation of generating equipment was reciprocating-engine driven. Later, when additional capacity was needed, turbines were installed, resulting in the present mixed equipment of the station.

The building is 188 ft. wide by 260 ft. long, inside, divided into boiler house, 89 ft. wide, and engine house, 99 ft. wide. The height from the basement floor to the roof is 125 ft.; from the engine-room floor, 100 ft.

15,000-KW. UNIT

The largest capacity unit in the station is the most distant turbine seen in Fig. 2. This was installed in 1911 and stands on the same foundation formerly occupied by a 5000-kw. Curtis turbine. It is a 15,-000-kw., maximum-rating, six-stage Curtis vertical turbine running at 750 r.p.m. and generating three-phase, 25-cycle current at 6600 volts. The condenser, which is of the base type, contains 30,000 sq.ft. of cooling surface. The condensate is pumped by two Alberger 4-in. single-stage centrifugal pumps, each driven by a 25-hp. induction motor, and an Alberger 5-in. two-stage pump driven by a 50-hp. horizontal turbine running at 2500 r.p.m.

The main dry-vacuum pump is an Alberger two-stage crank-and-flywheel machine. The two vacuum cylinders, 34 in. in diameter each, are in tandem with each other and with the single steam cylinder, 12 in. in diameter; the common stroke is 24 in. The steam valves are Corliss type with a nonreleasing gear. For simplicity the secondary vacuum cylinder is the same diameter as the primary and has similar parts, and the valves are operated by the same valve rod. The vacuum cylinder walls and heads are jacketed for cooling water and the piston-rod stuffing-box of the primary cylinder has a device which diverts to the suction of the secondary cylinder any air that may leak by the outer packing, thus preventing impairment of the vacuum in the primary or high-vacuum cylinder through leakage.

To attain high efficiency by eliminating friction the ports and passages of the vacuum cylinders are especially large and direct. The detrimental influence of the clearance space thus formed is offset by using an equalizing port in the body of the mechanically operated inlet valve which, when the piston is at the end of its stroke, connects the clearance space in front of the latter with the cylinder space behind it. This allows the air at discharge pressure in the clearance space in front of the piston to expand back into the cylinder space at the other end, thus reducing the pressure in the clearance space nearly to that of the vacuum being carried and greatly eliminating thereby the prejudicial effect of re-expansion when the return stroke takes place.

The auxiliary vacuum pump for this unit is a two-stage machine of the same type with 24-in. air cylinders, 8-in. steam cylinder and 24-in. stroke.

The two step-bearing pumps are Dean, duplex outside-end packed, 14x3¼x12 in. The step pressure in this turbine is 825 lb. per sq.in.

A second 15,000-kw. Curtis vertical turbine is now being installed and will be ready for service within a few months.

FIG. 1. MISSOURI RIVER POWER STATION

10,000-KW. UNIT

The next largest generating unit is the 10,000-kw., maximum-rating, Westinghouse-Parsons horizontal double-flow turbine seen at the left of Fig. 2. This operates at 1500 r.p.m.

The turbine is served by two Wheeler, admiralty-type, surface condensers containing 11,000 sq.ft. of cooling surface each. A 42-in. hydraulically operated gate valve between the turbine and each condenser allows a condenser to be cut out for cleaning without taking the turbine out of service. The condensate is handled by either of two 4-in. vertical, two-stage centrifugal pumps driven by direct-connected 15-hp. induction motors. The dry-vacuum pump is an Alberger of the same type as that for the 15,000-kw. unit except that the steam valve-gear is of the releasing type. The air cylinders are 24 in. in diameter, the steam cylinder 8 in. and the stroke is 24 in.

This unit is supported on steel struts and girders and the space beneath is occupied by the condenser pumps, gland-water pumps and the auxiliary oil pump which is used when the turbine is being started. By this arrangement considerable economy in space is effected. This unit was installed in 1910.

5000-KW. UNIT

The third turbine is a 5000-kw. Curtis vertical five-stage machine, running at 750 r.p.m., and having an overload capacity of 50 per cent. This unit also is supported on steel struts and girders instead of a solid concrete foundation, in

the same manner as the Westinghouse-Parsons unit. It was installed in 1906.

The condenser for this turbine is of the Wheeler exhaust-base type and contains 10,000 sq.ft. of cooling surface. Two 42-in. hydraulically operated gate valves between this turbine and its condenser make it possible to cut out this condenser also, without shutting down the turbine. Two single-stage 4-in. centrifugal pumps, driven by 15-hp. induction motors, take care of the condensate. The dry-vacuum pump is an Alberger two-stage of the same size and type as that by the General Electric Co. and are of 3000-kw. rated capacity, with an overload rating of 50 per cent.

Each engine is served by a separate surface condenser containing 10,000 sq.ft. of cooling surface. The condensers have Blake 14x38x21-in. combined air pumps.

The engines are oiled by a gravity system which gives a pressure of about 40 lb. per sq.in. at the bearings. The overhead tank has a capacity of 600 gal.

A few figures contrasting the size of one of these engine units with a turbine unit may be of interest. An engine occupies an area of 1776 sq.ft. while the 15,000-kw. turbine occupies 306 sq.ft. In an engine the total weight on the bearings is 520,000 lb. while the total weight of the moving parts of the turbine is 200,000 lb. Referring these figures to the unit of rated capacity, the engine area per kilowatt is 0.592 sq.ft. and the weight of the moving parts, 173 lb. The turbine area per kilowatt is but 0.0204 sq.ft. and the weight of moving parts 13.33 lb.

Fig. 3 shows the two motor-driven 150-kw. exciter sets running at 500 r.p.m. and one horizontal steam-turbine-driven 300-kw. unit running at 1800 r.p.m.; all generate current at 125 volts. A storage battery for excitation contains 78 cells. All of the exciters can be operated in parallel with each other or with the battery. The exciters may also be operated in series for charging the battery.

The high-tension oil switches are on the engine-room floor along the wall between the engine room and boiler room. There are 42 switches in all, six Westinghouse type and 36 General Electric all remote control. The former are solenoid operated and the latter motor operated.

The engine room contains a Pawling & Harnischfeger motor-operated crane with a main-hoist capacity of 60 tons and an auxiliary-hoist capacity of 15 tons. The span is 70 ft. and the lifting range 58 ft.

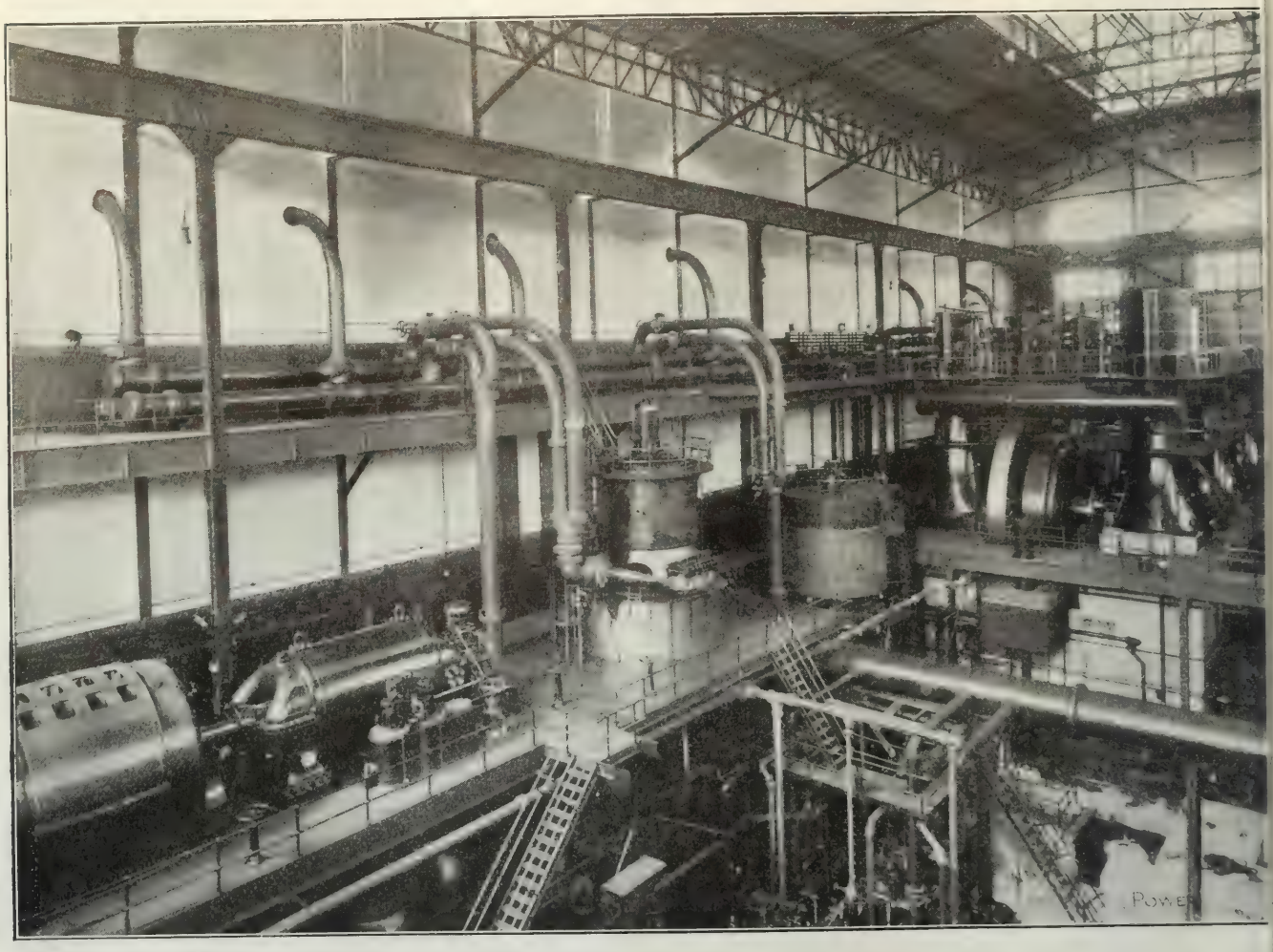


FIG. 2. THE ENGINE ROOM, SHOWING SOME OF THE AUXILIARIES IN THE BASEMENT

for the Westinghouse-Parsons unit. The same step pumps supply this turbine that supply the 15,000-kw. unit. The step pressure is maintained at 570 lb. per sq.in. A single accumulator supplies both of the Curtis units. This maintains a pressure of 1100 lb. per sq.in.

RECIPROCATING ENGINES

The three reciprocating-engine units at the far end of the station, as seen in Fig. 2, are all of 5000-hp. rated capacity, each running at 75 r.p.m. They are Allis-Chalmers 46 and 94 by 60-in. vertical cross-compound engines, fitted with Corliss valves. The dynamos were built

BOILERS

In the boiler room are 40 B. & W. 600-hp. boilers, arranged in two tiers one over the other. These generate steam at 175 lb. and have superheaters giving about 100 deg. of superheat. Four of the boilers have Foster superheaters and the rest B. & W. Coal is burned under 36 of the boilers and fuel oil under the other four. The coal-burning boilers have Green chain grates; two of the oil-burning boilers have C. Y. Roop burners and the other two Billow burners.

Two radial brick stacks, 265 ft. high above their foundations and 16 ft. in internal diameter, supply the draft. Two flue connections lead into each stack, and the stacks have brick baffle walls extending well above the flue openings.

Coal is unloaded from cars on the railroad siding at the end of the boiler house into hoppers feeding it into the buckets of two Mead-Morrison motor-driven conveyors which raise it to the overhead bunkers, both on the same level above the upper tier of boilers. The conveyors have a capacity of 70 tons per hour when traveling at 40 ft. per minute. The buckets are of the overlapping type, 24x24 in. in size. The total bunker capacity is 12,000 tons.

From the bunkers the coal is fed into traveling hoppers by which it is distributed to the stokers. At one end of the plant one hopper serves two boilers while at the other, one serves four.

The return runs of the conveyors extend under the ashpit hoppers and receive the ashes, which they carry to two cylindrical hopper-bottom tanks over the railroad siding, each holding three carloads.

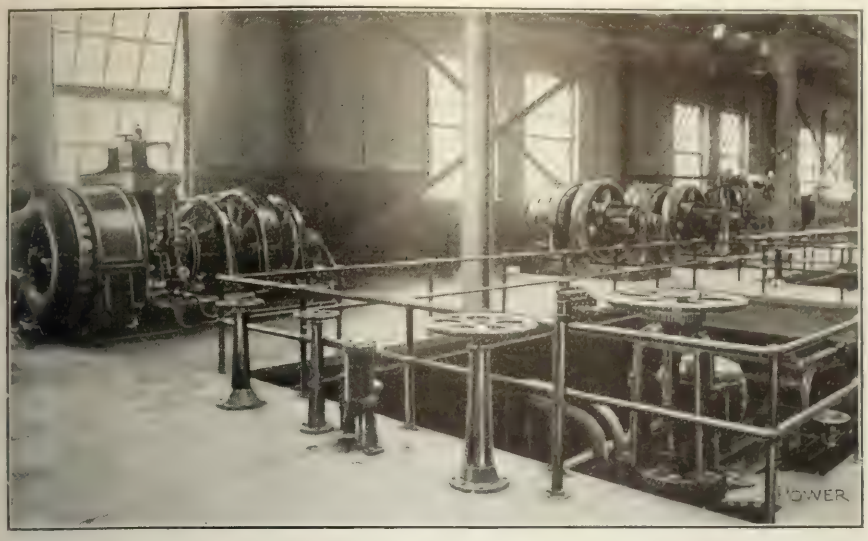


FIG. 3. EXCITERS IN THE MISSOURI RIVER STATION

FEED PUMPS

There are four Blake compound duplex feed pumps with steam cylinders 12 and 20 in. in diameter, water cylinders each 10 in. in diameter and a stroke of 18 in. The water ends are of the outside end-packed pot-valve type. The 5-in. feed-water mains are in duplicate throughout.

Three Blake 6x14x12-in. simplex pumps raise the water from the hotwell to the feed-water heaters. The hotwell for the engines is 12 ft. wide, 25 ft. long and 9 ft. deep. The condensate from the turbines is pumped direct to the heaters. Four Webster open-type induction heaters of 5000-hp. capacity each, raise the boiler-feed temperature to about 170 deg. F., and receive the exhaust steam from all of the auxiliaries.

PIPING

An interesting feature of the main steam piping is the fact that it is in duplicate throughout. Two main steam headers, each 18 in., extend along the engine-room wall on the mezzanine gallery, as shown in Fig. 2. These are anchored at the middle, allowing expansion both ways. Each pair of boilers connects through a long-radius bend to a header. Alternate pairs connect into the same header, so that not more than two boilers ever need to cut out to work on a branch. The leads to the reciprocating engines are 10 in. and in duplicate up to the throttle valve. The lead to the 5000-kw. turbine also is in duplicate to the throttle, but for the Westinghouse-Parsons unit only the bends from the headers are in duplicate and for the 15,000-kw. Curtis unit the lead is in duplicate half way to the throttle. All of the turbine leads are 12 in. in diameter.

All condensation from the piping system is returned to the boilers by a Holly system steam loop.

The steam mains for the auxiliaries are connected to the boilers by separate 4-in. connections containing automatic nonreturn valves. They form two inter-connecting loops varying in diameter from 6 to 5 in., one of which feeds the turbine auxiliaries and the other the engine auxiliaries and the steam-driven exciter. The feed pumps are supplied by two 7-in. branches which connect with the main headers and also with the two auxiliary loops.

A 48-in. main atmospheric exhaust header, to which all of the units are connected, extends the length of the building beneath the engine-room floor and connects with a 60-in. exhaust stack at one end and a 48-in. stack at the other.

CIRCULATION WATER SYSTEM

The cooling water for the condensing system is obtained from the Missouri River, 1200 ft. away. A small brick and concrete building, on the bank of the river, contains the equipment shown in Fig. 4. Four 30-in. steel-pipe siphons allow the water to flow from the river into four suction wells 12 ft. square. After passing through the screens, made of flat iron bars on edge, the water flows to the various pump-suction pipes. The seven vertical centrifugal circulating pumps were built by the Morris Machine Works and are driven by General Electric induction motors running on three-phase, 25-cycle alternating current at 440 volts. Three of the pumps are driven by 200-hp. motors, are of 10,000 gal. per min. capacity and have a 20-in. discharge opening; three are driven by 125-hp.

FIG. 4. CIRCULATING WATER PUMP HOUSE

motors, are of 6000 gal. capacity and have an 18-in. discharge, and one is driven by a 75-hp. motor, has a capacity of 3000 gal. per min. and a 12-in. discharge.

The pumps discharge into two 48-in. flow lines, which enter the generator house at the same end but at opposite sides and extend its length. They are cross-connected at each turbine by a 30-in. pipe so that they may be used interchangeably.

A small two-cylinder single-acting 8x 12-in. vacuum pump, gear-driven by a 10-hp. induction motor, frees the siphons of any air liberated from the water in flowing into the suction wells and trapped in the tops of the siphons.

Auxiliary Plant

This company also operates a direct-current plant in Kansas City, Kan., known as the Kaw River Station. It was erected in 1896 and contains four generating units. Two are horizontal tandem-compound E. P. Allis Corliss engines driving 550-volt direct-current generators. The larger is a 30 and 60 by 48-in. engine and runs at 80 r.p.m., driving a 1500-kw. dynamo and the smaller is 22 and 44 by 48-in. engine, running 90 r.p.m. and driving a 1200-kw. dynam The other two are horizontal cross-cor pound E. P. Allis Corliss engines. On a 36 and 72 by 60-in. engine, runs at r.p.m. and drives a 1500-kw. dynamo. T other is a 28 and 56 by 60-in. engin runs at 80 r.p.m. and drives a 1200-k dynamo. All have Wheeler surface co densers.

The boiler equipment consists of s B. & W. 250-hp. water-tube boilers a six Aultman & Taylor 500-hp. water-tu boilers.

Water Works of the Kansas Cities

Description of the pumping station and municipal light and power plant of Kansas City, Kan., together with the low- and high-head plants of Kansas City, Mo.

Centrifugal pumps are employed for the low-head stages and the standard types of high-duty pumping engine for the high-head.

Usually any large city's water-works is interesting if for nothing else than its massive engines, mighty in appearance but surprisingly small in actual power developed, because they ordinarily run so very slow. The works of the Kansas Cities are particularly so because the Missouri River water, which constitutes the supply, requires treatment before it can be used for general purposes. The water must be pumped twice for Kansas City, Kan., and three times for its neighbor across the Kansas River, Kansas City, Mo.

The intakes for both cities are within a few hundred feet of each other in Kansas City, Kan., on the Kansas side of the Missouri River above where the Kansas River empties into the former and north of the city. They were located there to avoid the sewerage from the two cities and any that might be brought down the Kansas River from towns above. No settlements of any consequence are on the Missouri River for 25 miles or more above the two cities and hence, except for its extremely muddy condition, the water is fairly good to drink, as the fact that Kansas City, Mo., is the second city in point of health, alone would tend to prove.

Kansas City, Kan., Plant

Kansas City, Kan., is one of the progressive cities of a progressive state and its original water-works, erected in 1895, recently were found to be inadequate and obsolete; hence, it has a new plant, erected in 1911, which may be fairly considered to represent the best practice in this type of works. In Fig. 1, the old pump house is the low, gable-roof building at the right. The new pump house is in the right end of the new building with its steel stack beyond. At the left end of this building, just being completed, is the city's electric light and power station, mentioned later.

Water is drawn from the river by low-head pumps and discharged into settling basins where, while flowing from one basin to another, lime and alum are added to coagulate and precipitate the fine particles of clay held in suspension which make the water muddy. The water is then passed through sand filters to remove other impurities and the greater part of any bacteria it may contain. High-pressure pumps then take the water and discharge it into the distributing mains.

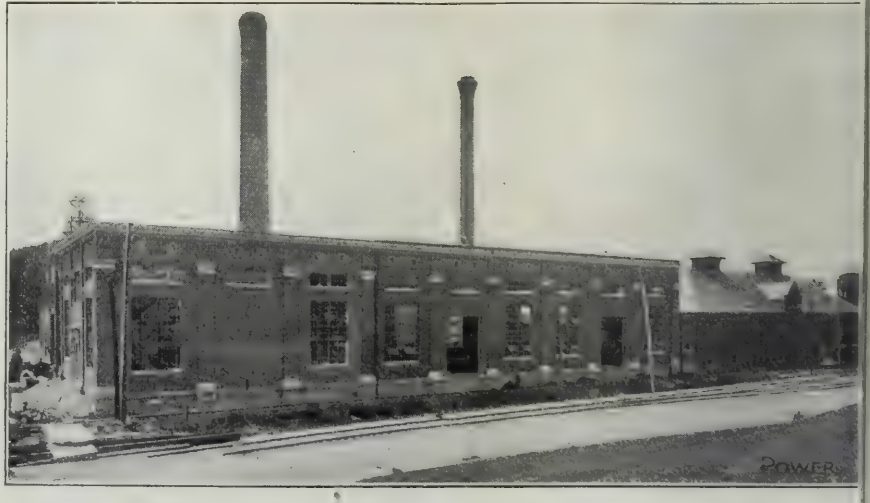


Fig. 1. Kansas City, Kan., Electric Light Plant and Pumping Station

Old Pumping Station

The boiler house of the old pumping station contains six water-tube boilers, four of 200-hp. and two of 250. These are permanently out of service now and will be removed when the old plant is demolished. The pumping equipment i cludes one Holly, vertical triple-expa sion high-duty pumping engine for t low-head stage and three Worthingt horizontal duplex pumps for the hig pressure service. The capacity of t low-head unit is 8,000,000 gal. per hr. against a total head which varies fro 24 to 44 ft., depending upon the conc tion of the river. This machine will remodeled, the water end will be enlarg and it will be moved into the new pun house at a cost of about $20,000. T capacities of the three high-pressure un are 3,000,000, 6,000,000 and 9,000,0 gal. per 24 hr. respectively. Althou these are still used during certain pa of the day, they will be dismantled ever ually and removed.

New Pumping Station

In the new pump house the water handled for the low-head stage by t unit shown in Fig. 2. The engine is horizontal tandem-compound condensi Monarch Corliss, 12 and 26 by 24 i built by the H. N. Strait Manufacturi Co. The steam inlet is 4 in. and the e haust 10 in. in diameter. The heads

both the high- and low-pressure cylinders are steam jacketed and the receiver has reheating coils which superheat the steam from 8 to 10 deg. The condenser is set alongside the pump discharge from which it receives the cooling water. The air pump is of the separate steam-driven type but exhausts into the main exhaust line and hence operates condensing.

The pump is of the double-inlet centrifugal type built by the same company. The impeller diameter is 7½ ft., the inlets are 18 in. and the discharge is 24 in. in diameter. When running at 150 r.p.m., the capacity is 15,000,000 gal. per 24 hr. On the official test this unit showed a duty of 95,000,000 ft.-lb. per 1000 lb. of dry steam at 150-lb. pressure, which is considered high for this type of pumping unit.

High-pressure Pump

The high-pressure pump is shown in Fig. 3. This is a Snow, horizontal cross-compound, crank-and-flywheel type, high-duty, direct-displacement pump, 27x60x19x42 in., having a capacity of 12,000,000 gal. per 24 hr. when running at its normal speed of 43 r.p.m. The steam-pipe connection is 6 in. in diameter and the exhaust 16 in. The receiver is supported on columns bolted to the engine frames and contains reheating tubes using live steam. A 1200-sq.ft. surface condenser is in the suction main between the two sides of the pump end and all the water pumped passes through the condenser and around the tubes. The air pump is of the single-acting bucket type driven by an arm from the low-pressure crosshead.

The valve-gear is the Snow Corliss wide-range type for both the high- and low-pressure cylinders, permitting cutoff at any point in the entire stroke. Single eccentrics are used. A Snow patent pressure regulator is connected to the high-pressure cutoff. Operated by the water pressure in the discharge main, it keeps the discharge pressure practically constant by varying the engine speed to meet the consumption.

The flywheel, 14 ft. 7 in. in diameter, weighing 35,000 lb. and cast in halves, is in the center of the engine midway between the steam and water ends. A special feature of the engine, added for efficiency and durability, is the tailrod of the low-pressure piston extending to a crosshead sliding on an outboard bearing. This is rather unusual on this type of engine except on the large sizes.

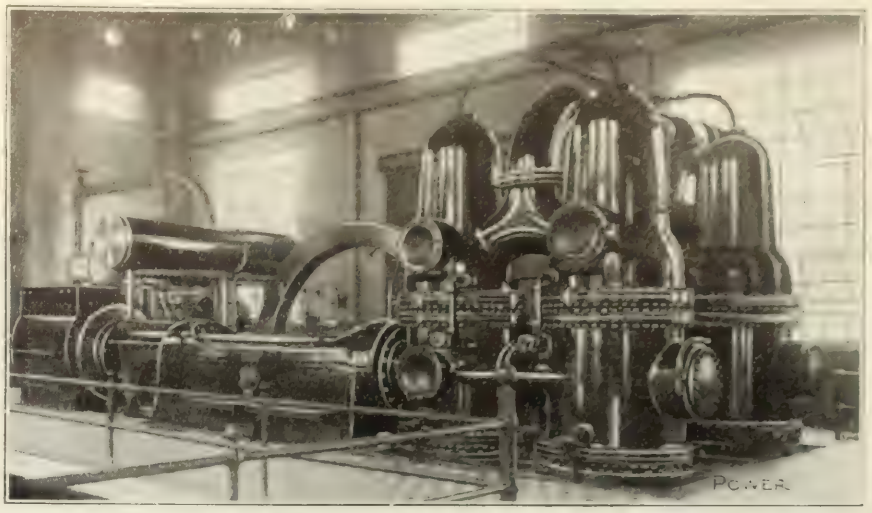


Fig. 3. High-pressure Pumping Engine at Kansas City, Kan., Plant

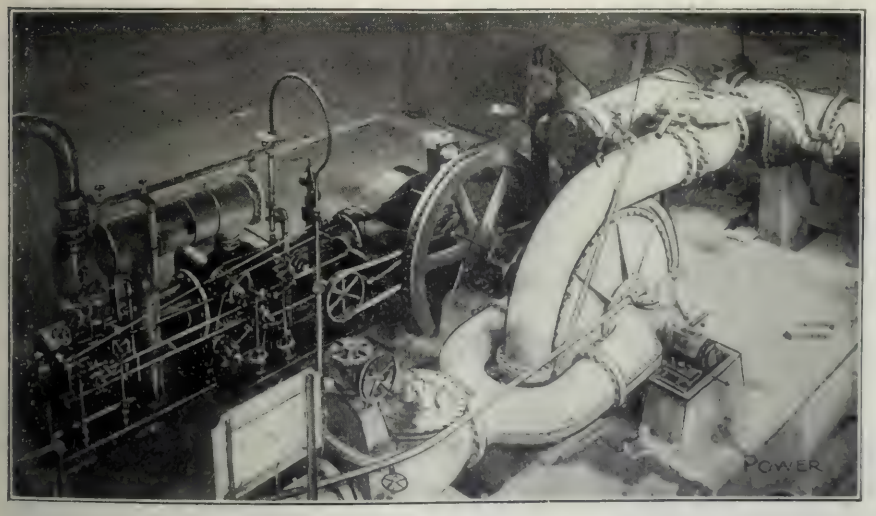


Fig. 2. Low-head Pump at Kansas City, Kan., Plant

The pump end is of the Snow outside-packed double-acting plunger built-up type. The principal castings are iron except the valve decks, which are cast steel of the double-flanged type. The pump valves, 3⅜ in. in diameter, are mounted on cages bolted to the deck. The valve area is such as to permit an average water velocity through the valve seats of 2½ ft. per sec.

In the boiler house, which is separated from the pump house by an alley 20 ft. wide, are four 300-hp. Heine water-tube boilers with Illinois chain-grate stokers. These boilers are served by the steel stack seen in Fig. 1, 135 ft. high by 6 ft. in diameter.

The boilers are fed by two Blake-Knowles 7½x5x10-in. horizontal duplex plunger pumps.

Municipal Electric Light and Power Plant

In the left end of the new building, Fig. 1, the municipal electric-light and power plant is now being installed. In addition to equipping its streets throughout with electric light the city plans to sell current for domestic and general lighting and power. It is expected that the lighting rate to small consumers will be 5 or 6 cents per kw.-hr. flat and the power rates will range from 1 to 3 cents.

Fig. 4. Quindaro Pumping Station of Kansas City, Kan.

The initial equipment will include a 500- and a 1000-kw. turbine generating alternating current at 2300 volts. Space has been provided for a future 2500-kw. machine of the same type. An interesting feature is the floor construction of the generator room. Instead of building independent foundations for each machine the floor itself was made sufficiently heavy

to carry the weight of the largest units that the space will accommodate. If the load outgrows the present equipment, substituting larger units will be simple, as tearing out old foundations and putting in new ones will not be necessary.

In the boiler house will be one 300-hp. Heine, one 300-hp. B. & W. and two 500-hp. Bonus-Freeman water-tube boilers. Room is provided for two additional 500-hp. boilers. The stack to serve these boilers is 150 ft. high by 11 ft. in inside-top diameter, of the tapered reinforced-concrete type.

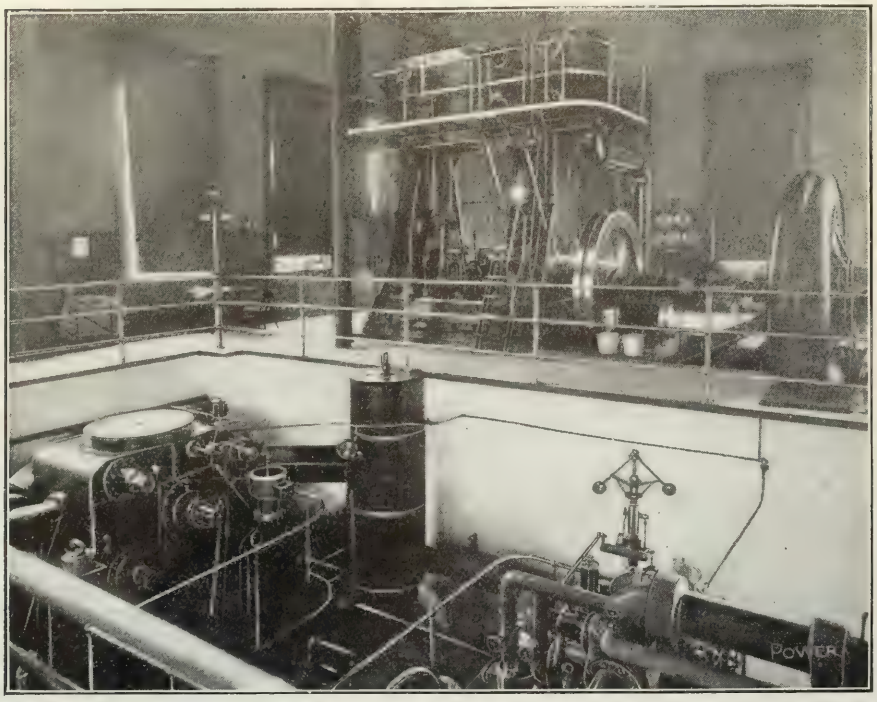


FIG. 5. PART OF THE EQUIPMENT IN THE QUINDARO STATION

The boilers of the lighting plant will interconnect with those of the pumping station and the two plants will be under the charge of one chief engineer.

KANSAS CITY, MO., LOW-HEAD PLANT

The low-head plant, known as the Quindaro station, of Kansas City, Mo., is within a few hundred feet of the plants just described. An exterior view is shown in Fig. 4, and an interior in Fig. 5. Here one set of pumps raises the water from the river to the settling basins against a mean head of 35 ft. These basins, five in number, hold altogether about 90,000,000 gal. The two suction lines from the river to the first set of pumps are 42 and 48 in. in diameter, respectively, and 800 ft. long.

There are three river pumps. One is a 25,000,000-gal.-per-24-hr. double-suction centrifugal pump, built by R. D. Wood & Co., directly connected to a Russell 13 and 26 by 24-in. horizontal tandem-compound engine, running at 200 r.p.m. Each suction inlet is 24 in. in diameter and the discharge is 30 in. The impeller is 76 in. in diameter. The two other units were built by the Allis-Chalmers Co., and are each of 30,000,000 gal. capacity per 24 hr. The engines are cross-compound vertical type with Corliss valve-gear throughout. The cylinders are 14 and 30 in. in diameter and the stroke is 24 in. The normal speed is 140 r.p.m. The three pumps discharge into a 48-in. main which delivers the water to the settling basins, 800 ft. away.

Three centrifugal pumps force the water from the settling basins to the receiving reservoir at the high-pressure pumping station at Turkey Creek, across the Kansas River in Kansas City, Mo. These pumps receive the water from the settling basins through two lines, one of which is 36 in. in diameter and the other, 42 in. One of the pumps was built by the Allis-Chalmers Co., and is the same size and design as the other two built by this company. The piping provides that, in emergency, this pump may be used for raising water from the river. The other two pumps are of 25,000,000 gal. capacity each. One was furnished by R. D. Wood & Co., and is a duplicate of their other pump. It is driven by a vertical cross-compound 14 and 28 by 24-in. Corliss valve engine, built by the H. N. Strait Manufacturing Co., running at 180 r.p.m. The other 25,000,000-gal. pump was furnished by the Byron Jackson Machine Works and is driven by a 14 and 28½ by 24-in. vertical cross-compound engine built by the Quincey Engine Co., running at 180 r.p.m.

These pumps discharge into a 36-in. and a 48-in. pipe 3½ miles long, which connect with a tunnel 7 ft. in diameter, 1125 ft. long, extending under the Kansas River. This tunnel discharges into two 30-in. pipes and one 48-in. on the other side of the river which extend 2¼ miles to the reservoir at the Turkey Creek Station.

A 50-kw. Kerr turbo-generator furnishes current at 220 volts for light and power about the plant.

The boilers, although all of water-tube type and the same general design, are of rather assorted manufacture. Of the seven boilers, two are Aultman & Taylor 300 hp., two are Stirling 300 hp., one is a Tudor 200 hp. and two are Erie 350 hp. The last two have just been installed. All are fired with coal. The Erie boilers have Green chain-grate stokers and the others Mansfield chain grates.

TURKEY CREEK PUMPING STATION

Fig. 6 is a view of the Turkey Creek pumping station, the largest of the plants. Here the water is taken by a third set of pumps from the small reservoir back of the station and delivered into the distributing mains at 150 lb. pressure.

The present building was erected in 1904 over the old station. The old building was not demolished until the new one was practically completed as it was impossible to leave the equipment without protection from the elements, and the water service had to be maintained while the changes were being made. The present plant is 188 ft. long, 92 ft. wide, and 77 ft. high from the basement floor to the roof.

FIG. 6. TURKEY CREEK PUMPING STATION

The present pumping equipment includes five units having a total capacity of 85,000,000 gal. per 24 hr. The first to be installed in the new building is shown in Fig. 7. This has a horizontal four-cylinder triple-expansion Corliss engine steam end. The high-pressure and one low-pressure cylinder are arranged tandem on one side, and the intermediate-pressure and the other low-pressure cylinder are similarly arranged on the other side. The cylinders are 24, 42, 50 and 50 in. in diameter and the stroke 42 in. The steam-supply pipe is 8 in. and the exhaust pipe from each low-pressure cylinder 24 in. Each side of the engine discharges into its own surface condenser, having 3842 sq.ft. of cooling surface.

The vacuum pumps on this and the other engines in the station are driven directly from the engine, as is customary in plants of this kind.

The water cylinders have Riedler patent mechanically operated valves. The water plungers are 20½ in. in diameter, and the capacity is 20,000,000 gal. per 24 hr.

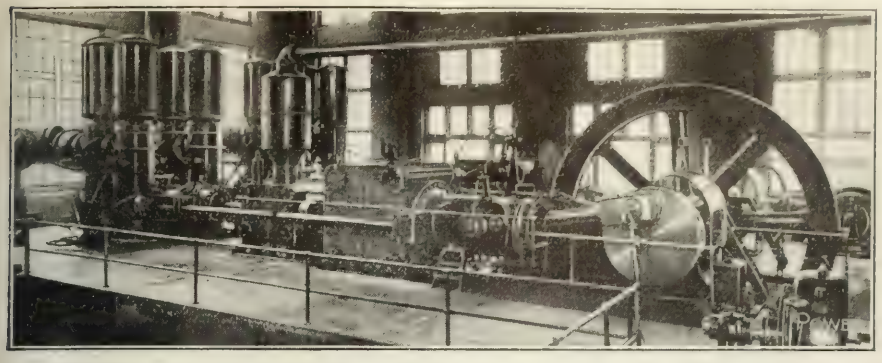


FIG. 7. HORIZONTAL TRIPLE-EXPANSION PUMPING UNIT IN THE TURKEY CREEK STATION

when running at the maximum speed of 60 r.p.m. The suction pipe is 36 in. and the two discharge pipes are 30 in. each. The suction head is 11 ft. This entire unit, including the condensers, was furnished by the Allis-Chalmers Co.

The next engine was installed in 1907. It is of the vertical triple-expansion type, as shown in Fig. 8. The steam cylinders are 38, 72 and 108 in. in diameter and the stroke 66 in. The pump plungers are 32⅜ in. in diameter and, when running at the maximum speed of 20 r.p.m., the pump will deliver 20,000,000 gal. per 24 hr. The diameter of both the suction and discharge is 36 in. The normal suction head is zero as the pump is on the level of the water in the reservoir under normal conditions. The engine has one surface condenser having 2509 sq.ft. of cooling surface. The condenser is in the pump suction line, according to the common practice with engines of this class, and hence receives the cooling effect of all the water handled. The steam and exhaust valves on the high-pressure cylinder and the admission valves on the intermediate are of the Corliss type; those on the intermediate exhaust and on both ends of the low-pressure cylinder are of the poppet type. This unit was built by the William Tod Co.

A 15,000,000-gal. Allis-Chalmers unit was installed in 1910. This also is of the vertical triple-expansion type and is 34, 62 and 94 by 60 in. in size. The high-pressure admission valves are Corliss while all the rest are poppet. The maximum speed is 22 r.p.m. The condenser contains 2016 sq.ft. of cooling surface and is in the suction line. The pump plungers are 28 in. in diameter, the suction is 36 in. and the discharge 30 in. The vacuum pump is directly connected in the usual manner.

The latest unit to be installed is shown in Fig. 9. It is also of Allis-Chalmers make and was installed last spring. The cylinders are 40, 73 and 110 in. in diameter and the stroke 66 in. The valves of the high-pressure cylinder and the intermediate admission valves are Corliss, while the rest are poppet. The condenser contains 2500 sq.ft. of cooling surface. The maximum speed is 20 r.p.m., giving a capacity of 20,000,000 gal. per 24 hr. The pump plungers are 32¼ in. in diameter. The suction is 36 in. and one of the two discharge pipes is 31 in. and the other 36 in.

The oldest engine in the station is shown in the foreground and at the right of Fig. 8. This is a vertical triple-expansion engine, equipped with Corliss valves throughout, built by the Barr Pumping Engine Co. It was installed in 1896 in the old station and stands on its original foundation today, having done duty during all the time that the new station was being built over it and the old one demolished. It has 28, 48 and 72 by 42-in. cylinders and runs at a maximum speed of 29 r.p.m. The pump plungers are 24 in. in diameter and the capacity is 10,000,000 gal. per 24 hr.

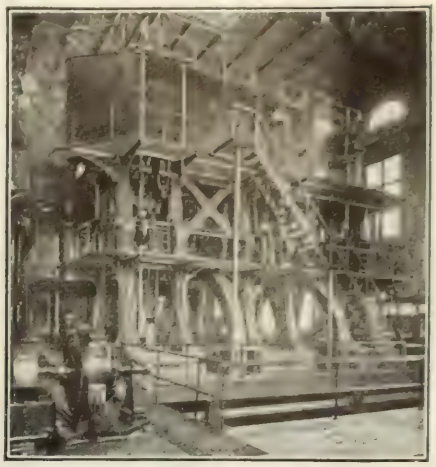


FIG. 9. THE NEW ALLIS-CHALMERS UNIT

Both suction and discharge are 30 in. in diameter.

BOILER EQUIPMENT

In the boiler room are six Aultman & Taylor water-tube boilers of 350 hp. each, two National water-tube boilers of 300 hp. each, and two Erie City water-tube boilers of 350 hp. each. The six Aultman & Taylor boilers, Fig. 10, have Billow fuel oil burners while the rest have chain-grate stokers. Fuel oil and coal at Kansas City for equivalent steam production cost practically the same; hence, oil-burning and coal-burning plants are found in close proximity and a few plants, like the Turkey Creek, use both fuels. From the increasing price of oil, however, its

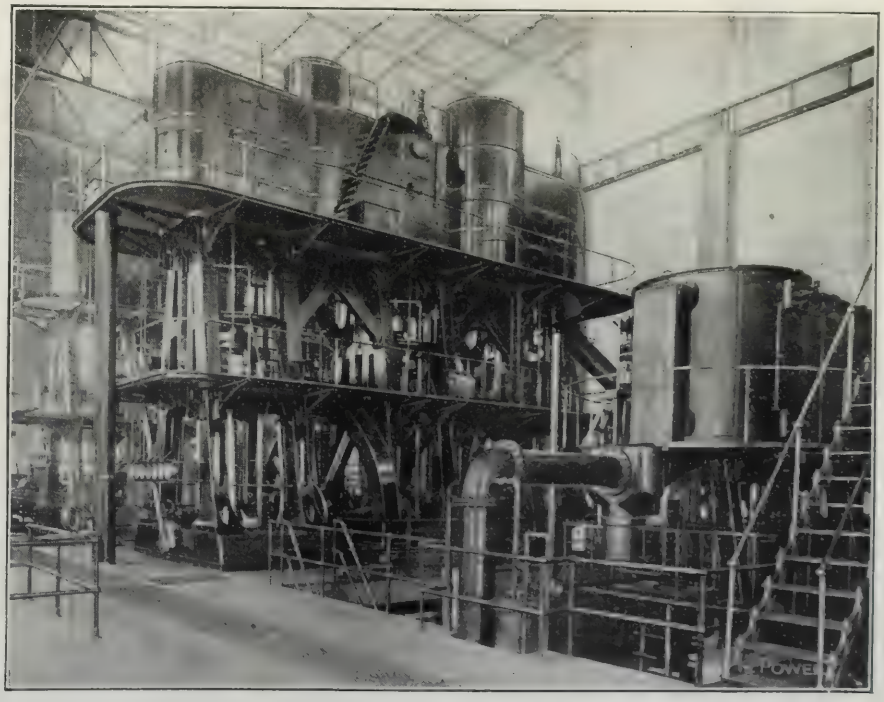


FIG. 8. BARR ENGINE IN FOREGROUND AT RIGHT; TOD ENGINE BEYOND

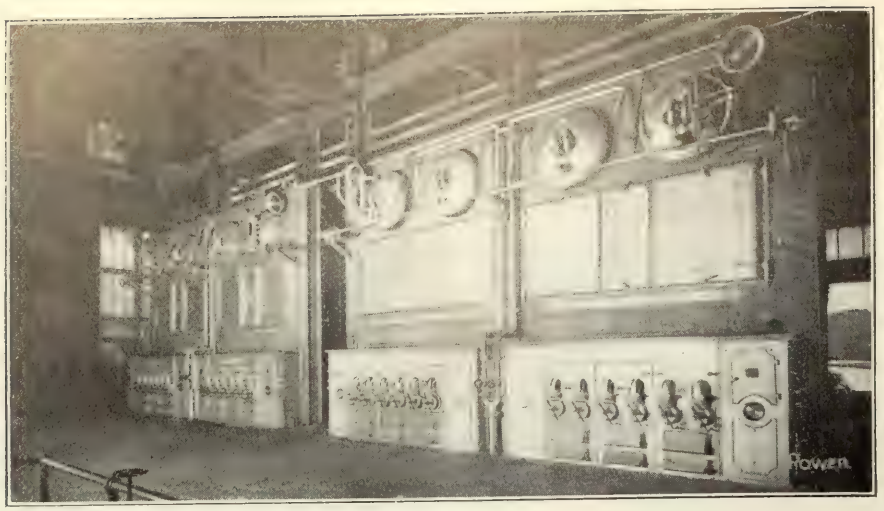


FIG. 10. GROUP OF OIL-BURNING BOILERS AT TURKEY CREEK STATION

use will probably soon be largely discontinued.

There are three boiler-feed pumps, one a Dean Bros. 8x12x8-in. horizontal duplex plunger type; one a Worthington 6x8x6-in. vertical duplex marine type, and one a Barr of the same type and size as the Worthington.

Fuel oil is stored in four steel tanks 200 ft. south of the station, each holding 500 bbl. The oil-pumping equipment is in duplicate and consists of two 4x4x3-in. duplex pumps with a heater attached to each and mounted on the same stand.

Coal is brought alongside of the boiler house on a siding and unloaded into industrial railway cars running into the basement, where they are elevated to the boiler-room floor on hydraulic lifts in front of the boilers. The ashes drop into hoppers under the boilers from which they are emptied into the industrial cars and from these loaded into wagons to be hauled away.

PIPING AND AUXILIARIES

One 8-in. header directly above the boilers extends the length of the room. This is cross-connected to the 10-in. main header supported on the engine-room wall. The 8-in. cross-connections are so arranged that any boiler or any engine may be isolated without interfering with any of the others.

Two small high-speed automatic engines driving 50-kw. direct-current dynamos in one corner of the engine house supply current at 220 volts for light and power in the station and the machine shop adjoining.

Two 50-cu.ft. capacity two-stage steam-driven air compressors force air into the air chambers of the various pumping engines. The air is injected daily.

MACHINE SHOP

Adjoining the boiler house is a well equipped machine shop in which all repairs for the entire plant are made. Also, the street hydrants are finished here, the rough castings being bought from a commercial foundry. By having this latter work to do between repair jobs it is possible to retain the machine-shop force practically intact and consequently maintain it in a well organized condition. The shop equipment comprises four lathes, one 20 ft. long with a 42-in. swing, one 16 ft. long with a 20-in. swing, one 12 ft. long with a 16-in. swing, and one 10 ft. long with a 14-in. swing; one American shaper with 20-in. travel; one Kempsmith No. 3 milling machine; one American radial drill; one American drill press, and one hacksaw. All of the tools are driven by individual motors.

Power Plant of the "Kansas City Star"

Description of a plant devoted exclusively to printing a newspaper. Crude oil is used for fuel and the plant contains three Corliss engine-driven generators aggregating 700 kw.

Details of the electric drive for the presses are given.

The *Kansas City Star,* one of the leading newspapers of the Middle West, has as fine a home as any paper in the United States, and one which is different. After crowding itself out of house and home several times in its 32 years of existence, its owner decided, when the present plant was built in 1909, that it should be made ample to permit considerable expansion in the working force and equipment without causing discomfort or inefficiency.

The building shown in Fig. 1 is the result. This entire structure, 250 ft. wide by over 110 ft. deep and three stories high with a basement and sub-basement, is devoted solely to the business of issuing a newspaper. Many of the features about the arrangement of the building and the equipment for manufacturing and setting the type, making the engravings and casting the cylindrical plates, are extremely interesting, but space cannot be given to their description, since they are not pertinent to POWER'S field.

As can well be imagined, the power-generating equipment is very modern. The boilers are in the sub-basement, 11 ft. below the engine-room floor; two of the three installed may be seen in Fig. 2. The third faces these two, across the firing aisle. Each is of the Bonus-Freeman box-header water-tube type and of 350 hp. There are 155 tubes in each boiler, 4 in. in diameter and 20 ft. long, and two drums 36 in. in diameter. The feature of these boilers is their extremely free circulation due to the large cross-sectional area of the headers and nipples and the high pitch of the tubes, which is 1½ in. to the foot. The large sectional area of the headers also gives them great water-storage capacity and steam-liberating area, which latter tends to produce dry steam.

FIG. 1. HOME OF THE "KANSAS CITY STAR"

Oil Burning System

Oil is burned under the boilers at present, but coal storage facilities have been provided and a change to this fuel can be made easily and quickly. The burners are of the C. Y. Roop internal-mixing pattern. The fuel oil is delivered in wagons to three rectangular storage tanks buried under the sidewalk. Each tank is 16 ft. long by 8 ft. wide at the bottom, 9 ft. at the top and 15 ft. deep, holding 15,500 gal. The oil flows by gravity to the pumps which supply it to the burners. These pumps are in duplicate and are on the same stand with the oil heater which receives the exhaust steam from the pumps. They are Fairbanks-Morse horizontal duplex pumps with 1½ in. suction and 1-in. discharge. The pump speed is controlled by adjusting the throttle by hand. An automatic relief valve on the discharge line, set for 80-lb. pressure, bypasses the oil to the suction when the pump is working too fast. An automatic pump governor was not considered necessary as the fireman must regulate the supply of steam to the burners by hand and can adjust the pump speed at the same time. A Worthington piston-type oil meter in the pump-discharge line registers the quantity of oil consumed.

The feed water passes through a 1000-hp. Hoppes live-steam feed-water heater and purifier to two Burnham 12 by 7½ by 6-in. duplex plunger boiler-feed pumps, made by the Union Steam Pump Co.

The air compressor and the hydraulic elevator pumps are also in the sub-basement. Compressed air is used for forcing the printing ink from storage tanks in the sub-basement to the fountains on the six large octuple printing presses which share the basement with the electric generating equipment. Compressed air is also used to raise the oils for washing the inking rollers and for lubricating the presses and steam engines. The lubricating oil is elevated to a tank from whence it flows to the various bearings. From the latter it flows to filters in the sub-basement and thence to a small receiving tank from which it is returned to the supply tank.

Three Westinghouse motor-driven air compressors, two of 50 cu.ft. and one of 25 cu.ft. per min. capacity, compress air, for the purpose just mentioned, to 80 lb. pressure.

Ten hydraulic lifts handle the rolls of paper in receiving them from the delivery trucks in the street and raising them to the press-room floor from the storage room in the sub-basement. The water for these is pumped at 120 lb. by two Burnham simplex pumps with 10x20-in. water ends and 14 and 22 by 20 in. tandem compound steam ends.

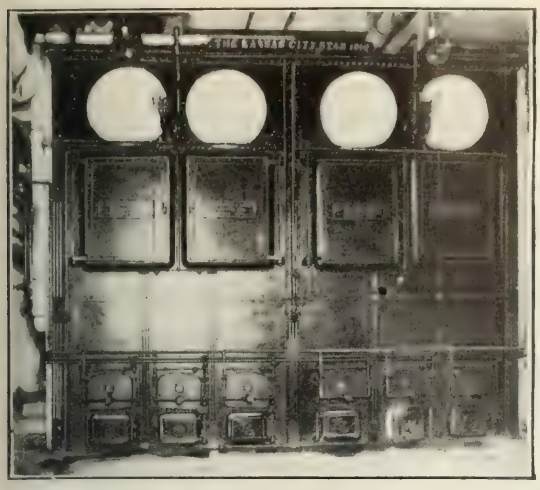


Fig. 2. Two of the Boilers

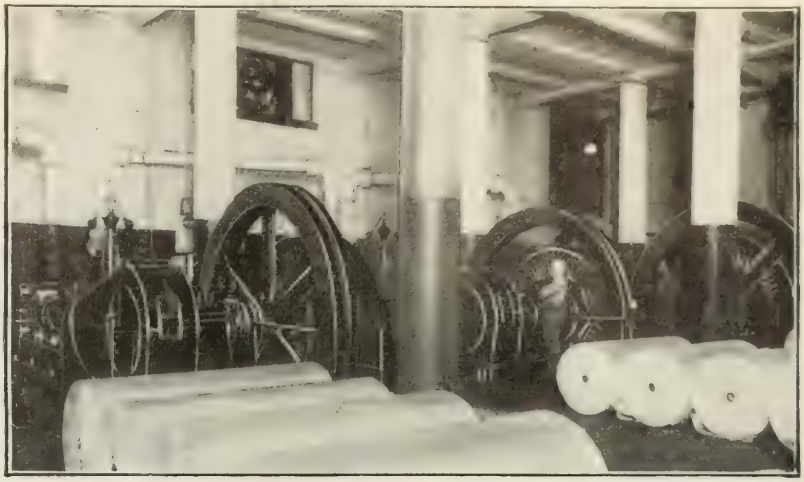


Fig. 3. The Engines and Generators

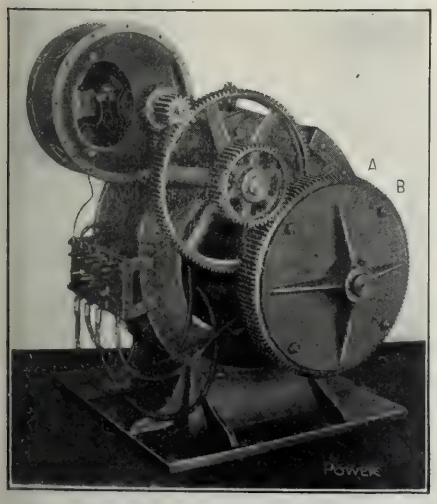


Fig. 4. One of the Press-operating Motors

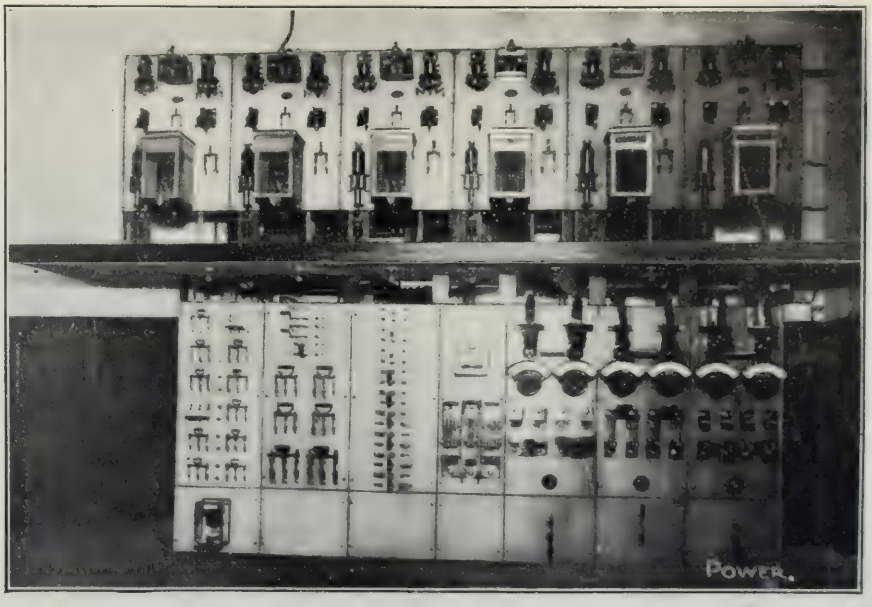


Fig. 5. The Main Switchboard

Genfrating Equipment

Three Westinghouse generators, Fig. 3, supply direct-current at 220 volts. Two are of 200 kw. and the third is of 300 kw. capacity. They are driven by H. N. Strait Manufacturing Co.'s Monarch

Corliss double-eccentric simple engines, running at 95 r.p.m. The two smaller engines are 18x36 in. and the large one is 20x8 in.

There are nearly 100 motors throughout the building of capacities from ¼ to 60 hp. which with the equipment of electric lights, constitute the load for the dynamos.

Electric Drive for Presses

The most important and most interesting motors are those which drive the presses. They have the Jenney system of control and were furnished by the American Rotary Valve Co. The equipment for each press includes two motors, one of 60-hp. for driving the press when printing an edition and one of 7½-hp. for operating at a very slow speed when "making-ready."

The chief reason for using two motors is safety, and a secondary object is economy. When a man is threading the "web" through the press it must run very slow and its speed not increase accidentally or otherwise until he can get clear. Likewise, when he is adjusting a plate, the press cannot be operated until he is finished.

The smaller motor is mounted on the frame of the larger, as shown in Fig. 4. The large motor is coupled directly to the main driving shaft of the press. A special clutch is connected with the small motor through gears as shown. The outer part *A* of the clutch carries the gear teeth which mesh with the gears of the small motor, and rides loosely on the shaft of the large motor. The inner face of the rim which carries these gear teeth has ratchet teeth. The inner portion *B* of the clutch consisting of a disk carrying a set of dogs, is rigidly fixed to the shaft of the large motor.

The dogs are pressed out by springs so as to engage the ratchet teeth of the rim *A* when the large motor is not being operated. Thus, closing the circuit of the small motor causes it to drive the press through the gears and clutch. The dogs are counterweighted so that when the large motor picks up the load and attains a slightly higher speed than that at which the small motor drives the clutch, centrifugal force, acting on the counterweights, throws the dogs clear and the small motor with its gears becomes entirely disconnected.

At various convenient points about the press are several push-button control boxes from which the slow-motion motor may be started or the press may be stopped. Each of these boxes also contains a positive disconnecting switch so that, when one of these switches is opened, the press cannot be started from any other box. This prevents a man being caught while working in an otherwise dangerous position. In addition to these boxes there is one main box from which the running speeds are controlled. This box contains an "increase"- and a "decrease"-speed button which operate a contact drum for cutting the resistance of the main motor in or out. This drum is driven by a small motor through reducing gears, the small motor being controlled by the push-buttons just mentioned.

Main Switchboard

The main switchboard, Fig. 5, is in two tiers. The upper set of panels carries the press-motor controllers, switches, and circuit breakers. Of the lower set, three panels carry the usual equipment of switches, circuit-breakers, and instruments for the dynamos. A fourth panel carries a wattmeter for measuring the total current consumption and the three remaining panels carry the switches for the various light and power circuits. A second wattmeter, on the lower part of the left end panel, measures the current used for lighting; the light and power circuits are separate. Thus, by subtracting the current used for the lights from the reading of the main wattmeter, the motor current consumption can be obtained.

Steam Header

The leads from the boilers collect into a 12-in. main steam header extending along the wall dividing the upper part of the boiler room from the engine room. From this the lines to the engines extend through the wall and connect with the several units as shown in Fig. 3. This header is so fitted with valves that any section can be cut out for repairs, or parts of the engine and boiler equipment can be isolated for testing purposes.

A National Biscuit Company Plant

Brief description of a small plant which possesses ideal working conditions. A scheme for pipe identification by different colors is worked out to good advantage.

The Kansas City plant of the National Biscuit Co. is one of the newest and best equipped of the numerous plants that this company operates. It contains the latest improved machinery for mixing, rolling and cutting the various cracker and cake doughs, for conveying and packing the finished products and for folding, wrapping and pasting the cartons in which the products are packed. As would naturally be expected of a plant devoted to the making of food products, the building is light, well ventilated and scrupulously clean throughout.

These characteristics extend even to the power plant which is contained in the low one-story addition at the right end of the building, as seen in Fig. 1. The power equipment is small, its arrangement is simple, and hence its story is quickly told, but it deserves attention here because it represents almost ideal conditions from the standpoints of both economical operation and the physical comfort and welfare of the working force. To begin with, crude oil is used as fuel, a circumstance contributing to both of the last-mentioned conditions, and to the general cleanliness of the plant as a whole. The wall facing the boilers contains large windows, an ample double door opens at the middle and, as the boiler and engine-room floor level is but a few feet below the street, plenty of light and air is admitted. In the engine room large windows on two sides afford ample light and ventilation there also.

Fig. 1. Kansas City Plant of National Biscuit Co.

A view in the boiler room is given in Fig. 2. Steam is generated in two B. & W. water-tube boilers of 265 hp. each, fitted with the Hammel back-shot oil-burning system. Steam is carried at 110 lb. pressure. Two underground steel tanks outside of the plant store the fuel oil. Their combined capacity is 30,000 gal. The feed water is heated in a Stillwell 500-hp. open heater. The two feed pumps are Wilson-Snyder horizontal duplex plunger pumps, one 5x3½x8-in. and the other 7x4x12-in. A Marsh vertical deep-well pump with a 5x24-in. water end raises water from a well to the ammonia condenser of the refrigerating system. On the heating system is a Wilson-Snyder simplex 8x10x12-in. vacuum-return pump.

In the engine room are two generating units, an air compressor and one engine-driven and two motor-driven exciters. The main unit is an 18x36-in. Allis-Chalmers Corliss engine driving a Westinghouse

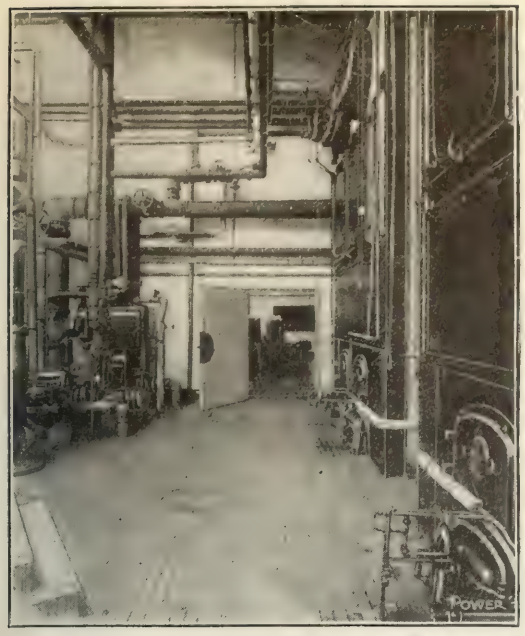


FIG. 2. THE BOILERS, PUMPS AND HEATER

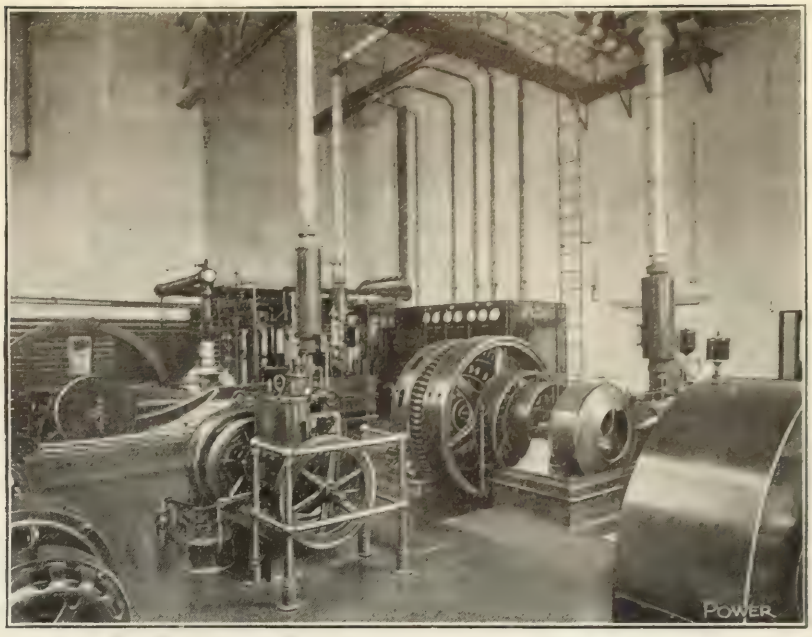


FIG. 3. PART OF THE GENERATING EQUIPMENT AND THE ICE MACHINE

200-kw. alternating-current generator. The current is generated at 220 volts and is two-phase, 60-cycle. This unit runs at 121 r.p.m. The other main generating unit is a Buckeye piston-valve engine, running at 257 r.p.m., driving a 100-kw. Westinghouse dynamo. Throughout the building are 66 motors, ranging from ½ to 15 hp. A small two-cylinder simple-acting vertical Westinghouse high-speed engine, running at 400 r.p.m., drives a 16-kw. exciter used when starting up. After the plant is started, excitation is furnished by either of the two motor-driven exciters, one of which is of 10 kw. and the other 15 kw. capacity.

The ammonia compressor, of 20 tons capacity, is a Linde, driven by an Ohio Corliss 12x30-in. engine at a maximum speed of 65 r.p.m. Cold brine is piped through the building to various cold boxes used for storing eggs, milk, butter, etc., by two Laidlaw-Dunn-Gordon 4½x6x6-in. duplex pumps.

The sponge, or proving room, where the biscuits are placed to raise, is maintained at or very close to 80 deg. F. all the time. On some days during the summer this requires artificially cooled air. To get as close regulation as possible, air is circulated through either a heating or a cooling chamber, as required, and delivered into the room by a Sturtevant ventilating fan. A thermostat in the room automatically controls the fan and the dampers.

PIPING

From each boiler an 8-in. lead connects into a short 12-in. header which extends through the engine-room wall and then branches each way, as shown in Fig. 3. A short separate auxiliary header supplies steam to the pumps and other auxiliaries.

Ready identification of the different piping throughout the plant is afforded by painting each system a distinguishing color: white, live or high-pressure steam; brown, exhaust or low-pressure steam; green, hot water; red, fire system; orange, cold-water service; yellow, gas, and black, sewer. By this means it is possible to tell instantly what a given pipe contains and thus mistakes may be avoided and time saved in tracing out the various lines.

Over the Spillway

Out in Beach City, Ohio, a fond mother bought her son a power plant as a "place to show his skill and keep busy." Friend son was recently graduated from an engineering school. Of course, he'll keep busy. And if your skill short-circuits, sonny, blame it on the plant, and perhaps mother will buy you a nice new one.

Hex Nutt, employed in the Holdfast (Tex.) power plant, who has been suffering from a severe wrench, took a bad turn on Tuesday.

B. T. Unit, night engineer at the Holy Terror (N. D.) pumping station, has added another Unit to his battery of five. It's a boy. "Brit" says the youngster will probably be named after the town of his birth.

A contemporary apologizes for a typographical error in its article on transmitting capacities of pulleys. As pulleys appeared as "pullets," the printer evidently thought the publisher was advancing a new theory of eugenics for raising chickens.

Poor Draft! Everybody seems to mourn his loss. Draft had a wide acquaintance among operating engineers and was a near connection of the well known Stack and Furnace families of Plantville.

Ig. Nition is back firing at the gas plant. Ig. is some het up over the rumor that his absence was due to carrying an improper mixture.

An Oregon civil engineer is real peeved because the word "engineer" confounds him with the operating engineer, whom he terms the "monkey-wrench man," the "starter and stopper." He would just dote on being called an "energier"—just like that. Perhaps if the operating engineer changed his designation to "monkey-wrencher," "stopperette" or "starterino," the more or less civil engineer would be content.

Hard Coale, fireman at the Asheville (N. C.) Soap Works plant, has been discharged for paying more attention to his "suds" than his scoop.

Carrie Sene has put the kibosh on young Brush's sparking and chattering. We are indebted to Carrie for cleaning up several little difficulties in our midst.

T. Crane has a fine position traveling back and forth for Lift & Haulem.

Hen Dill, engineer of the coal barge "Pickel," is dead set on finding the party who put the missis in the Mississippi. Hen's missis, Daffie Dill, being considerable hefty, is afraid the government is going to hold her liable for bursting the levee near where she went in.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Inspecting and Testing Electrical Apparatus—III

By A. L. Cook

Inspecting Switchboard Instruments

The switchboard instruments usually furnished with an alternating-current generator, are one or more ammeters, a voltmeter, wattmeter and sometimes a power-factor meter. The readings of these instruments should be compared to detect any serious inaccuracies. If there are two voltmeters, they may be compared when connected on the same circuit, or checked with a portable meter known to be correct. The generator ammeters can be checked with the sum of the feeder-ammeter readings on any one phase. This should equal the reading of the generator ammeter if only one machine is in operation, but with two or more generators in parallel, there will be cross-currents between the machines and the generator ammeters will read higher. Generator wattmeters usually indicate the total output of the machine in kilowatts. Sometimes two single-phase wattmeters are used for a polyphase generator, in which case the sum of the readings would, in general, indicate the total output; the only exception is with three-phase generators where the power factor is less than 0.50, in which case the smaller reading must be subtracted from the larger. The wattmeter can be checked approximately by assuming a power factor; that is, the ratio of actual watts to volts × amperes. For lighting service only, this would be about 0.90 and where a large number of induction motors are used, about 0.80. For a single-phase generator the kilowatt output would be:

Kilowatts =

$$\frac{Volts \times Amperes \times Power\text{-}factor}{1{,}000}$$

For a three-phase generator:

Kilowatts =

$$\frac{Volts \times Amperes \times Power\text{-}factor \times \sqrt{3}}{1{,}000}$$

For a two-phase generator:

Kilowatts =

$$\frac{Volts \times Amperes \times Power\text{-}factor \times 2}{1{,}000}$$

In each case the voltage is that of any phase and the amperes is the current in any lead, assuming a balanced system. The last equation applies to a four-wire two-phase system. For a three-wire two-phase system, the current to be used in the equation is that in either outside wire, and the voltage is that between either outside wire and the common wire.

The power-factor meter can be compared with the wattmeter by substituting its readings in the foregoing equations and solving for the kilowatts, which should equal the readings of the wattmeter.

Watt-hour meters are generally furnished either for the generators or the feeders to record the total output in kilowatt-hours. These meters should be checked to see that the front edge of the disk rotates in the proper direction. Sometimes it will be found that the reading of a polyphase wattmeter or the speed of a polyphase watt-hour meter, when connected to a generator will change when the field rheostat is adjusted with no change in load on the generator. This is due to the potential and current coils being connected to the wrong phases.

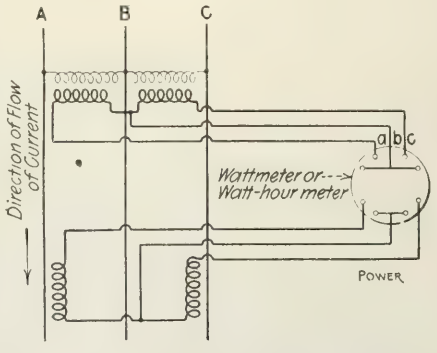


Fig. 8. Wattmeter Connections for Three-phase System

These meters really consist of two single-phase wattmeter elements assembled on the same shaft. Before checking the wattmeter connections it is necessary to determine the voltage and current terminals corresponding to each set of elements. Usually this may be done by inspection, but if this is not possible, a simple test will determine the proper connections. With both current coils connected and load on the meter, one pair of voltage terminals should be disconnected. The meter should deflect or the disk rotate. First one pair of current terminals and then the other should be short-circuited. The pair which when short-circuited causes the pointer to return to zero or the rotation to cease is connected to the element corresponding to the voltage coil in service, and the terminals should be marked accordingly.

Having determined this, the connection to the current and potential transformers should be checked; Fig. 8 gives connections for a polyphase meter in a three-phase circuit and Fig. 9, similar connections for a two-phase circuit. The top pair of meter terminals on each side are connected to the voltage coils and the lower pairs to the current coils, the corresponding sets being on the right and left respectively. In Fig. 8 the current terminals of one element are connected in lead *A* while the corresponding voltage terminals are connected between this lead and that which has no current transformer. In Fig. 9, the voltage and current terminals of one element are connected in phase A_1A_2 and the other in phase B_1B_2. Generally the connections between the instrument transformers and the meter can be traced, as they should have been tagged when installed, but if for any reason it is desired to make a test to insure that the corresponding voltage and current terminals are connected to the transformers on the same phase this may be done as follows:

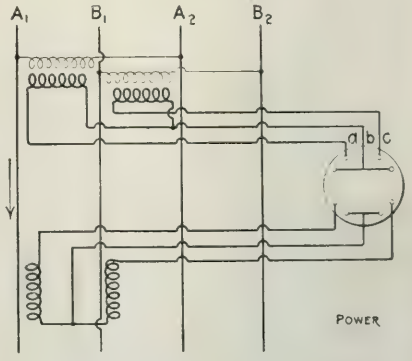


Fig. 9. Wattmeter Connections for Two-phase Four-wire System

The common wires for both the voltage and the current transformers should first be traced out and connected to the proper terminals of the meter; then the other wires may be connected to the remaining terminals. The meter should now operate if there is load on the circuit. One of the outside voltage leads, such as *a*, Fig. 8, should be opened at the transformer, and the current transformer in phase *A* short-circuited. If the meter reads zero it is an indication that the current and voltage leads of phase *A* are connected to the same element, which is correct. If the meter does not return to zero the connections are wrong and volt-

age leads *a* and *c* should be interchanged at the meter.

Having connected corresponding voltage and current leads to the same meter element, it is necessary to check the polarity. With load on the machine, one of the outside voltage leads (for example *a*) should be removed from the meter. The instrument should deflect properly or the rotation should be in a forward direction. If the wrong deflection is indicated, the two voltage leads connected to the element in service should be reversed. Lead *a* should then be replaced and lead *c* disconnected. The meter should still deflect correctly and if not the voltage leads in service should be interchanged. Generally it will be found that the deflection when one side of the meter is disconnected is different from that when the other side is out of service. The readings would only be alike when the power factor is 100 per cent. and this very seldom occurs in practice. At 50 per cent. power factor, one side will not deflect at all and for lower power factors this side will give a reversed reading; hence the foregoing rule for checking polarity applies only to a power factor greater than 50 per cent. Since most systems have power factors between 70 and 90 per cent., however, this rule is generally correct. In dealing with current transformers, the secondary circuit should never be opened when current is flowing in the main leads as a very high and sometimes dangerous potential in the instrument circuit may result, and if the circuit is left open for a considerable length of time the transformer will become overheated and probably burn out.

Most modern alternating-current plants are equipped with a Tirril voltage regulator which automatically maintains normal voltage on the busbars under all conditions of load by varying the exciter voltage. This regulator is carefully adjusted at the factory and usually requires very little attention to put in service. The manufacturers always furnish a diagram of connections together with complete instructions for installing and operating and it would be outside the scope of this article to give such instructions. While the instruction book gives complete directions for adjusting the various parts, it is seldom necessary to do this as each regulator is carefully adjusted at the factory. The adjustment is a rather delicate operation and should not be attempted unless one is familiar with the apparatus. Certain tests should be made, however, before the regulator is put in service, and these will be outlined briefly.

Tests should first be made to determine the proper setting for the exciter field rheostat and it can then be sectionalized and connected to the relay contacts on the regulator. After the regulator has been connected up, cleaned, and the moving parts inspected to see that they move freely, the dashpot may be filled with oil and the counterweight adjusted to balance with normal alternating-current voltage on the regulator. During this operation, the switches connecting the regulator with the exciter field rheostat should be open and the regulator lever balanced until the main contacts just close at a voltage about 2 per cent. less than normal. The generator field rheostat is generally all cut out when operating with a voltage regulator, unless the voltage across the field of the generator is less than 70 volts with no load. A test should be made to determine this, and if necessary, sufficient resistance should be left in the generator field rheostat so as to give 70 volts under these conditions. After this has been determined, the regulator can be put in service according to the instructions. The alternating-current voltage should then be adjusted until it is exactly normal.

Ordinarily the regulator is used to maintain a constant potential at the busbars for all loads, but if it is compounded, this adjustment should be checked and corrected if necessary. When the generators are operating at not more than full load, the regulator relay contacts should vibrate rapidly, the rate being highest at light loads. The regulator should need no further adjustment until the contacts become worn.

Transmission Notes

General practice in transmission systems is discussed to some length in a report presented at the recent convention of the Ohio Electric Light Association. It says, "The practical working radius of 2300-volt transmission is very limited and under ordinary conditions is about one mile for 25 cycles and ¾ mile for 60 cycles when the wire is loaded to its full current carrying capacity, with a 10 per cent. drop in voltage." For higher voltages the report recommends, as a good rule, 1000 volts per mile of line.

Regarding the spacing of conductors it specifies the following:

	Spacing, Inches	
Volts	Minimum	Maximum
6600	12	18
13200	18	36
22000	30	36
33000	32	72

The arrangement of conductors is either triangular or all in one plane for three-phase distribution, the former being preferable, owing to the balanced inductive effect.

As to the choice between aluminum and copper conductors the report says: "For equal transmission conditions, the cost of an aluminum line is about equal to that of copper when the aluminum sells at about 2.1 times the price of copper. Aluminum, however, must be handled more carefully and it has greater expansion and more sag, which in turn requires greater spacing between wires, and this raises the reactance of the line."

In discussing lightning protection it says:

"Lightning protection is at all times necessary. The more ways in which the lightning is taken care of, the greater is the possibility of uninterrupted service. Of the numerous arresters the two most prominent are the multigap and the aluminum cell. The most effective mechanical means of preventing lightning disturbance is the overhead ground wire. The aluminum cell arrester has proved very efficient, as it not only affords a path to ground for lightning but aids in removing all internal line disturbances. It is used for excessive voltages and is adaptable for almost any voltage. However, this arrester has a disadvantage in that it is necessary to charge it each day; in this way the efficiency of the arrester is dependent upon the station operator. The multigap arrester also has the feature of relieving surges and is found very efficient for voltages up to 13,200. The ground wire is best placed at the top of the structure so that it has a shade angle of 45 deg. to the outside conductors. It is best constructed of ⅜-in. stranded galvanized plow steel or bimetallic wire. The ideal construction for overcoming the dangerous effects of lightning is the placing of the ground wires at the highest point of the poles and also placing aluminum cell arresters at both ends of the line and in the center."

In discussing this phase of the report, Prof. F. C. Caldwell, of the Ohio State University, made some interesting observations. He attributed the failure of lightning arresters to prevent discharges from the line in their immediate vicinity, to the presence of nodes or points of no vibration in the line. To illustrate this point he drew comparison with the vibration of a string of a musical instrument; the string may vibrate as a whole, the greatest vibration occurring in the middle, or it may vibrate in two parts, in which case there is a point of no vibration at the middle. In this case the string may be held at the middle without affecting the vibration. Similarly, if the lightning arrester is placed at one of the nodes on the transmission line there may be no tendency for a discharge through it, although high potentials and discharges may develop at a comparatively short distance from it.

The one difficulty, as pointed out by Prof. Caldwell, is in determining the probable location of these nodes so that the arresters may be placed accordingly. However, if inductance is inserted in the line by the introduction of a solenoid, and several gaps are tapped along the coil, the probability of all the gaps coming at nodes will be very small, especially as each turn would correspond to a considerable length of line in its effect on the wave of lightning discharge.

A 2400-Volt Direct-Current System

According to recent advices, a contract has just been closed for the electrification of the Butte, Anaconda & Pacific Ry. with a 2400-volt direct-current system. Power will be supplied by the Great Falls Power Co. over a three-phase, 60-cycle, 50,000-volt transmission line to two substations, one at Butte and the other at Anaconda, 26 miles apart. At the substations the voltage is to be stepped down to 2300 volts at which potential the current will be supplied to synchronous motor-generator sets, each consisting of two 1200-volt compound-wound commutating-pole direct-current generators and one 2300-volt, three-phase, synchronous motor. The generators will be arranged to operate in series to supply 2400 volts to the trolley wire.

The entire equipment is to be furnished by the General Electric Co.

CORRESPONDENCE

Pilot Lamps

In the June 25 issue, Mr. Chamberlain suggests that, to avoid absolute darkness in an engine room when the machine breakers blow on the generators at night, one or two of the lamps on the switchboard be connected directly to the generator leads and inside of the breakers. With this arrangement, so long as the generators can supply current at even a low voltage, there will be light.

While such an arrangement is a great improvement over the plan of having no light in such an emergency, it would be still better to be able to have light even if the generator itself should fail, or, where all of the current is obtained from outside sources, as in a rotary converter substation, if the feeder from the power station should break down.

The only ways to do this are to have a second source of current, such as a storage battery, with a breakdown connection, or to keep a lantern lighted after dark. It may be objected that oil lanterns around an electric plant are out-of-date, but they will be found always ready for use in some of the largest power stations. They are always turned low so that they require little oil or attention, but can always be found and used in an instant.

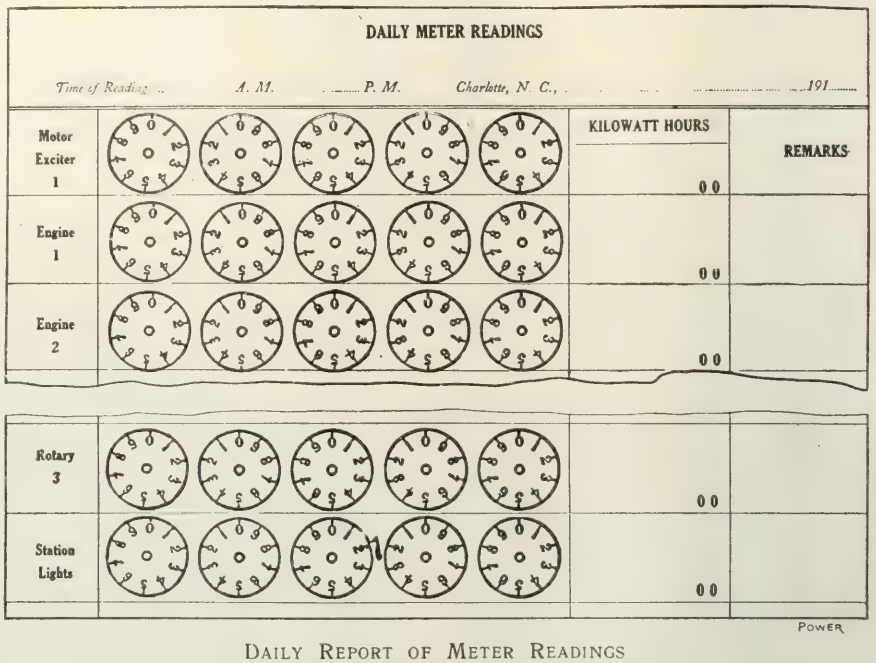
If the pilot lamps are used, they should not be "tied in solid" to the generator leads, but a switch and fuse should always be between the lamps and the source of power. Then, in case of trouble on the wiring to these lamps, they will be cleared by the blowing of the fuse, and a short-circuit will not mean losing all the wiring. Without the fuse, there is also danger of fire, if the circuit should become crossed; an installation of the wiring without it would not be approved by the fire underwriters.

G. H. McKelway.

Brooklyn, N. Y.

Correct Meter Readings

After experiencing considerable trouble in obtaining correct meter readings, due to errors in taking them and also in transferring them to the log sheets, I had the accompanying form printed. The operators now read each meter by marking the proper dials and the sheets are bound into the daily report.


DAILY METER READINGS

Time of Reading .. A. M. ..... P. M. Charlotte, N. C., ... 191.....

		KILOWATT HOURS	REMARKS
Motor Exciter 1		00	
Engine 1		00	
Engine 2		00	
Rotary 3		00	
Station Lights		00	

POWER

Daily Report of Meter Readings

J. B. Linker.

Charlotte, N. C.

Heating of Storage Batteries

I read with interest that part of the "Catechism of Electricity" dealing with storage batteries, in the issue of June 4. Their ventilation, the lighting of the room, and also the location of the room were discussed, but I failed to find anything on cooling the battery while charging.

I have had considerable experience with storage batteries, and one of the greatest difficulties has been to overcome the excessive heating while charging. This was finally accomplished, however, in the following way: I made a wooden tank large enough so that the battery (in hard rubber jars) could be set into the tank and a stream of running water be allowed to circulate around the jars, and run off at the opposite end of the tank, the surface of the water coming to within 1 in. of the top of the jars.

I used this system of cooling through the hot weather last summer, and am using it now. Although the battery has been charged at an exceedingly high rate, I have never known the temperature to go higher than 80 deg. F.

F. C. Heather.

Sterling, Colo.

Sparking Brushes

Carbon brushes are generally rather hard, and when in contact with copper commutator segments, cutting or grinding is likely to occur if close attention is not given. If too tight they will cut down the commutator and soon wear it out; if too loose sparking will result.

I had trouble of this kind and to get satisfactory service had to turn down the commutator about once a week. New armature segments were put in but did not help much and soon these were nearly gone.

After trying several suggestions without success, I finally turned down the commutator to a smooth finish and cut the brushes so that the holders fitted close to the commutator without too much tension in the springs. On account of the segments being narrower after being worn down it was necessary to have the brushes thinner, so I filed each brush off $\frac{1}{32}$ in. on both sides for about $\frac{1}{2}$ in. back. After bringing each brush to true alignment perfect commutation resulted and no trouble was experienced in maintaining the voltage. Each morning after shutting down I take off the brushes and leave them in kerosene and graphite all day. This seems to make them softer. If the brush is very long it is likely to wear sideways or get out of alignment axially. If short it is held more securely in place with less tension in the spring.

William E. Miller.

Jackson, Tenn.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gas Engine Valve Setting

By Olaf Olafsen

Were the true Otto cycle possible the following would be the order of events: First stroke—from inner dead center to outer dead center, suction stroke. Second stroke—from the outer dead center to inner dead center, returning, compression stroke. Third stroke—from inner dead center to outer dead center, power stroke. Fourth stroke—from outer dead center to inner dead center, returning, exhaust stroke. The cycle is then completed.

First consider the exhaust. To allow the pressure to fall to atmospheric before the piston starts on the return stroke and to insure sufficient valve opening to avoid back pressure, it is necessary that the exhaust valve open before the end of the power stroke. In most engines the release takes place about 45 deg. before the outer center is reached or 45 deg. "early." Any exhaust may start 30 deg. early, however, without losing power. Fig. 1 is a diagram from an engine of well known make with release taking place 60 deg. early. Fig. 2 is a diagram taken from the same engine with an indicator having a light spring and a stop to avoid breaking the pencil motion. The scale in this case was 6 lb. per inch as against 240 lb. with the first diagram. Fig. 3 shows a light spring diagram from an oil engine with the exhaust valve opening 45 deg. early. In each case the fall of pressure to atmospheric is completed very nearly at the end of the stroke, which is proper, showing no loss of available expansion and no back pressure.

Certain four-stroke-cycle engines have an exhaust port through the cylinder wall, which is uncovered by the piston at the end of the stroke. This leaves a comparatively small volume to pass the poppet valve, relieving the valve-gear of the strain of opening under pressure and adding to the life of the valve and seat. In these engines it is customary to open the valve on the outer dead center to let the fall of pressure take place through the side port. These side ports are uncovered at about 30 deg. early by one builder. The time of opening will depend upon the speed of the engine, the area of port and the relation of width to length of port just as the valve timing will be determined by the size and lift of the valve, size of ports and layout of lifting cam. The valve lift must be nearly proportional to the piston velocity at various parts of the stroke, or the outlet may be ample at mid-stroke and restricted at the ends even though the valves, ports and valve lifts are correct. So much is to be found in the better works on engine design concerning cam layouts, port areas and valve sizes that these details will here be omitted.

Where the exhaust piping provides free exhaust but is not too short, the burnt gases, upon being released, rush out through the pipe with high velocity, due

FIG. 1

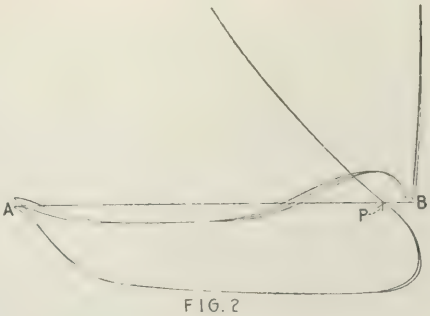


FIG. 2

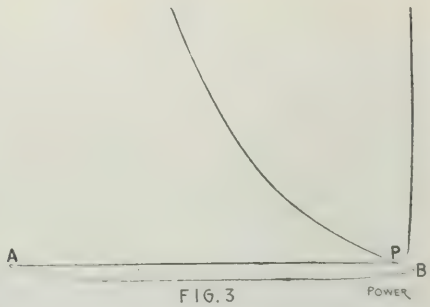


FIG. 3

Upper Diagram Taken with Heavy Spring, and Lower Diagrams with Light Spring

to momentum, and continue to travel even after the pressure in the cylinder has been relieved. This plug of exhaust gas has an effect upon the cylinder like a long piston drawn through the exhaust pipe. This is plainly shown in Fig. 3 where the exhaust line falls below the atmospheric line. It did not take place in Fig. 2 because two three-cylinder engines of large size were exhausting into the same main at the time the diagram was taken. Advantage is taken of this phenomenon as follows: The exhaust valve is not closed on the inner dead center but is held open for 10 or 15 deg. late. The inlet is opened on the same cylinder 10 or 15 deg. early or before the dead center is reached. There is then a period of 20 or 30 deg. during which the exhaust gases draw fresh air through the clearance, more or less completely scavenging it, depending on the position of the valves and the perfection of the action. On engines having timed gas admission, it is safe to allow an overlap of 30 deg., as no loss of fuel can take place, and 20 deg. lap may be used without waste of mixture on engines employing some form of simultaneous admission of gas to air unless both valves should be located in one small pocket on one side of the cylinder.

Certain conditions arise which sometimes make it impossible to make use of the foregoing. Automobile motors are usually muffled and under some back pressure; also, multicylinder engines having more than three cylinders exhausting into one pipe must close the exhaust before the inlet is opened to avoid the exhaust blowing back from one cylinder into another. In spite of the danger it will be found advantageous on high-speed engines to hold the exhaust open a short time after passing center and open the inlet still later to insure as much reduction of pressure in the cylinder as possible.

Three- or six-cylinder stationary engines with two or more exhaust pipes make it possible to utilize such action very nicely. Three exhaust periods must be completed in two revolutions giving 240 deg. for each exhaust. Taking 180 deg. for the theoretical exhaust stroke and 10 deg. for overtravel, there remains 50 deg. for lead, and the few degrees of pause between exhaust from the different cylinders to insure the closure of one exhaust port before opening the other. Moreover, where the gas is rich in hydrogen and easily inflamed or where the gases are very slow burning and linger in pockets serious backfiring may result if the mixture is drawn into the clearance before the exhaust is complete.

Where the valves are so placed as to insure scavenging and where the fuel or other conditions do not introduce difficulties a decided gain in power results from proper application of the foregoing.

On heavy engines having valves operated by wiper cams and eccentrics it is found advisable to sacrifice some rapid-

ity of opening and closing in order to overcome the inertia and hammering of the valves. In this case it may be necessary to open earlier and close later than usual. These motions give very rapid travel during mid-stroke, but raise and seat the valves very gently. It is usually possible to employ a timing with lap of valve openings on these large tandem double-acting engines even when there are four working ends, as the length of pipe between each valve and the underground main is considerable. Twin tandem engines usually have two mains.

With present piston speeds it is impossible or at least impracticable to make the valves and ports of such size that the gases will be able to keep the cylinder filled during the suction stroke at atmospheric pressure; hence at the end of the theoretical suction stroke the pressure in the cylinder is somewhat below atmospheric. However, as the gases are traveling through the ports at high velocity the combined effect of their inertia and the partial vacuum in the cylinder will allow the filling process to continue even after the piston starts on its return stroke.

In stationary engines of good design the inlet may be held open for 20 to 30 deg. beyond dead center without loss of charge. On old engines where the cams are improperly laid out or the valves and ports inadequate they may be held open as late as 60 deg. without loss, although it is true that such an engine will fall short in power as full cylinder displacement is not effective.

In Fig. 1 the suction drop does not show as the spring is too stiff to indicate much movement under such low pressures, but Fig. 2 from the same engine with a light spring and stop illustrates this very well. These diagrams are from an old engine with automatic inlet valves. The suction is expressive, although it is desirable to allow more drop to take place with automatic inlets than with mechanically operated valves to insure prompt opening and closing of the valves.

Fig. 3 shows a very good suction line and was taken from a larger engine with cam-operated inlet valves opening 10 deg. early and closing 30 deg. late.

Referring to *P* where the compression line crosses the atmospheric line on Figs. 2 and 3, it is obvious that the ratio of *AP* to *AB* will be an index to the volumetric efficiency. In taking such diagrams care must be exercised in handling the indicator, the piston and cylinder of which must be free from grit and be frequently oiled. There must be no appreciable lost motion in the pencil motion or the diagrams will be useless.

The foregoing theories will hold true with the addition of some thought and common sense applied to individual cases by the man on the job. Above all, when the valve travel is found to be too short in period, take up all lost motion in the linkage except what little clearance is necessary to insure closure of the valve, thus getting all the available travel, and then shift the gears so that the exhaust does not close before the dead center but do not try to get too much overtravel at the expense of the release. Likewise the inlet should not close before the outer dead center is reached, even at the expense of opening a trifle late. On high-speed engines the inlet should have some overtravel (late closure) if the valve motion will permit. In any case the closure is the important point when conditions cannot be altered to give the desired valve setting. Unless one is sure of his ground, it is preferable to make the best of what settings can be had from a well adjusted valve-gear in good repair than to attempt changes in linkage, cams or rollers.

Builders often furnish typical diagrams, but it is not always possible to duplicate these, because of backlash of the gears, slight differences in or wear of cams, collers or links. However, it is usually possible to comply with the sample diagram within 3 deg. on either side at closure and let the openings come as nearly right as they may. When the valve-gear is operated by a train of spur and bevel gears and shifting one tooth of one gear advances or delays the events too much, if there are pairs of gears of different pitches or different numbers of teeth on any of the connecting shafts, by shifting one pair of coarse pitch or large size ahead one tooth and a pair of finer pitch or smaller size back one tooth at the same time, or *vice versa,* the desired setting may be obtained.

On engines that have separate cam-operated inlet valves, such as the Otto (Columbia type), Fox, Callahan, White and Middleton, the inlet should not open until after passing dead center (not earlier than 10 deg. late, nor later than 30 deg. after center), when the exhaust is closed to avoid backfiring and loss of fuel. These should close on center with automatic intakes. On small engines they often close as late as 30 deg. after or as early as 30 deg. before dead center. The later setting seems preferable, but the charge seems to mix on compression stroke and no bad results occur with the early closure. In fact, one of the most economical small city gas engines on the market today by actual test closes the fuel inlet 30 deg. early while another of equal rank closes it 28 deg. late.

On suction-gas engines where the engine must draw its fuel through the produced bed and scrubber, the fuel valve if separate, may be opened simultaneously with the inlet valve in order to start the gas toward the engine in good time. This applies particularly to single-cylinder hit-and-miss engines, a type now seldom used with producers in this country.

Fuel Economy of Gasoline Engines

By A. A. Potter

Gasoline-engine manufacturers usually claim that their engines will develop 1 hp. for 10 hr. on a gallon of gasoline. Believing that few operators are able to realize this economy on small engines tests were conducted at the Kansas State Agricultural College under the direction of the arthur on five gasoline engines of different makes, and varying in power from 1¾ to 10 hp., to determine the actual fuel economy per brake horsepower-hour under ordinary working conditions.

The engines tested were all of the four-stroke-cycle type, water cooled, governed by the hit-and-miss method and provided with a make-and-break ignition system. They were loaded by Prony brakes and the fuel was accurately weighed. Three to six check runs were made at each load, each lasting 30 min.

The gasoline used was 62 deg. Bé. and had a specific gravity of 0.731 at 60 deg. F., making the weight 6.09 lb. per gal. The heat of combustion of the gasoline, as determined by a Junkers constant-pressure calorimeter was 19,411 B.t.u. per lb. (high), or 18,415 B.t.u. per lb. net.

FUEL ECONOMY OF GASOLENE ENGINES

Cylinder Diameter, inches	Stroke, inches	R.P.M.	Rated B.HP.	Gasolene Consumption per B.h.p.-hr.							
				Full Load		¾ Load		½ Load		¼ Load	
				Lb.	Gal.	Lb.	Gal.	Lb.	Gal.	Lb.	Gal.
3.5	5.25	500	1.75	1.030	0.169			1.35	0.222	1.99	0.327
5.0	8.0	350	4.0	0.915	0.150	1.03	0.169	1.24	0.204	2.08	0.342
5.5	9.0	325	6.0	0.834	0.137	0.81	0.133	0.99	0.163	1.47	0.242
6.5	9.5	300	6.0	0.855	0.140	1.06	0.174	1.14	0.187	1.61	0.264
7.25	14.0	250	10.	0.791	0.13			1.46	0.24		

The average results of the tests are given in the table, from which it is evident that 1.3 to 1.69 gal. of gasoline will be required to run the average small engine 10 hr. at full load.

Tests by C. E. Lucke and C. M. Woodword (see Bulletin No. 191, U. S. Department of Agriculture) show an average of 0.25 gal. per b.hp.-hr. for a 2-hp. marine engine and 0.12 gal. for an 18-hp. engine.

Commercial Utilization of Peat for Power*

By H. V. Pegg

Utilizing peat commercially would seem to require as a first consideration using it, as nearly as possible in the condition in which it leaves the boglands without preliminary and expensive machine treatment.

About seven years ago the author experimented with air-dried hand-cut peat fired into a special form of gas producer. With all gas producers using bituminous coal the main trouble is to get rid of the tarry byproduct. In this instance the producer was arranged to work intermittently, with periods of "blowing" during which the fuel was brought to incandescence and periods of gas making during which the tarry byproducts were passed through the incandescent fuel and converted into gas. The chief difficulty with this plant was the high thermal value of the gas generated—about 330 B.t.u. Owing to the high and varying percentage of hydrogen, it proved unsuitable for the gas engine. Although the plant ran more or less continuously for ten days, considerable trouble was experienced, owing to the varying moisture content of the peat.

From this experience it seemed wiser to extract the tar, rather than try to convert it into gas; furthermore, the producer must not be sensitive to the moisture in the peat. Later various tests were run with an experimental plant at the works of Messrs. Crossley Bros. Portadown, and as a result of these, a special plant was built from their designs and patents and to the author's specification. This plant has been running since last September and has been described in the daily and technical press;* hence there is no need to dwell upon its details. However, a few of the difficulties experienced with it will be mentioned.

Air-dried peat is an inconvenient fuel to fire into a producer; therefore originally the peat was reduced in size to blocks of about 5-in. cube; but later it was found possible to dispense with this preliminary treatment, and the construction of the plant was thereby simplified.

Last October the plant was subjected to a test run of 6 hr. with a load of 250 b.hp., the peat consumption per b.hp.-hr. averaging 2.55 lb. and the peat fuel containing 18.98 per cent moisture; this was with both producers running, although the load was considerably below the total capacity of the plant. The low percentage of moisture was due to the unusually dry summer of 1911. During November and December, however, the fuel was extremely wet, but the plant worked just as well with the sodden peat as with the drier peat, the only difference being the amount consumed. The amount of water in this "sodden peat" varied considerably, being sometimes as high as 70 per cent.

The separation of the tar from the gas was the chief difficulty, and it was found far better to rely on an ample water-spray through which the gas passed than upon any form of coke scrubber. The large part of the tar was thrown out into a tar sump by a centrifugal tar extractor; but unless the gases were subjected to a thorough washing and cooling by the water-spray, some tar got past the extractor, collected in the gas mains and finally reached the engines.

When first started, the plant in general, and especially the producers, required thorough cleaning once a week, but at present it can be run if necessary for three weeks without cleaning, although cleaned weekly as a matter of precaution. This is owing to the increased washing water used, which is now about 7 gal. per b.hp.-hr. The tar recovered is about 5 per cent. of the weight of peat consumed.

Experiments were made in burning the tar under boilers, but owing to the high percentage of water present and the large quantity of solid matter, considerable preliminary treatment was necessary. For a time the tar at Portadown was mixed with coal and burned under a Stirling boiler, but now the whole factory is run entirely on peat fuel, the consumption being about 44 tons per week, of which the producer plant takes about 22 tons.

The plant is of 400 hp. capacity and up to June of this year the use of peat has shown an average weekly saving of about $80 over periods of similar operation when using coal.

*Excerpt from paper read at the Belfast meeting of the Institution of Mechanical Engineers, July, 1912.

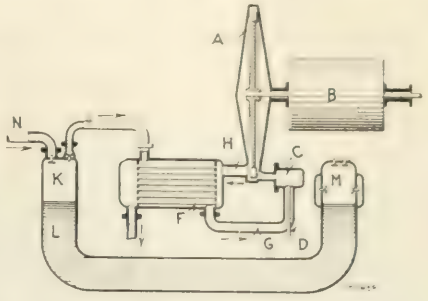
*See Power, Jan. 23 and May 7, 1912.

Gas Turbine

Engineering mentions a patent issued to Brown, Boveri & Co., Baden, Switzerland, for a gas turbine. According to the processes heretofore proposed for working gas turbines, the preliminary compression of the motive medium is effected in piston or revolving compressors. The use of such compressors renders the gas-turbine installation complicated.

This invention consists in the compression, by the action of a water column to which kinetic energy has been imparted, without the coöperation of the gas turbine. In the form illustrated, a gas turbine *A* is directly coupled to an electric generator *B*. The gas turbine is provided with a combustion chamber *C*, into which are led a fuel supply *D* and an air pipe *G*, leading from a regenerative air heater *F*, the space around the tubes of which is fed from the air-compression space *K* above a water column *L*, the air being drawn into the space by a lead *N*. The passage through the pipes of the air heater is connected at one end *H* to the gas turbine, and at the other to an exhaust. At the end of the water column, remote from the air-compression space, is provided a separate explosion chamber *M*.

In operation, explosions are caused to take place in the chamber *M* whereby



Gas Turbine

energy is imparted to the water column, as in a Humphrey pump, and air is in turn drawn in at *N*, compressed in the space *K*, forced over the pipes of the heater *F* and through the passage *G* to the combustion chamber *C*, where, with the fuel supplied by the pipe *D*, the mixture is formed for driving the turbine. The exhaust gases leave at the port *H*, and flow in a counter-current through the heater *F*, imparting heat to the air supply.

CORRESPONDENCE

A Substitute Piston Ring

Some time ago a trouble man was called in to repair a small air compressor used in supplying air for starting a 50-hp. gas engine. He found the pump badly worn, and was advised by the operators that it took nearly 4 hr. to pump two small receivers to a pressure of 200 lb., whereas it originally took only ½ hr.

Upon starting the compressor, which was a single-acting, single-cylinder, belt-driven machine, with a 3½-in. cylinder and 4-in. stroke, considerable leakage was noticed past the piston. Removing the piston showed that a section about 2 in. long was missing from one of the two rings, which evidently was causing the leak. No extra ring being on hand, it was decided to try an experiment, to tide over until a new ring could be ordered. The ring groove in the piston was ¼ in. wide and about the same depth. A strip from a piece of ⅛-in. hard canvas rod packing was pressed tightly into the groove; then shaved down, leaving about ⅛ in. above the periphery of the piston. It was then taken out and reversed to have the even side out. After putting the piston back, the pump was started and worked as well and pumped up to the pressure as quickly as when new. This ring of hard packing kept the pump in good working order for over two weeks until a new cast-iron ring arrived.

L. M. Johnson.

Glenfield, Penn.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Meter Troubles and Shop Equipment for Testing*

System troubles which prevent correct metering come under three general heads: leaks, traps and seals blowing, and surges of condensation upon the meter. Causes of leaks are poorly packed valves, fittings leaking "90 drops per minute" at threads, and return lines which have rusted badly.

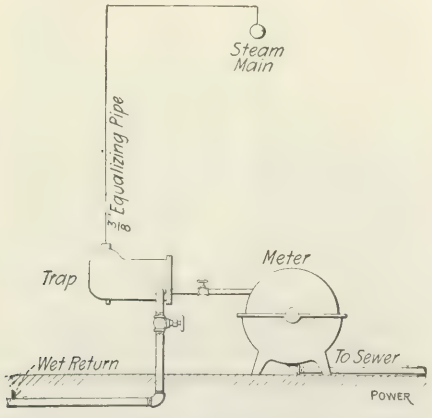


FIG. 1. INSTALLATION WITH WET RETURN

Rusting causes the most noticeable losses. One case coming to view recently, amounting to a tomato can full in 30 min., was found six weeks later to amount to 150 lb. per hour.

Traps and seals blowing, often practically stop meter registration while permitting steam to escape to the sewer. In one case, using metered injection water with the returns permitted checking the

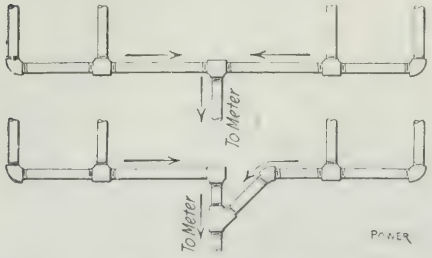


FIG. 2. BULL-HEADED CONNECTION TO METER AND REMEDY

loss of steam with and without one high-pressure bleeder trap; the measurement showed that about 300 lb. of steam had been escaping per hour. Traps sticking or seals choking interfere only by preventing legitimate use of steam. The remedies are, for seal blowing, to replace the seal with a "self-respecting" trap; for traps blowing, to adjust both the regulating valves and the trap; and for traps sticking, to clean and adjust the trap, with better finished valves and more care in installing. When a seal is found leaking, it is generally easy to make repairs or put in a trap at once.

Surging of condensation upon meters is usually caused by wet return jobs, return pipes brought together to the meter "bull headed" and bleeders which, poorly installed, cause "bucking." The most common troubles with suggested remedies are shown in the accompanying illustrations. Where such change is not immediately feasible, one of several devices may be installed to give relief. A receiver, generally submerged, having capacity for most of the surge without a rise in receiver water level of more than about 6 in., may be installed between the trap and the meter. Occupying less space

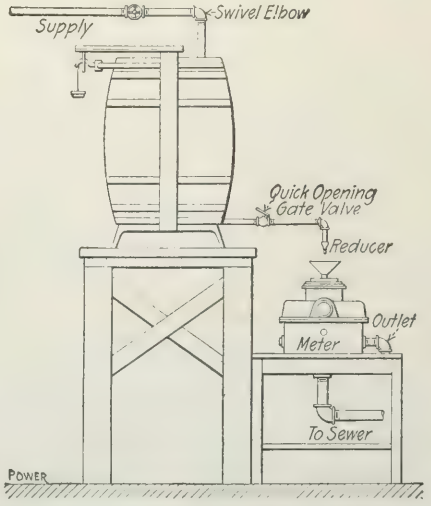


FIG. 3. SIMPLE TESTING OUTFIT

and requiring less investment, a muffler, built up of iron pipe and containing a number of notched rubber disks, so arranged as to form a zigzag path for the condensation through the restricted opening in the disks, is used between the trap and the meter to give a more even flow of condensation during surges, and to permit proper operation of the meter even if the trap is blowing.

Grease accumulation has been found so thick as to entirely stop a number of meters. The grease in these cases has either come from pump cylinder drips or from heating systems into which the pump was exhausted without proper oil separation. In one case, a prolonged kerosene bath changed a meter showing over 4 per cent.

SHOP TEST EQUIPMENT

The simplest equipment, Fig. 3, consists of a platform scale, a barrel fitted with a discharge-pipe connection, a water

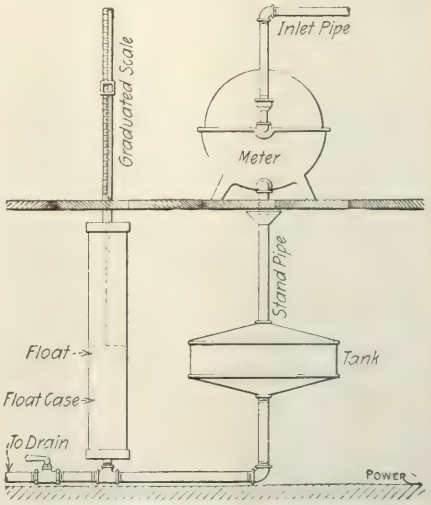


FIG. 4. TANK EQUIPMENT WITH GRADUATED SCALE TO INDICATE WEIGHT OF WATER

supply and a sewer-connection pipe, the meter to be placed upon the floor or a bench. Variations from this, due to the amount of testing to be done and the volume of water to be handled, consist of increasing the scale size to possibly 1500 lb. capacity with corresponding tank capacity in place of the barrel.

The inlet pipe should have a valve and swivel elbow, with a sewer connection handy by which any drip from the pipe or leakage through the valve may be diverted from the tank after the initial weight measurement is made. The discharge pipe should also have a valve and at the outlet a reducer or plug having a definite opening so that all tests of one size of meter may be made under conditions of uniform rate of flow. Quick-opening gate valves have been found advantageous for this work.

In addition to the equipment noted, a large bulb mercury thermometer with a range to 230 deg. inclosed in a wooden case open on two sides, is necessary where any work in hot water, steam or condensation is done. Standard weights for scale testing should also be provided.

Fig. 4 illustrates one form of tank equipment designed to avoid the use of scales. The apparatus consists of two cylinders of No. 14-gage galvanized iron, one of which, 6 in. in diameter, holds a galvanized-iron float for indicating the

*Excerpts from the report of the meter committee to the National District Heating Association at Detroit, A. P. Biggs, chairman.

water level; the other is a combination of a 1½-in. cylinder enlarging for some 8 in. of its length into a cylinder of 24 in. diameter. The enlargement is to contain the bulk of the water used in the test. The float rises rapidly until the water reaches the enlargement, and again rapidly and at uniform speed after the water level passes the top of this drum. A slider attached to the float, passes over a graduated scale and indicates the weight of water in the apparatus. The relative sizes of cylinders may be varied to give different scale divisions. The equipment described provides scale divisions of about 1 in. to 1 lb. of water.

Oak-Park's Hot-Water Heating System*

By G. E. Chapman

The plant of the Public Service Co. of Northern Illinois is in the village of Oak Park, in a one-story structure with a basement. The building is of red brick and reinforced concrete and has a floor space of approximately 22,000 sq.ft. The stack is reinforced concrete lined with firebrick, approximately 200 ft. high from the sidewalk level.

East of the plant is storage room for 10,000 tons of coal, which is received on an elevated switch from the Chicago & Northwestern R.R. tracks. The coal is dumped from the cars directly below the track, as long as there is room for it, and after that the cars are unloaded by a large Gantry crane, which is also used for conveying the daily supply to the bunkers.

In the boiler plant are 14 Stirling boilers, 10 of which have a rated capacity of 300 hp., and four of 450 hp. each; all have automatic stokers. Ashes are

*Abstract of a paper read before the National District Heating Association at Detroit.

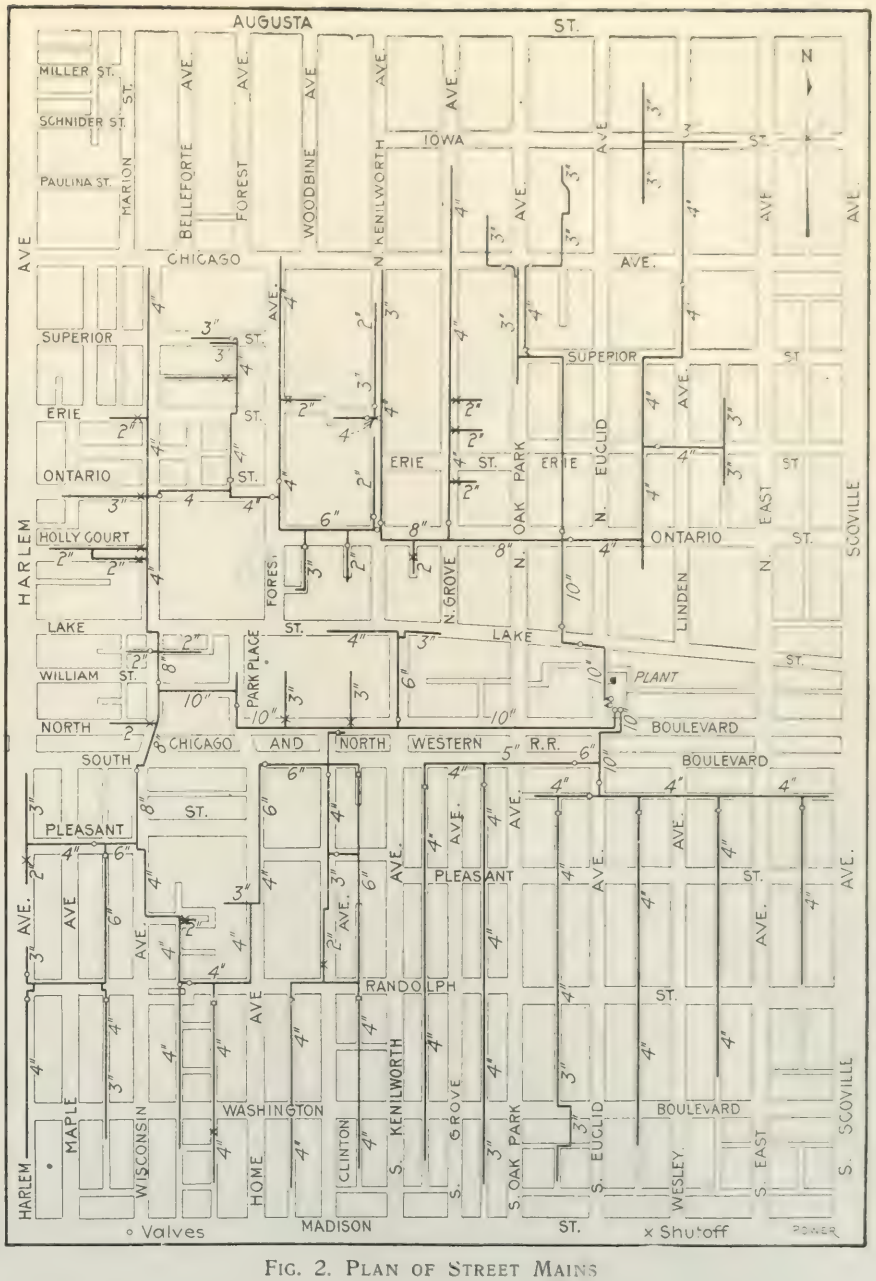


Fig. 2. Plan of Street Mains

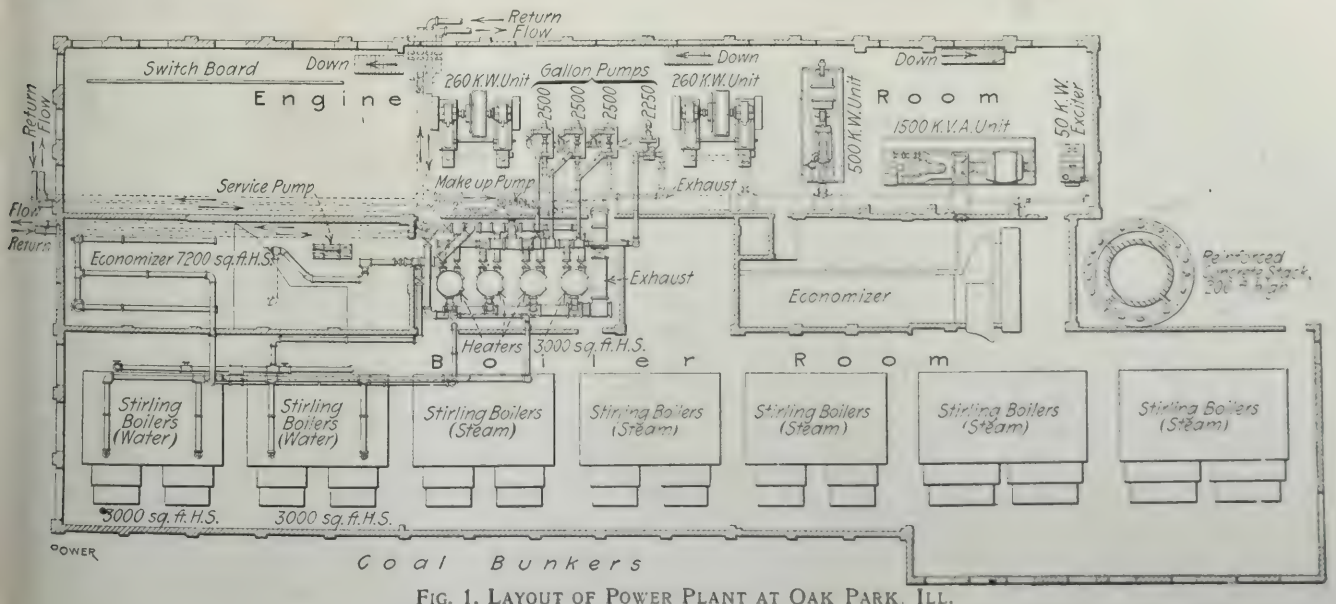


Fig. 1. Layout of Power Plant at Oak Park, Ill.

removed from the furnaces by a steel pneumatic suction system.

The four boilers at the south end of the plant are used exclusively for the heating system, and the remaining ten for generating high-pressure steam for the pumps and electrical machinery.

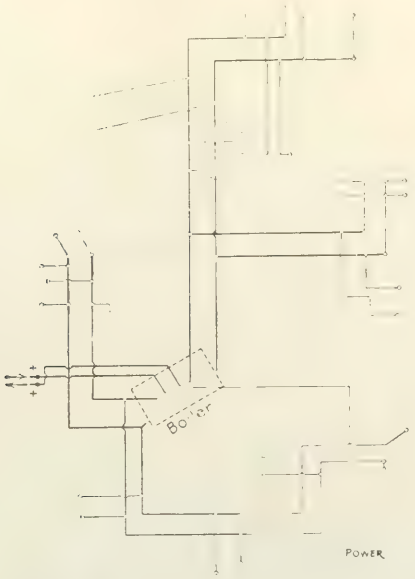


FIG. 3 A TYPICAL SERVICE LAYOUT

A Green economizer, having 7200 sq.ft. of heating surface, operates in connection with the battery of heating boilers, each of which has 3000 sq.ft. of heating surface.

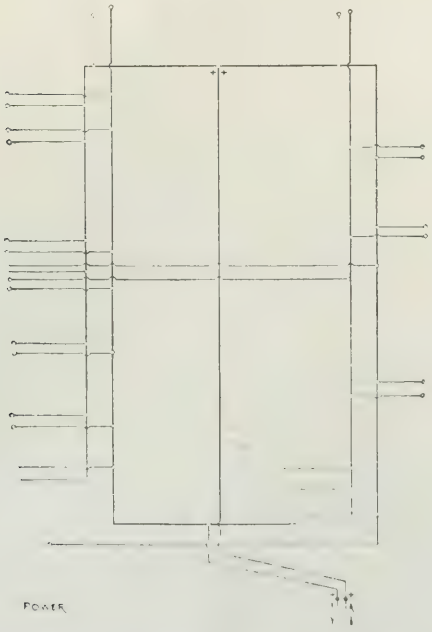


FIG. 4. STANDARD CONNECTIONS FOR HOUSE SERVICE

In Fig. 1 may be seen the four centrifugal steam turbine-driven circulating pumps for the hot-water heating system. One handles 2250 gal. and the other three each 2500 gal. per min. All are directly connected to the closed underground heating system. In connection with this pumping system is a duplex balanced-column make-up pump, automatically governed by the water pressure in the main return pipe. This pump makes up for any water which may be lost through leaks.

The underground system, Fig. 2, is laid out in accordance with the standard Yaryan methods, being a complete two-pipe system. The piping is protected by boxing built on all sides, with a double air space. The lumber is rough-sawed hemlock, preserved by dipping in crude oil. In addition, oiled shavings are tamped in the boxing around the pipes.

There are now approximately 12 miles of two-pipe heating mains, located as shown on the map. Connected to these mains are 710 houses with an aggregate of 550,000 sq.ft. of radiation.

The generating plant includes two Westinghouse turbo-generators, one of 1500 kv.-a., the other 500, both generating 2200-volt, three-phase, 60-cycle current, and two Ball & Wood reciprocating engines, each connected to a 250-kw. Western Electric direct-current generator. The exhaust from all steam-driven machinery is discharged into four Worthington heaters, which can be run in multiple, series or singly, as required. These heaters have a combined heating surface of 12,000 sq.ft. and are sufficient to take care of the entire heating system during mild weather; the heating boilers and economizer are used only in more severe weather.

The accompanying tables give the schedules, which were followed with good results during the heating season. Figs. 3 and 4 show typical piping arrangements for house service. One of these gives poor results, due to complications and short-circuits, but Fig. 4 practically follows our standard practice.

TABLE I. HOT WATER SCHEDULES

Schedules	1				2		3		4	
	Perceptible Wind 0 to 2 Miles				Gentle Breeze 3 to 10 Miles		Brisk Wind 10 to 20 Miles		Very High 25 to 50	
	Temp. Deg. F.				Temp. Deg. F.		Temp. Deg. F.		Temp. Deg. F.	
Gage Press. lb. per sq.in.	Flow	Ret.	Outside Temp. deg. F.	Gage Press. lb. per sq.in.	Flow	Ret.	Flow	Ret.	Flow	Ret.
32	120	100	65	32	135	118	145	112	155	127
32	125	110	60	35	140	120	150	125	160	130
33	135	118	55	35	145	122	155	127	165	132
33	145	122	50	35	150	125	160	130	170	135
33	150	125	45	35	155	127	165	132	175	137
35	155	127	40	40	160	130	170	135	180	140
35	160	130	35	40	165	132	175	137	185	142
35	165	132	30	40	170	135	180	140	190	145
35	170	135	25	40	175	137	185	142	195	147
40	175	137	20	45	180	140	190	145	200	150
40	180	140	15	45	185	142	195	147	205	152
40	185	142	10	45	190	145	200	150	210	155
45	190	145	5	47	195	147	205	152	212	156
45	195	147	0	47	200	150	210	155	215	156
45	200	150	5	47	205	152	212	156	217	157
45	205	152	10	47	210	155	215	156	220	158
45	207	153	15	47	212	156	217	157		
47	210	155	20	48	215	156	220	158		

TABLE 2. U. S. WEATHER BUREAU'S ANEMOMETER REPORT COMPARED WITH STATION'S DAILY AVERAGE

1910	Weather Bureau Report			Station Report				
	Daily Average			Indicated on gages			Operation	
Nov.	Vel. Miles	Direction	Schedule required	Direction	Schedule required	Vel. Miles	Schedule operated	Vel. Miles
1	16	S.	3	S.	2	5 to 10	2½	10 to 20
2	14	W.	3	W.	2	5 to 10	2	5 to 10
3	9	N.W.	2	N.W.	2	5 to 10	2	5 to 10
4	7	S.	2					
5	9	W.	2					
6	7	W.	2					
7	9	W.	2	W.	2	5 to 10	3	10 to 20
8	6	S.	2	S.	2	5 to 10	2	5 to 10
9	16	S.W.	3	S.W.	2	5 to 10	3	10 to 20
10	18	N.W.	3	W.	3	10 to 20	3	10 to 20
11	19	N.W.	3	N.W.	3	10 to 20	3	10 to 20
12	15	N.W.	3	N.W.	2	5 to 10	2	5 to 10
13	6	W.	2	N.W.	1	0 to 4	1½	5 to 10
14	13	N.W.	3	N.W.	2	5 to 10	2	5 to 10
15	13	N.W.	3	N.W.	2	5 to 10	2	5 to 10
16	11	W.	3	N.W.	2	5 to 10	2	5 to 10
17	12	S.W.	3	W.	2	5 to 10	2	5 to 10
18	10	N.W.	3	N.W.	2	5 to 10	2	5 to 10
19	8	S.	2	S.	1	0 to 4	2	5 to 10
20	13	S.	3	S.	2	5 to 10	2	5 to 10
21	16	W.	3	W.	2	5 to 10	2	5 to 10
22	11	S.	3	S.W.	2	5 to 10	2	5 to 10
23	19	S.W.	3	W.	3	10 to 20	3	10 to 20
24	15	S.	3	S.E.	2	5 to 10	2	5 to 10
25	12	W.	3	N.W.	2	5 to 10	2	5 to 10
26	16	E.	3	S.E.	2	5 to 10	2	5 to 10
27	14	S.E.	3	S.E.	3	10 to 20	3	10 to 20
28	18	N.W.	3	N.W.	3	10 to 20	3	10 to 20
29	16	N.W.	3	N.W.	2	5 to 10	2½	10 to 20
30	25	N.W.	3	N.W.	3	10 to 20	3	10 to 20

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POWER

Rights of the Consumer

That it is not always advisable to accept a very low rate from an electric company is strikingly illustrated by a recent decision of the Appellate Division of the New York State Supreme Court in the case of the Globe Woolen Co. versus the Utica Gas & Electric Co.

The Globe Woolen Co. made a contract with the electric company for five years at a rate of 1.04 cents per kilowatt-hour with the privilege of renewing the contract for five more years. Furthermore, the electric company guaranteed to save the woolen company $600 a month in the operation of its plants as against the previous operation by steam power. In case of shortage of power the woolen company and the city of Utica were to be served ahead of all other customers.

This contract was negotiated by the president of the woolen company, who was also chairman of the executive committee of the electric company and owned a large part of the stock of the woolen company, but had no financial interest in the electric company. It is claimed that, although attending all the meetings of the executive committee of the electric company at which this contract was discussed, he did not vote on its acceptance.

After a few years the electric company found that it had delivered to the woolen company current to the value of over sixty-nine thousand dollars for which it had received no compensation under the contract, and in addition owed the woolen company nearly twelve thousand dollars on account of the promised savings. It thereupon offered to remove from the woolen company's mills, without cost, all the central-station service equipment, to reimburse the latter for its cost and to reinstall the necessary boilers and other steam equipment to permit the operation of the mills by steam as formerly.

The woolen company allowed the electric company to reinstall the steam equipment at an expense of eighteen thousand dollars, stipulating, however, that this was to be without prejudice to any rights under the contract.

Shortly after the woolen company elected to renew the contract for a second period of five years but the electric company refused and served notice that the power would be cut off on a certain date. The woolen company immediately brought suit in equity for an injunction, and a temporary injunction was granted pending the trial.

The outcome of the trial was that the Appellate Division of the Supreme Court held the contract not to be enforcible against the electric company and said in part:

"It would seem to be unnecessary to multiply the citation of authorities in support of the proposition that where contracts are made or entered into between two corporations under the circumstances disclosed, and the result thereof is that the contracts are greatly to the advantage of the plaintiff and to its president individually, who negotiated the same, and are grossly inequitable and burdensome to the other party, such contract may be repudiated at the option of the party aggrieved, and if such repudiation is accompanied by an offer of the aggrieved party to place the other party in the same position in which it was before making such contracts, they thereby cease to be valid and binding and are not enforceable either in an action at law or in equity."

Presumably the electric company had carefully investigated the operating conditions at the woolen mill before offering the exceptionally low rate, but it is apparent that it had neglected to take into consideration the value of the exhaust steam for heating and for manufacturing purposes.

While the case at hand savors of the personal influence of the woolen company's president, still it is by no means an isolated case, as contracts of this nature are continually being entered into. The customer is naturally desirous of obtaining as low a rate as possible while the central stations in their eagerness to get the business (sometimes to effect a

monopoly, and other times to serve publicity) often make propositions which they are unable to meet without loss.

The rights of the consumer under such a contract is of vital importance and the recent court decision would not seem to strengthen these rights. Hence it behooves the consumer to think twice before entering into a contract for electric power at what appears an attractive rate.

New York Boiler Inspectors again Discredited

Quietly, obscurely, in inconspicuous corners of the daily papers overshadowed by the reports of graver charges against his department, comes an announcement by Police Commissioner Waldo, of New York, that policemen Owen Kelly and John Heffernan, of the boiler squad, have been suspended, charged with accepting money in connection with their duties.

Is not the time propitious for getting the examination and licensing of engineers and the inspection of boilers into a better atmosphere? The past attitude of some of the New York City engineers would indicate that they rather enjoyed putting up from twenty to fifty dollars every time they went or sent to headquarters, or that they hoped to avoid by friendly service the personal "touch." An intelligent united effort for a state license law, uniform with those of Massachusetts and Ohio, would find some of the most formidable opponents of past efforts for such legislation very busy trying to keep themselves out of jail.

Kansas City

On Sept. 10 the thirtieth annual convention of the National Association of Stationary Engineers will be held in Kansas City, Mo. Those who attend are to be congratulated if on no other score than that they will visit this great and going metropolis of the Missouri and her mighty little sister lying just across the state line and the Kansas River.

Although combined these two cities would be only seventeenth in population in the United States, they are seventh in postal receipts, sixth in bank clearings, third in telegraphic business, horse and mule trade, flour output and lumber, second in meat-packing, railroads and in grain receipts and first in yellow-pine lumber, sale of agricultural implements and Pullman business. What is perhaps most significant of the future, however, is that they are first in tributary territory and first in the richness and extent of contiguous farming territory. At any rate, the 18 railroad systems, representing 34 lines that enter Kansas City, Mo., seem convinced that a splendid future is in store for these municipalities, for they are now spending $35,000,000 for improved terminal facilities alone. The new union station, now being erected and which alone will cost $4,500,000, it is said will be the most modern in the world.

As may be inferred from the foregoing, the Kansas Cities are good business and manufacturing communities. The Mercantile Club of Kansas City, Kan., lists over 175 manufacturing plants in that city and representing over 60 different lines of business. Kansas City, Mo., contains a proportionately greater number. John F. Linden, boiler inspector for the latter city, is authority for the statement that there are 190 power plants in Kansas City, Mo., containing an aggregate of 837 boilers and 160 plants containing a total of 220 high-pressure boilers and giving employment to 518 first-class licensed engineers.

Criminal Negligence

The need of a state license law in Kansas was emphasized by the recent explosion of a thirty-inch by six-foot vertical boiler at a rock-crushing plant in Rosedale. The engineer was killed and another severely injured. The boiler, which was connected to another vertical boiler of larger size, was thrown sixty-five feet from its setting.

A boiler maker, appointed by the coroner to determine the cause, found no safety valve or steam gage. The packing-gland nuts on the water column showed that there had been no glass and only one of the gage cocks was in working order; the other two were found plugged. Corresponding with the general state of dilapidation, the blowoff valve had no wheel. The back plate of the firebox and the tube sheets of the damaged boiler had been bulged in and twenty-seven out of the fifty-five tubes were out of place, the ends giving evidence of considerable deterioration. The boiler was reported to be a castoff from Kansas City, where a license and boiler-inspection ordinance is in force.

It is the same old story, with a little more of it. We have reported similar explosions, but have never heard of a case where the operating conditions were quite so adverse to safety. The practice of buying old and discarded boilers is too dangerous. What does the small saving in initial investment amount to when compared with the loss of life and destruction which is almost sure to follow?

Evidently legislation providing competent inspection and regulation is badly needed, and this applies to many other states in the Union besides Kansas.

'Twas ever thus. John Bourne in the preface to the first edition of his "Treatise on the Steam Engine," published in 1846, says: "I have been obliged to confide the greater part of the theoretical part of the present work to some mathematical assistant whose algebra has, I fear, sometimes risen to a needless luxuriance, and in whose superfine speculation the engineer may perhaps discern the hand of a tyro."

An act providing for the inspection of steam boilers has been passed by the Legislature of New Zealand and goes into effect on Sept. 1 of this year. Of these United States only two have such laws in universal application and we count up a boiler explosion of sufficient importance to get into the papers at the rate of more than one every working day.

The operating engineer who attempts to perform work foreign to his vocation or outside his capacity—and his tribe is decreasing—calls to mind the chameleon who tried to make good on a Scotch plaid, and proved a dead failure.

A contemporary has recently been the medium for exchanging the views of the head of an engineering supervision company and a chief engineer which it displaced. According to the latter, he had been trying for years to get the management to adopt certain improvements but without success. When the supervision company stepped in, however, it easily convinced the management as to the needed improvements. It would seem that if the operating engineer had presented his case as clearly and convincingly (as the supervision company evidently did) there would have been no intrusion of outside interests.

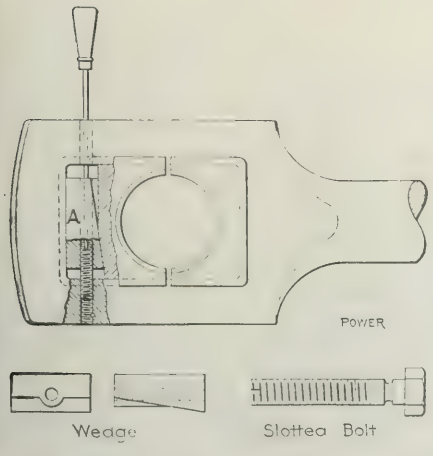
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Slotted Connecting-Rod Wedge Bolts

The solid-end rods used by some manufacturers of large engines have the wedges placed in the slot of the rod with the brasses before the rod is put on the crankpin or connected in the crosshead. This makes a very desirable form of adjustment to take up the wear of the brasses, but with some forms of these wedges it is difficult to remove the brasses or wedges to put shims under the brasses to true them up and raise the wedge. With such types of rod ends removing a broken wedge bolt is also very difficult. This could be avoided if the wedge bolts were slotted in the end to allow using a screw driver, as shown, for removing them. The following circumstance well illustrates this:

The bolt broke on the crankpin end of the rod just as I was keying up at noon.



REMOVING BROKEN WEDGE-BOLTS WITH SCREWDRIVER

Thinking this wedge was only slipped in back of the brass and flange, which I knew was bolted to the ends of the brasses, I proceeded to remove the washer from the end of the pin only to find a wedge I could not remove without taking out the wristpin and pulling the crosshead back. This allowed removing the wedge and brasses from the pin by moving the connecting-rod outward at the same time.

The wedge *A* in the illustration was very thin at the top, and the bolts were ¾ in. To get strength enough at this point of the wedge to stand the drawing strain of the bolt, the wedge was cast with a ½-in. projection on the incline along the bolthole, and this projection fitted into a slot in the back of the brass next to the wedge. The wedge could not be removed without removing the brasses. The brasses could not be removed on account of the flanges on the side of them. To do this the connecting-rod would have to be taken out first.

This would require about two hours, and as it was the bottom bolt that broke, I left the job until evening, the top bolt keeping the wedge up tight. After much trouble I succeeded in slotting the end of the bolt after which a screwdriver was used as shown to back it out. The end of the new bolt was slotted before being put in.

The builders should slot the ends of the wedge bolts as it saves much time and trouble in taking them out when they break off in the rod.

R. A. CULTRA.

Cambridge, Mass.

Rosin for Bending Boiler Tubes

Using sand for bending pipe and tubes is not always successful, as kinks are likely to form despite care in bending. To obviate this trouble a friend recommended rosin to fill the tubes instead of sand, so the tubes were filled with melted rosin which was allowed to get quite cool. Then they were taken to a flange press where, one at a time, they were bent over a former to a neat quarter circle with 20-in. radius, and scarcely a kink was noticeable. In most instances, perfect results were obtained.

The use of rosin is quite inexpensive, as it can be melted out of the tubes after bending and can be used over and over again. I have never seen rosin tried in bending pipes, but see no reason why it would not do as well for pipes as for boiler tubes. I presume there is scarcely an engineer who has not been disgusted at various times with his efforts to make a neat pipe bend.

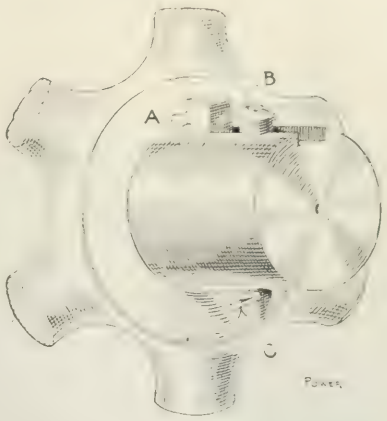
C. R. SUMMERS.

Chattanooga, Tenn.

Safety Pin for Flywheel Keys

The sketch shows a device used by a firm making portable engines to safeguard against accident. They are largely operated by unskilled labor for agricultural work. Under these conditions it is possible that if the key slacks back it would pass out unnoticed and lead to serious accident. With the provision illustrated, the key cannot work out of the slot and is bound to be noticed and tightened before harm is done.

Each engine has two flywheels, each secured by a draw key *A*, as shown. After the key is driven home, a hole is drilled in the keyway of and through the shaft, and a short taper pin *B* is fitted in, having a split pin *C* at its smaller end to prevent it from coming out.



SAFETY PIN TO PREVENT ACCIDENTAL REMOVAL OF FLYWHEEL

The cost of the device is trivial and the security against accident is undoubted.

A. L. HAAS.

London, England.

Writing for the Papers

Not many years ago, writing was supposed to be an accomplishment possessed by the professional engineer, but seldom indulged in by the so-called practical engineer. The latter thought he could not write a satisfactory letter, so simply let it go without trying. Much information of value was thus lost that would have been valuable to others. Now POWER makes it plain that it wants to hear from anybody who has anything to say that will benefit others of the craft. This is an age of giving away, particularly so in the realm of education. As Elbert Hubbard says, "The more education we give away to others, the more will we have left for ourselves."

Lots of engineers have lots of information to give away, or at least to exchange for some they need. They have the chance to give and to get; to cultivate any latent talents they may have; to

broaden out mentally—but not in the sense of getting a swelled head..

Get out of self, and get into the arena where the battle of progress is going on. Be on the side of those who are helping it along. Don't be a hinderer—though passive. Don't be like the person in the following verse:

"As I walked by myself, I talked with myself;
Says myself to me,
Lookout for yourself, take care of yourself,
For nobody else cares for thee!"

That is the attitude assumed by many, but is a dangerous rut to get into. Service to others, helping them to help themselves that they in turn may help others, is the truly satisfactory life to live, and one very good way to help others is to write and tell them what you know and what you do, and it will come back to you multiplied.

Byron wrote:

"But words are things, and a small drop of ink,
Falling, like dew, upon a thought, produces
That which makes thousands, perhaps millions,
Think."

CHARLES J. MASON.

Scranton, Penn.

Pressure Chart Reveals Negligence

The accompanying charts from a recording-pressure gage graphically illustrate the importance of records in a plant.

A large flywheel steam pump delivered water through a 10-in. main from which large quantities were being continually taken at different points. The men in charge of the pumping outfit being negligent, insisted that they could not control the pressure to show a chart any better than Fig. 1, and even this chart is much better than some recorded while the pump was under their charge. A new chief engineer was appointed who undertook to run things for a straight 24 hr. to determine the cause of the fluctuating pressure. He obtained chart, Fig. 2, with only reasonable attention to his work. The men were then notified that they could either attend to their duties properly or resign. They decided to retain their jobs. They were on their regular shifts with the chief when he ran his test, and had no excuses to offer, as they were shown plainly that all that was necessary to obtain a fair record was honest attention to their duties. Their pressure diagrams are now as good as and often better than those obtained by the chief.

JOHN THORN.

London Junction, Ont., Can.

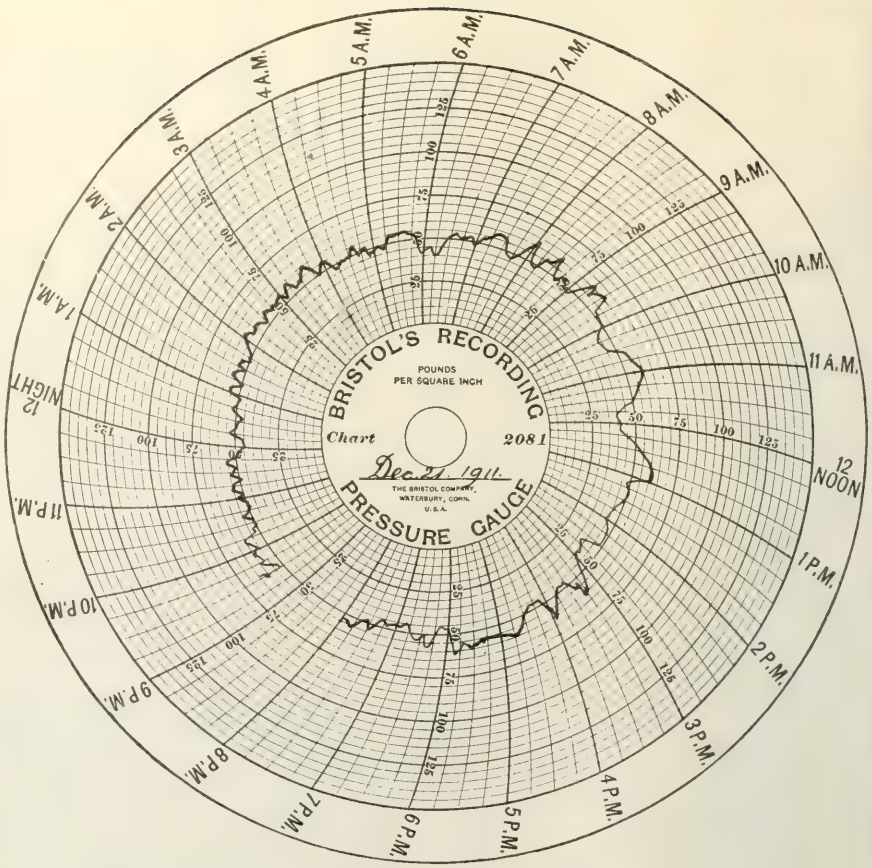


FIG. 1
PRESSURE MAINTAINED UNDER NEGLECTFUL OPERATION

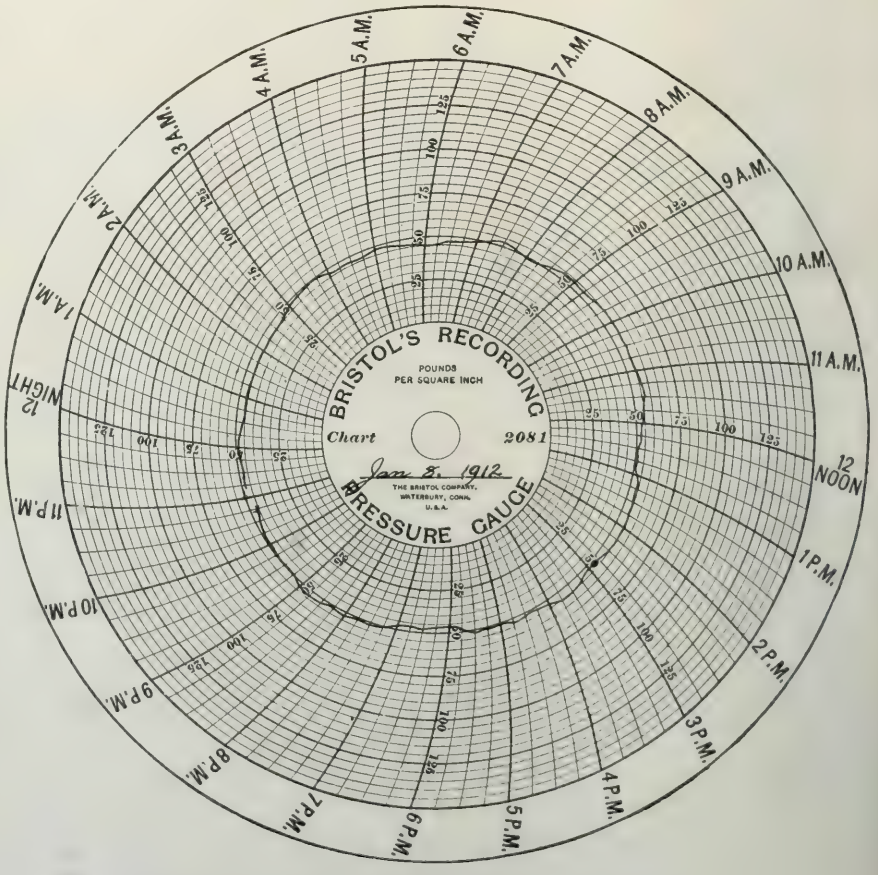


FIG. 2
PRESSURE MAINTAINED UNDER CAREFUL OPERATION

Baffle Plates in Water Tube Boilers

Strange as it may seem, there appears to be a tendency to neglect to an alarming extent the baffle-plates in a water-tube boiler. In some cases, ignorance, and in others sheer laziness is the cause. I have heard it said positively that the plates were no good, were only dirt catchers, and in one case they even went so far as to take them out of the boiler entirely "to give her a free draft," as they explained later. Needless to say they were replaced. When an engineer has had opportunity to see the detrimental effects of broken and leaky baffle-plates upon the efficiency of the boiler, and upon the coal pile, he will never neglect them again.

Men are prone to neglect the nasty, hot jobs around a boiler plant, especially where they are not easily seen. To the average boiler cleaner, no doubt it seems a small matter if a few of the bricks in a baffle-plate are gone, and a few holes burned in the plate. He probably figures that the chief will never bother to climb into the boiler and examine them, and often he is right, as some engineers are satisfied to take the word of their subordinates rather than get into overalls and get dirty in a personal examination. The chief of a large plant cannot attend to every little detail, but he should always have time to examine his boilers.

Replacing the bricks is a hard, nasty job. To do it quickly, takes three men, two with bars to pry the tubes apart and one to insert the brick. Often the corners are knocked off the brick so that it will slip in place. This is bad as the space is not completely filled, and a firebrick once broken apparently continues to break until it falls out again. A tool now on the market will do the job very neatly and without breakage, so not keeping the baffle-plates in good condition is inexcusable.

WILLIAM N. WING.

Brooklyn, N. Y.

The Use of Fuel Oil

That fuel oil is not more used under boilers is not from lack of good oil burners. Even where fuel oil is cheap, the deterioration of boilers from burnt out tubes for the first year or two will more than offset the saving in fuel, and some users will go back to coal. Experts will appear, offering a special plan of bricking up the combustion chamber, each claiming that the boilers will then last as long as when burning coal. Generally they do but little good, for the work along this line seems to be in the wrong direction.

In coal firing the object is to generate a certain amount of heat, confine it as much as possible to the heating surface of the boiler, and have it evenly distributed, without any cold streaks along any part of the heating surface. As Kent's "Steam Boiler Economy" says, the largeness of the firebox does not detract from its efficiency, as the heat generated cannot escape except by going through the regular paths and giving up its proper portion of heat to the surface, once the walls have been heated up.

When such a boiler is to be converted for oil burning, a hole is drilled between the two firedoors, a burner inserted, all connections made and the blast turned on. A hot fire is the result, for the same amount of heat must be produced by this one flame as by a bed of incandescent coal evenly distributed over the whole grate area. Therefore the flame temperature must be considerably higher than with the coal fire to produce the same amount of heat. The blast of an oil burner draws in a stream of cold air on either side of it, which follows along either side of the current of hot gases throughout the heat passages to the stack. So a part of the boiler evaporates more water and faster because of the higher temperature than was intended by the designers, while other parts possibly receive more heat from the water circulating in them than from the firebox.

With converted boilers in a new field, the above conditions hold in nine cases out of ten. Some makers send out boilers arranged for burning crude oil; one such boiler that I happened to see had three burners for only 1200 sq.ft. of heating surface.

Personally, the writer favors using a burner every 6 in. across the front of the firebox, and obtaining the required amount of heat without excessive temperature in any one place.

Smaller burners and more of them would seem to be the proper idea.

SAM FOSTER.

Tampico, Mexico.

Systematic Note Keeping

Those who write for the technical press often think of ideas or incidents, that would interest readers, but, failing to note them down, forget them, and thus much that would be of value to engineering literature is lost.

To keep track of all such ideas and experiences I have procured a desk calendar pad, with a separate sheet for each day and date and thereon record everything worthy of notice. At the end of the week I transfer these notes to an indexed ledger, somewhat on the order of the indexes issued by POWER. I can refer to it at any time with the knowledge that the incident will be vividly recalled by the information conveyed in the heading.

The dates are kept up as mercantile accounts, and when I have written and mailed an article on a subject, I indicate it by the mark "x," and when the article has been published, I check it off, designating in figures, as "6-15-12," the month, day and year published.

From my experience I believe a greater number of engineers would write occasionally, and many could write more and better letters if they would but keep things of this sort collected more properly, since, by the haphazard methods usually employed, they lose interest and put off writing upon a timely topic until the subject begins to wane in the columns of their paper.

If anyone can suggest an improvement, or a better way altogether than mine, I would be pleased to see it published.

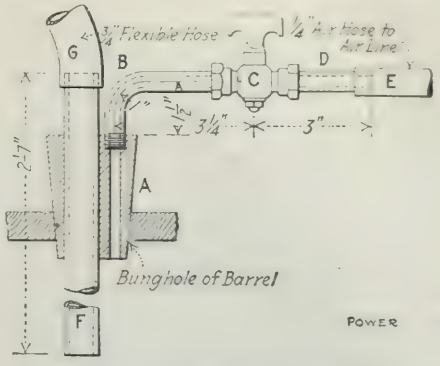
LLOYD V. BEETS.

Nashville, Tenn.

Emptying Oil Barrels by Compressed Air

The means a company employed to transfer oil from barrels to storage tanks was very cumbersome. The tanks were on the first floor, and a barrel to be emptied had to be hoisted above the tank inlet. I designed the device illustrated, which will empty a barrel in a few minutes without much labor.

A cast-iron plug *A* is made to fit the bung hole of the barrel, a ¼-in. air pipe *B* is tapped into the top of the plug, as shown, and a ¼-in. air valve *C* is screwed on this air pipe. A short nipple *D* and a ¼-in. air hose *E* are secured on the end of the nipple and connected to the main air line. The hose should be long enough for the device to be moved from



AIR-LIFT FOR OIL BARREL

tank to tank. A ¾-in. standpipe *E* should slide snugly through the cast-iron plug so no oil can escape. On the end of the ¾-in. pipe is a ¾-in. flexible hose *G* long enough to reach the inlet of the tank. The operation is as follows:

Roll the barrel near the tank, take out the barrel plug and insert the cast-iron plug by giving a few taps with a hammer so as to make a tight fit, force the ¾-in. pipe to the bottom of the barrel, connect the flexible hose to the tank, turn on the air, and in a few minutes the barrel will be empty.

W. H. WOLFGANG.

Toledo, Ohio.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Smoke Prevention with Automatic Steam Jet

In the Jan. 16 issue there appeared reports of two boiler trials conducted by the writer, showing how a fuel saving of 10 per cent. was effected together with increased plant capacity by the installation of smoke-preventing apparatus.

A criticism by Henry Misostow appeared in the Feb. 20 issue in which the critic makes a rule-of-thumb statement to the effect that the difference in draft in each set of tests is wholly inadequate to account for the saving of fuel seemingly realized; and yet it is sufficiently great to prove to his satisfaction that the increase in economy was due to a difference in the manner of handling the fires. In conclusion Mr. Misostow says: "So far as an economy of 10 per cent. of fuel goes, the steam-jet man must be given credit for doing the right thing at the right place in these two cases, obtaining the same result by different changes in the two furnaces, and the credit for the whole change is given to the steam jets."

Before replying specifically to the criticisms, I might say that at each plant the draft was practically unchanged for the two tests; furthermore, the handling of the furnaces was almost identical for both tests. The "steam-jet man" had nothing to do with the tests, not even being present as a spectator. Moreover, credit for the fuel saving is given by the writer not to the steam jets but to the automatic steam-jet apparatus—a distinction which I hope to make clear.

The crux of the entire criticism lies in these words of the critic: "The rate of combustion depends on the draft." This is a very orthodox statement; if coal were pure carbon it would be strictly true, and with anthracite coal the rate of combustion does depend almost entirely upon the draft. But the statement emphatically does not apply to soft coal; and especially where hand-fired.

To corroborate my statement, I shall follow very closely the matter presented in a recent bulletin of the United States Bureau of Mines, upon "Steaming Tests of Coals and Related Investigations." When a fresh charge of bituminous coal is spread over the hot coke on the grate a considerable part is rapidly distilled off. This distillate consists of gases and liquids—by liquids being meant the heavy hydrocarbons which are carried upward in a pasty condition and which probably exist as small globules of coal tar. This gaseous and liquid combustible must be burned in the combustion chamber, and the completeness of the process depends upon the length of time it remains within the combustion space, on the temperature, on the supply of oxygen admitted through the firing doors and on the thoroughness with which the oxygen is mixed with the combustible. If the mixture is good the gases burn in possibly a fraction of a second, but the heavy hydrocarbons require a much longer time. Furthermore, to burn them completely requires a violent mixing with the air since the oxygen can act only on the surface of each globule, and each globule contains many times more combustible matter than a like volume of gas. If these globules are carried out of the combustion zone and cooled below the ignition temperature before the process is complete, they appear at the top of the stack as heavy smoke and probably form the greater part of the loss, due to incomplete combustion.

Poor combustion due to small combustion space is usually made worse by inadequate mixing of the gases. It has been shown that the gas leaving the fuel bed is very rich in combustible and deficient in free oxygen. Therefore, it is always necessary to add air to the gases rising from the hot fuel. If the air is admitted through large openings, such as the firing doors, it may pass through the furnace as one solid stream alongside of another large stream of combustible gas, without materially improving the completeness of combustion. To help the combustion, the air must not only be admitted but must be intimately mixed with the combustible gas if the latter is to be completely burned in the short time that it is allowed to stay within the furnace.

From this discussion it should be clear that it is not the rate of combustion which depends on the draft, as Mr. Misostow states, but the rate of driving the combustible out of the fuel—an altogether different matter.

However, my purpose in quoting from the bulletin is not primarily to controvert my critic, but to emphasize two important considerations in connection with smoke prevention. The first is: The appearance of smoke from a chimney is evidence of incomplete and wasteful combustion. The second consideration has to do specifically with the automatic steam-jet apparatus which was the subject of my former article. This apparatus consists, in part, of steam jets which begin automatically to discharge steam into the combustion chamber whenever a fire-door is opened, in part, of special fire-doors carrying large dampers which remain open when the doors are closed, and finally, of a dashpot arrangement whereby the steam is gradually cut off and the dampers are gradually closed after a predetermined period following stoking has elapsed. The apparatus accomplishes precisely the important function referred to in the preceding discussion from the Bureau of Mines bulletin. The open dampers afford the needed avenue for the inspiration of air. The steam jets act as an inspirator and heater of the air, causing the relatively slow influx of a relatively large volume of heated air just when the combustible distillate is rising from the fuel bed, and finally, the steam jets cause such a swirling within the combustion chamber as to insure mixing of the gases.

The economy effected is therefore due partly to the more complete combustion and partly to the resultant cleanness of the boiler tubes which ensues from the almost complete absence of the formation of smoke. To the latter effect may be ascribed the increased steaming capacity.

J. A. SWITZER.

Knoxville, Tenn.

"Principe di Piedmonte" Accident

The account of this accident and the editorial in the July 30 issue recalls a similar accident to the old steamer "Osceola" on the Great Lakes some years ago when a cast-iron ell on the steam pipe to the main engine burst. The chief engineer, whose room was directly above the ell, was fatally scalded.

Cast-iron ells or fittings are not now permitted in marine service, but copper pipes, unless properly inspected and annealed from time to time, become almost as unreliable as cast iron when allowed to become crystallized as must have been the case on the "Principe di Piedmonte." The rolling and working of a ship in a heavy sea sets up a great variety of strains unknown in stationary service and require extreme vigilance on the part of the engineers to prevent accidents.

I have taken down and inspected cop-

per steam and water pipes on steamboats and have very often found cracks reaching nearly around the pipe which sometimes carried steam at 100 lb. or more. A man standing alongside or directly over a pipe so affected while handling the engine is in constant danger of being scalded to death. Marine inspectors should compel all copper piping carrying steam at ordinary pressures to be annealed at regular intervals sufficiently close to prevent crystallization of the pipe.

A. A. BLANCHARD.

Lakeland, Fla.

[The rules of the U. S. Board of Supervising Inspectors of Steam Vessels relative to copper pipe state that flanges on such pipe of over 3 in. shall be of bronze or brass composition and securely brazed to the pipe. All such flanges shall have a boss projecting from the back of the flange a distance sufficient to be properly riveted to the pipe and of ½ in. thickness.

The breakage of such pipe usually occurs along the rivet circle due undoubtedly to crystallization. No mention of annealing is made in the rules.—EDITOR.]

Securing a Loose Piston

While running a small engine the piston became loose on the rod as in the case mentioned by John Creen in the July 2 issue. The rod screwed through the piston and was riveted tight. The riveted end broke off, so I had the rod screwed into the piston as tight as possible, then two holes were drilled and tapped for ½-in. capscrews on the periphery of the rod and the hole in the piston. After making the screws tight, they were cut off flush with the piston. This engine was in operation without any further trouble for three years after the repair, until a larger engine was installed.

ROY STAHL.

Rising Sun, Ohio.

Gate Valve Repair

Several critics, accustomed doubtless to the smaller valves used for steam piping, have scouted the account of a repair of a gate valve by through bolts between the flanges, described in POWER for June 25, page 891, as impossible because "the body is always nearly as large as the flanges." Anybody who is sufficiently interested will find the valve repaired as described, in the Municipal Lighting & Pumping plant at Orange, N. J. This is a big water-works valve on the suction of a pump, as the article states. Furthermore, there are many standard valves made which will permit of rods passing through the flange bolt-holes, past the body of the valve at the sides and bottom.

They are also facetious about the remark that the fracture was caused by shutting the pump down too suddenly. Water comes to the pump under a 49-ft. head, as stated in the article describing the plant, and published in the May 9, 1911, issue of POWER. The pumps take the water from the suction pipe about as fast as it is supplied, and at times a sudden stoppage of the pump has built up a pressure in excess of 90 lb. per sq.in. Water-hammer was the direct cause of the broken valve as attested by the fact that the installation of a relief valve on the suction pipe set to operate at 25 lb. per sq.in. has eliminated the trouble, and no further shocks have been produced on the pipe and valve.—EDITOR.

High Place on Diagram

Probably the high point at the top of the admission line of the diagram submitted by W. J. L. in the June 25 issue is due to a small steam line with a large separator near the throttle. The separator will act as a receiver to force the pencil to indicate a greater steam pressure than can exist throughout admission due to the small steam line. Reducing the lead will no doubt tend to partially remedy the difficulty.

R. G. LOVELL.

Courtland, Ariz.

Speed of Flywheels

The answer to P. S. on calculating the safe speed of flywheels, page 749, May 21 issue, was interesting. With cast iron the low stress per square inch, as given (or the high factor of safety), is used for several reasons, principal among which are, the unreliability of cast iron, its liability to shrinkage strains, and the serious results of failure. While engine speeds and flywheel diameters are ordinarily such as to make a peripheral speed of about a mile a minute perfectly convenient, it sometimes happens that, in machinery other than steam engines, more inertia or flywheel effect is needed than can be obtained in the limited space available. This must then be obtained by velocity. Steel readily suggests itself but, as cast steel costs considerably more than cast iron, the cost may at first seem prohibitive. This objection will disappear when calculations are based upon the higher safe speed permitted by steel and the consequent smaller dead weight required to obtain any given inertia.

When space is not limited, the same idea applies to lighter rims and greater diameters, since, at a given rotative speed, inertia varies as the square of the diameter.

When it is desired either to speed up a given wheel or to reinforce it, or both, a convenient way of using steel of very high tensile strength is to use passenger-car wheel tires, or locomotive-driving tires, depending upon the desired size. They may be purchased without flanges, already bored to the exact diameter desired, at prices close to the price of steel castings and sometimes for less. If the full thickness of a new tire is not required, old tires, which have repeatedly been turned down on account of wear until they are about 1½ in. or 2 in. thick, may be obtained from any railroad at a low figure, and these, when shrunk on, make a good means of reinforcing an old wheel or building up a new one for special purposes. With new tires the tensile strength varies with quality, between 90,000 and 120,000 lb. per sq.in. This is because some classes of service require soft tires and others hard. A somewhat lower figure should be used with tires that have been subjected to vibration for years.

E. G. MEDART.

Quincy, Ill.

Safeguarding Life at Sea

Your editorial comment in the June 11 issue, "After Honors—What?" I consider to the point, and I trust you will continue to urge the need of safeguarding the lives in ship's boiler and engine rooms until your efforts are rewarded and those who go to sea are properly considered and provided for.

There is much room for improvement in properly equipping ships for safety, and, while the horrors of the "Titanic" disaster are fresh in the minds of all, is the proper time to begin to analyze the suggested improvements and select those that are practical and effective.

I think, as you suggested in your editorial, "The Inconsistency of the 'Titanic' Disaster," May 7 issue, that it is reasonable to believe that this ship could have been kept afloat until all on board were rescued, if some way had been provided for quickly connecting the engines to powerful centrifugal pumps. This seems a simple enough matter, as the pumps might be located forward of the engines with disengaging couplings, or if it is desirable to run the pumps faster than the engines, which is very likely, suitable gearing could be provided.

LLOYD V. BEETS.

Nashville, Tenn.

Repairing Boiler Tubes

Mr. Burns' article under this title, in the June 25 issue, was interesting. It is not new to attempt to regain the original form of a bagged tube by heating and driving the metal back to place. Using a roller for this purpose is unusual to me, and the question arises; is it safe?

Many boiler attendants have been scalded to death by the bursting of tubes in water-tube boilers, and many have received painful and serious injuries from

the same cause. No manner of repairing tubes should be adopted that can in any way produce a greater hazard in this class of accident. A small cylinder, of the proportions of a boiler tube as regards its thickness and diameter, may have enormous internal strains set up in it by being heated locally.

I believe that some of the apparently inexplicable tube accidents, which have resulted from the blowing out of a section of the tube, may be possibly accounted for by such repairs as described by Mr. Burns, or by continued overheating and cooling occasioned by the presence of scale. Merely being bagged makes a tube dangerous because the bagging of the metal would be certain evidence of local overheating, but the change in shape caused by the bagging would tend to produce flexibility which would reduce the amount of strain set up in the structure by subsequent cooling.

Why is the driving up of a bagged tube a benefit? If a tube is bagged so little that the condition of the metal would warrant it being set up to its original shape, there can be no special reason for the further lodgment of scale at this point, for the rapid circulation of water tends to prevent a slight depression from being different from any other portion of the tube in this respect. About the only advantage in setting up a bag will be derived from the ability to clean the interior surface of scale with the ordinary cleaning tools, but a few blows with a light hammer should dislodge the scale effectively from this part of the tube. A tube with a bag so pronounced as to be likely to give further trouble by scale lodging in it, should be replaced, and tubes less bagged are probably safest as they are. A tube which has split should certainly not be rolled or driven up, and it is almost incredible that a repair as described by Mr. Burns, would be attempted on a tube in this condition. Such a repair would render the further use of the tube hazardous, and should not be permitted.

J. E. TERMAN.

Hartford, Conn.

Why Cylinder Tops Wear

Referring to William E. Dixon's letter in the general correspondence columns of the June 11 issue, and to C. E. Anderson's letter in the refrigeration department of the June 18 issue, I would state that I have had similar trouble with a heavy mill engine.

The cause of the trouble was not far to seek. The adjustable shoe on the bottom half of the crosshead had become slack, thus letting the crosshead down on the guide with the result that the piston rod, with the crank-end stuffing-box as a fulcrum, raised the piston causing it to bear continually against the top of the cylinder. This caused noise, wear and trouble Adjustment of the crosshead shoes soon put this matter right.

I would suggest that Messrs. Dixon and Anderson examine the crosshead shoes to see that they are correctly adjusted or that the guides are not worn excessively. If the engine and compressor are "built-up" machines, the guide frame should be tested for correct alignment. It appears to be the lifting of the piston which is the cause of the trouble in both cases, and the above explanation is the best one I know of.

JOHN S. LEESE.

Manchester, England.

Water in the Low-Pressure Cylinder

H. R. Low's article in the June 25 issue, relative to this subject, was very instructive. But I would like to inquire if it is general practice to have a fixed cutoff for the low-pressure cylinder of compound engines. If not, I would like Mr. Low to tell us through POWER the receiver pressure and other results, if the governor on the low-pressure cylinder had caused its valve to cutoff early in response to the light load, the same as the high-pressure governor caused its valve to cutoff early.

B. COESRIB.

Detroit, Mich.

[F. F. Nickel's letter, page 27, July 2 issue, on the reasons for a fixed cutoff on the low-pressure cylinder of compound engines bears on this question.—EDITOR.]

Centrifugal Pump Troubles

Under this heading in the June 25 issue, F. Webster uses the formula,

$$F = \frac{WV^2}{Rg} \qquad (1)$$

for finding the centrifugal force. This formula is correct, but why not use the shorter one,

$$F = \frac{WRN^2}{35,196} \qquad (2)$$

which gives the same result?

In the letter referred to

$W = \frac{1}{32}$ lb.;

$N = 3600$;

$R = 10$ in. $= \frac{5}{6}$ ft.

Substituting these values in (1) gives;

$$F = \tfrac{1}{32} \times (3.1416 \times \tfrac{5}{6} \times 2 \times 3600 \div 60)^2 \div \tfrac{5}{6} \times 32.16 = 115 \text{ lb.}$$

Substituting in (2) gives;

$$F = \tfrac{1}{32} \times \frac{10 \times 3600^2}{35,196} = 115 \text{ lb.}$$

The last operation is evidently much simpler.

I have seen formula (1) used so frequently in the technical press that it seems as though few were familiar with the shorter one.

E. D. GAGNIER.

Salem, Ohio.

Babbitting a Large Engine Bearing

The job described by J. M. Burns under this heading in the June 25 issue can be done much easier and quicker. By wrapping the shaft with one thickness of strong brown wrapping paper, the different sections of the bearing can be put in place one at a time. They can be heated before placing in position, and, under ordinary circumstances, will retain enough heat to make the babbitt flow properly.

The paper around the shaft prevents chilling the babbitt, and insures a smooth surface. It also gives just about the right allowance for oil room, so that about all the finishing necessary is making the oil grooves, and trimming the edges. I usually wrap some asbestos wick around the shaft at the ends of the bearing while pouring the babbitt, as it helps to hold the paper smooth and prevents the babbitt from running out. The principal advantage of this method is the positive fit of the bearing, as it is cast in the exact position in which it is to run.

A. G. KNIGHT.

Omaha, Neb.

Sudden Contraction Ruptured Boiler

Mr. Rider's question on this subject in the July 2 issue may be explained as follows:

After the water was blown out, the dry boiler cooled so rapidly that the abnormal contraction caused the rupture.

A similar accident occurred in our plant. A boiler tube failed and let the water out of the boiler in less time than it takes to tell it. The boilers are of the vertical type, with 4-in. tubes, 22 ft. long. The room was so full of steam I could not see where to go for several minutes. After it cleared away we found the fire and ashes blown entirely off the grates out of the arch door, and on the floor. After the boiler had cooled down we found a tube had opened at the seam 31 in. up from the bottom and 50 in. long; it was open flat at the center of the rupture. The crown-sheet was raised up an inch from the top of a few of the middle tubes.

We found water in the bottom of the boiler up to the opening of the tube. We believe that the crown-sheet was so suddenly left exposed to the fierce heat at the top that it warped up and away from the tubes.

A. I. HAIGHT.

Kalamazoo, Mich.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Constancy of Governor Action

Do the governor rods which control the cutting off of a Corliss engine, stand stationary when the engine is at work?

E P.

In all reciprocating engines the rotative effort transmitted to the crank is constantly changing, both from variation of pressure acting on the piston and change of angular position of the connecting-rod with respect to the crank, and, although the weight of the flywheel may make the rotation sufficiently uniform for all practical purposes, it is made up of accelerations and retardations during every revolution and the velocity cannot be maintained absolutely uniform throughout a revolution without employing a flywheel of infinite weight. The governor is therefore subjected to constant fluctuations of speed, and the governor rods cannot stand stationary unless there is so much lost motion that they are not moved in obedience to the centrifugal influences which act on the governor.

Advantages of Centrifugal Pumps

What are the advantages of centrifugal over reciprocating pumps, and more especially as boiler feeders?

W. W. D.

The leading advantages of centrifugal pumps are compactness, silent running, durability and superior economy in cost of power, attendance and repairs, and the facility with which they may be adapted to any location where they may be supplied with power direct connected to an electric motor or steam turbine.

As boiler feeders they have the advantage over reciprocating pumps of continuous delivery without shock or hammering, and of producing no excessive pressure on feed mains for any adjustment of feed stop valves or other stoppage of pipe connections.

Placing Elliptical Manholes

Why are elliptical manholes in the cylindrical part of drums and boilers laid out with the length across the shell?

E. L. V.

The stress per square inch which is produced girthwise by a bursting pressure within a cylindrical shell is twice as great as the stress produced longitudinally. Elliptical manhole openings in the cylindrical part of a shell, therefore, should be made with their length across the shell and with narrowest dimension longitudinal with the shell so as to retain the greatest amount of the material of the shell which is capable of girthwise resistance.

Loss of Heat from Throttling

Is any loss of heat sustained by passing steam through a throttle or pressure-reducing valve, and if not, what becomes of the greater amount of heat and higher temperature which the steam had before reduction to lower pressure?

A. B. E.

There is practically no loss of heat. If, after passing through the valve, the steam at the reduced pressure is brought to rest, or to a reduced velocity, the heat employed as energy during expansion to the lower temperature is reconverted into heat which is imparted to the steam as superheat in its new condition, or is employed in evaporation of entrained water carried over from the higher pressure. While there are, therefore, as many heat units per pound in the throttled as in the higher pressure steam, a lesser proportion of this heat is available for power-producing purposes. If the steam were wiredrawn to atmospheric pressure, each pound would contain the same number of heat units as it did before it was throttled, but they would be of no use in a noncondensing engine.

Tremor of Boilers

What causes some boilers to tremble while in use? Is the tremor a serious menace to the safety of the pipe connections, and if so, how can the annoyance be prevented?

J. R.

Tremor of boilers is usually produced by the reaction of intermittent discharges of steam causing main steam pipe lines to recoil at bends and elbows. Some motion thus set up in pipe lines is inevitable and is most likely of occurrence to an annoying extent when a pipe line supplies high-pressure steam to a single engine which has a sharp cutoff, the vibrations often extending not only to the boiler but also to the boiler setting. Connections and fittings are undoubtedly weakened when thus strained, though as a rule they are not seriously damaged.

Tremor of a boiler from vibration of a pipe line may be almost entirely overcome by providing substantial abutments and anchorages at elbows, but these should be so arranged as not to interfere with expansion. The initial recoil can be almost entirely prevented by placing a large enough receiver in the steam pipe near the engine.

Force of a Blow

How many pounds blow will be delivered by a ram for driving bolts out of locomotive frames, the weight of the ram being 62 lb., its speed 400 ft. per min., length of stroke 8 in., diameter of cylinder 6 in. and air or steam pressure 90 lb. per sq.in.?

J. A.

The pressure to which a blow would be equivalent, for instance, as pounds pressure required for removing a bolt from a locomotive frame, cannot be estimated. When one body exerts an impact on another, the body which was first put in motion does not come to rest until the work which was stored up in it has been transmitted in many ways that are incalculable, such as in setting up motion in the body struck, in abrasion, in rebounding in its own path, in developing heat, etc.

Weight of Water

What is the weight of water per cubic foot at ordinary temperatures? What weight and number of gallons per cubic foot are usually assumed in calculations of water supply, and what weight per gallon?

T. W.

The weight of 1 cu.ft. of pure water at

32	deg. F. (freezing point)	= 62.418 lb.
39.1	deg. F. (greatest density)	= 62.425 lb.
62	deg F. (standard temperature)	= 62.355 lb.
212	deg F. (boiling point under 1 atmosphere)	= 59.76 lb.

In ordinary calculations it is usual to assume that fresh water weighs 62½ lb. per cu.ft.

An American gallon is equivalent to 231 cu.in., there being 7.480519 gal. per cu.ft., which for purposes of calculations is taken as 7½ gal per cu.ft. A gallon of fresh water at 62 deg. F. weighs 8.3356 lb. and is usually taken as 8⅓ lb.

Regulation of Wick Oilers

What is the best way to check the flow of oil from a wick oiler?

J. T. M.

By lateral compression of the wick or by removing some of the strands of the wick.

Study Questions

This Week's Questions
Last Week's Answers

(81) What is the temperature of the mixture when 36.71 lb. of condensing water at 50 deg. F. is used per pound of steam at 216 deg. F.?

(82) Neglecting friction and inertia, what work is done in moving a piston 3 ft. in diameter through a distance of 5 ft. in a horizontal cylinder, against atmospheric pressure?

(83) Two shafts are 18 in. apart. The driving shaft runs at 192 r.p.m. and the driven shaft is to run at 240 r.p.m. in the opposite direction. Required the pitch diameters of a pair of gears connecting the two shafts.

(84) How far apart should the ends of 60-ft. steel rails be laid at a temperature of 32 deg. F. so that they will just touch at a temperature of 98 deg. F.?

(85) With the same temperature range as in the last problem what compression stress per square inch would be created if the rails were laid touching when cold?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(76) The area under shear is an annular one corresponding to the lateral surface of a cylinder the diameter of the bolt and the height of the head, or

$$\pi DL = 3.1416 \times 1\tfrac{1}{4} \times \tfrac{3}{4} = 3.5343 \text{ sq.in.}$$

The shearing stress per sq.in. is, therefore,

$$\frac{12{,}000}{3.5343} = 3395.3 \text{ lb.}$$

(77) The area of an ellipse is found by the formula

$$A = \pi ab$$

where

$\pi = 3.1416$;
a = Half the major axis;
b = Half the minor axis.

Then

$A = 3.1416 \times 12 \times 8 = 301.5936$ *sq.in.*

(78) By the relations of similar triangles their bases are proportional to their altitudes. Then the height of the stack is to the length of the rule as the man's distance from the stack is to the distance of the rule from his eye, or

$$x : 12 :: 162 : 18$$

$$x = \frac{12 \times 162}{18} = 108 \text{ ft.}$$

(79) The volume of the cylinder is its cross-sectional area ($0.7854D^2$) multiplied by its length expressed in the same units or

$$(74)^2 \times 0.7854 \times 7\tfrac{1}{2} \times 12 = 387{,}076.5 \text{ cu.in.}$$

or

$$\frac{387{,}076.5}{1728} = 224 \text{ cu.ft.}$$

Then

$$\frac{224}{1670} = 0.134 \text{ cu.ft.}$$

of water required and as 1 lb. of water occupies about 0.16 cu.ft.

$$\frac{0.134}{0.16} = 0.8375 \text{ lb.}$$

(80) The total heat evolved in burning 1 lb. of carbon to CO_2 is 14,500 B.t.u., and in burning it to CO is 4400 B.t.u. Then

$$14{,}500 - 4400 = 10{,}100 \text{ B.t.u.}$$

more heat is realized in burning 1 lb. of carbon to CO_2.

Daly and Doolin

By Danny Hogan

"It's good sound laws we nade previntin' ther loikes of yer, Doolin, from puttin' onsafe b'ilers inter ther control ov ther ign'rant an' unsusphectin' buyer. Now, wait a minut! Considher ther dacint, wholesome laws made by Massyshusetts, Ohio and Detr'it, lavin' be ther other pla-aces, what forced yez fellas ter build betther b'ilers than ye have been buildin'. An' I'm tellin' yer, Doolin, ther credit ov ther matther b'longs t' ther engineerin' soci'ties—not ther b'ilermakers. Thim laws is sphreadin' over ther entoire land, an' in ther long run 'tis iviry state as will adopt thim. An' we did ut, Mister Doolin—we, ther engineers!"

"Thrue fer ye, thrue indade," replied Doolin. "'Twas done by yez engineers; that can be proved by ther mishtakes ye made.

"As a case in p'int, cast yer oiye over ther rule on convex heads that was sthuck in ther U. S. rules—an' copied by Massyshusetts an' ther rist of ther intilligint engineers as is manufacturin' rules. Wan flash ov yer oiye will convince ye no b'ilermaker was iver ashked t' revoise ut.

"Now, listhen, Daly, an' howld yer whisht. Wid a wather-chube b'iler havin' a forty-eight-inch drum an' havin' a tinsile strength ov 60,000 lb., ye'd git a t'ickness a bit over foive-sixteenths fer a head widout a manhole.

"If ye'll sthick that hairless knob av yez inter ther manhole av a noomber six wather-chube b'iler wid a forty-eight-inch drum, ye'll obsarve that ther b'ilermaker was daffy enough, or had gray matther enough, ter put in a head double ther t'ickness ther rules hollers fer.

"Thrue, they had ther rules, an' knowin' ye foolish engineers wuz ign'rant ov rules an' their fal-lacies, ther nee-farious an' aqually ign'rant b'ilermaker seen his dooty an' done ut widout troublin' ter argefy ther matther.

"Av coorse, Daly, ye moight remonsthrate, whim ye put in ther three new wans, over ther woild waste ov mateerial in thim heads, an' save ther ol' man a few paltry dollars. But will ye?

"An' by ther same token, Daly, whin a-are yez engineers iver goin' t' sthop thot divil's own deed ov schrewing down pop safety valves? Thot ice-house slaughter in Massyshusetts an' thot locomotive massacree in San Antonio—Oh, ther black shame ov ut! Thim two woild deeds alone is all ther raysons in ther worruld why ivery engineer—ther blue an' ther gray, ther licensed an' ther unlicensed—should be compilled t' reshtrain their mad desoire ov monkeyin' wid safety valves. Phut is ut timpts thim annyhow, Daly? Is ut a mic-crobe, a disayse, or phut?

"Sthuff this in yer poipe! Whin thim b'ilermakers wroites a buk ov b'iler rules thay'll provoide a pinalty fer tamperin' wid safety valves (if ther offinder isn't killt in ther manetoime) and ther rule will mar-rch soide be soide ov a betther rule for convex an' concave heads.

"Now, run this t'rough yer turboine, phile O'im drivin' up ther bag on ther pump-house b'iler."

Opportunity for Technical Instruction

The Ohio State University offers an exceptional opportunity to secure a two years' course in industrial arts, which with practical experience enables a student to enter without an examination. The course includes in the first year, elementary algebra, plane and solid geometry, trigonometry, mechanical and shop drawing, pattern making, forging, founding, bench work; in the second year, mechanical sketching, descriptive geometry, mechanisms used in machines, machine-tool practice, and elementary physics.

If any of the subjects are known, another may be substituted. This gives two years of good foundation study and practice with personal teaching under experienced and practical men.

The work begins about the middle of September and closes about the middle of June. This gives the student a chance to secure a position during the summer to help pay expenses the next year which can be kept as low as the following: University fees, $30; books and stationery, $15; room, $35; board, $100; laundry, $15; miscellaneous, $20; total, $215.

Anyone wishing to enter this course should write to Prof. F. E. Sanborn, Ohio State University, Columbus, Ohio, state his position and ask for any further information he desires.

What to See in Kansas City

As is natural in a rapidly growing city, many of the power plants in Kansas City are new and most modernly equipped. A few typical plants are illustrated and described in this issue. To the water-works man the pumping stations will be of interest employing, as they do, two types of units: centrifugal pumps driven by compound Corliss engines for the low-head stage, and the standard type of compound- and triple-expansion high-duty units for the high-head stage. The combined light and power plant and pumping station of Kansas City, Kan., at Quindaro is of especial interest as it represents a new type of municipal plant. At the time our article on the water-works was written, the equipment for the light and power plant had not been installed. But the plant will probably be in operation by the time the convention is held and a visit to it should be interesting.

The Missouri River generating station of the Metropolitan Street Railway Co., also described, is also of interest, as it possesses some unusual features of design. Here the visitor will find two types of turbine and a group of reciprocating engines representing the progress made in the design of prime movers during the last decade. The arrangement of the boilers in two tiers, the duplicate system of steam piping and the circulating-water system are other interesting features.

For the man interested in the smaller plant we present two articles. At *The Kansas City Star* plant the direct-current system is employed. Here the most interesting feature is the system of electric drive and control for the printing presses. At the National Biscuit Co. plant alternating current is employed. While this plant contains nothing particularly unusual in its design or equipment, the feature which makes it worthy of a visit is its exceedingly clean, well kept condition and the ideal arrangement from the standpoint of physical comfort for the operating force.

Numerous other plants throughout the city deserve a visit from interested people and only lack of space prevents us from publishing articles concerning them. However, those who attend the convention and wish to visit or secure information about any plant or type of plant in the city may do so by applying to the reception committee or at the POWER booth in the exhibit hall.

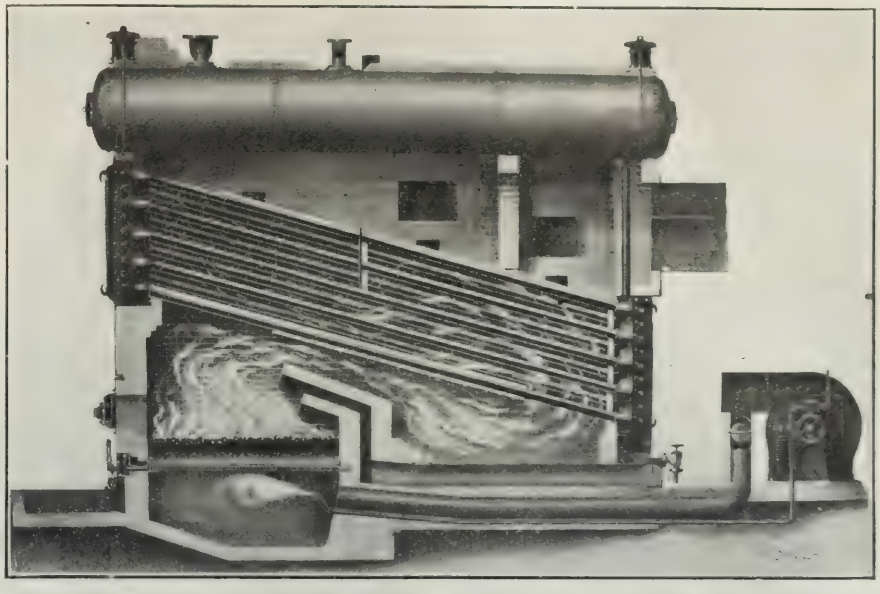
No article on Kansas City is complete unless it contains some reference to her splendid park, parkway and boulevard system of which she is justly proud, and no visit to the city should be concluded until at least a part of the system has been traversed. Over 50 miles of boulevards, parts of which are exceedingly beautiful, connect the 20 or more parks scattered throughout the city. The latter vary in size from one city block to 1350 acres. Perhaps the most noted section of boulevard is the Cliff Drive in North Terrace Park, which winds about a natural ledge under a densely wooded overhanging bluff that overlooks the Missouri River 200 ft. below and the country on the opposite side of the river.

Parkison Patent Furnace

With many grades of coal, especially those of the lignite variety where the moisture, volatile and ash are high, it is difficult to obtain efficient combustion. The Parkison patent furnace, illustrated, is especially designed for use with low-grade fuels.

Its five distinctive features: large ashpit which closes air-tight; individual rocking water-cooled grates; deep firebox accommodating a fuel bed 14 to 24 in. deep; primary air system supplying air to the ashpit under the fuel bed together with the required moisture, usually in the form of exhaust steam from the fan engine, and a secondary air system supplying preheated air over and beyond the fuel bed.

The primary air is obtained from an engine-driven blower, the exhaust from the engine of which is utilized in the ashpit. This air is introduced at a pressure just sufficient to force it through the fuel, which depends largely upon the coarseness of the coal. The draft over the fuel is generally maintained at 0.1 to 0.13 in. water gage. The quantity of the secondary air is regulated by the stack damper and the damper in the supply pipe, as shown. This secondary air is preheated in flowing between the two firebrick arches, as shown.



SECTIONAL ELEVATION OF THE PARKISON PATENT FURNACE

The manufacturers describe the operation of the furnace as follows:

The fuel bed may be considered to be divided into four zones: First, the upper layers of fresh or green fuel from which the volatile matter is being distilled. Second, directly beneath this a bed of fixed carbon burning with insufficient air to produce complete combustion and producing CO. Third, a layer of incandescent fixed carbon burning with sufficient air to produce complete combustion or CO_2. Fourth, a layer of ash or residue. These zones are not defined but blend gradually from the layer of fresh coal at the top to the stratum of clean ash at the bottom. Only enough primary air is admitted to completely burn the fixed carbon in the lower part of the fuel bed. Some of the CO_2 thus formed recombines with the carbon higher up in the fuel bed to form CO which passing upward joins the volatile gases.

The steam admitted to the ashpit, coming in contact with the hot clinkers, serves to soften them so that they may be readily broken up and discharged through the grates. This steam, in passing up through the incandescent fuel bed, is dissociated by the heat into hydrogen and oxygen. These both unite with the carbon, the hydrogen forming water gas, which passes up through the fuel and mingles with the distilled volatile matter in the combustion chamber. Although this water gas does not liberate any more heat in its combustion than was required to cause its formation, it has no derogatory effect.

The secondary and preheated air is admitted to the furnace across the entire width of the arch, as shown, and mingles directly with the combustible gases coming up from the fuel. Under suitable regulation, complete combustion of these gases then takes place rapidly and without the retarding and cooling effect caused by the admission of a supply of cold air.

This furnace is manufactured and installed by the Western Furnace Co., Gas and Electric Building, Denver, Colo.

Motor Control Panel

Where motor-driven fire pumps are used the motor and control apparatus must be absolutely reliable.

A fire-pump motor-control panel has been designed for emergency conditions by the General Electric Co., Schenectady, N. Y. All operating parts have bearings of noncorrosive material so that they cannot rust on account of long disuse.

The complete panel is inclosed in a splash-proof case which protects it from water. The panel is mounted on four pipe supports to place it above any accumulation of water in the basement where the panel is generally located.

The pressure governor, connected to the water-pressure system, closes the control circuit of the panel as soon as the water pressure falls to a predetermined value, starting the motor with its starting resistance in circuit. Current limit contactors automatically accelerate the motor by cutting out successive steps of this resistance.

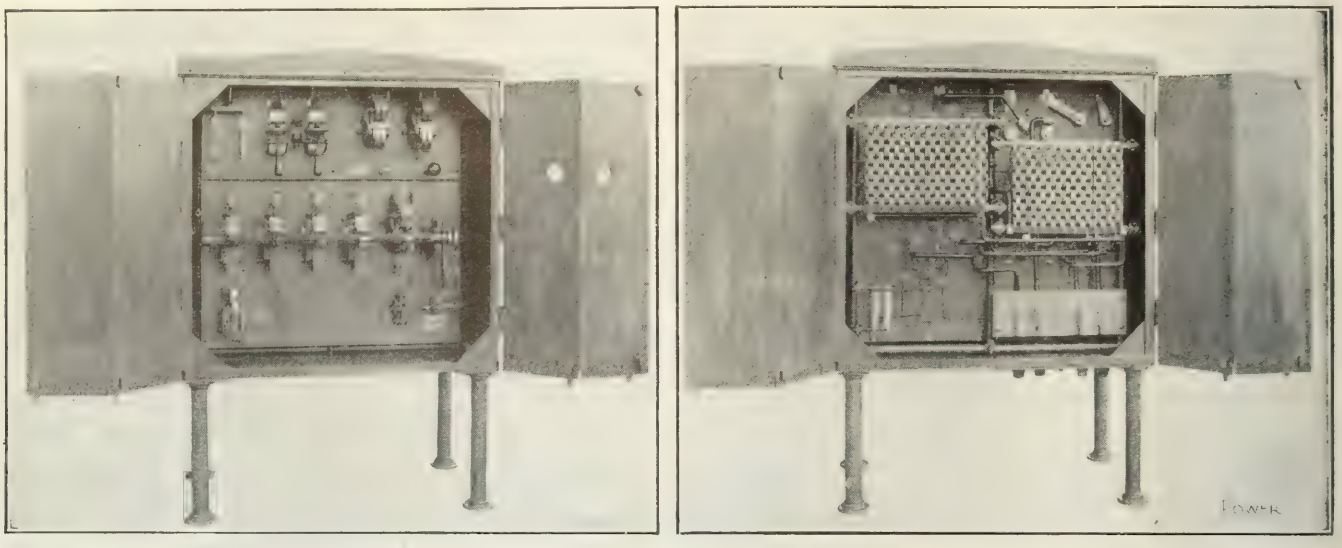
The motor may be started by hand by the operating lever at the right of the contactors. This forces the contactors to close in their proper sequence by a camshaft.

The panel is fully protected against failure of voltage and also from overloading. Two lamps are mounted on the panel, one indicating when there is voltage on the line, the other that the motor is running. The lights are visible through glass bullseyes in the inclosing cover.

Tne contactors which open the main-line circuit have powerful magnetic blowouts which snuff out the arc when the circuit is opened and prevent burning the contactor parts.

A similar panel has been designed for controlling slip-ring induction motors, where automatic control is obtained by a motor-operated dial switch. The latter may also be operated by hand.

These panels may be used to advantage in hotels, office buildings, stores, etc., for supplementing the city service and insuring sufficient water pressure for everyday use in isolated places where electric power is available.



FRONT AND BACK VIEWS OF AUTOMATIC CONTROL PANEL

Broken Crank Rod Wrecked Engine

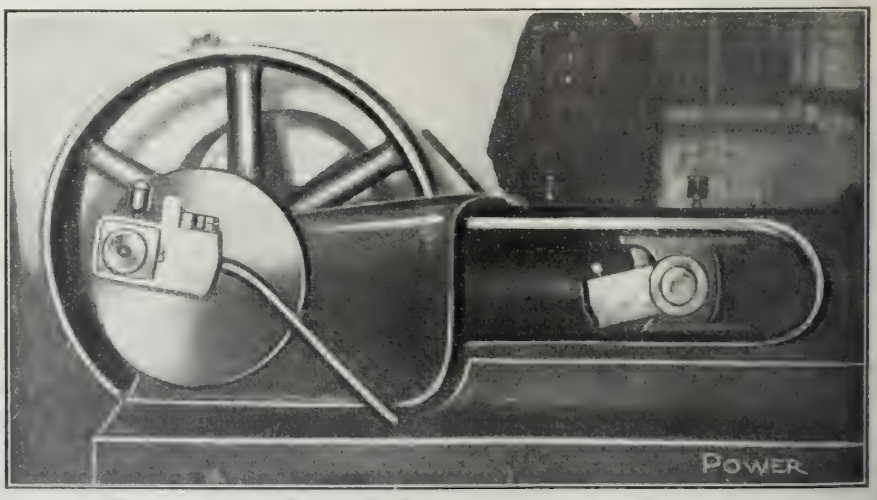
BY C. A. SCOTT

An engine wreck, in which fortunately no one was hurt, was recently witnessed by the writer. It was a 65-hp. slide-valve engine running at 170 r.p.m., and exhausted to the atmosphere. A 48 and 10 by 2-in. balance wheel and a 24x12-in. driving pulley were mounted on a crankshaft extending about 6 ft. to an outboard bearing.

The rod bent close to the crankpin two years ago, but the engine had continued to run about two months each year. Recently a bend was noticed at the crosshead end of the rod which gave the rod the appearance of an elongated letter S made backward. Both bends got worse, and the crosshead end began striking the guide. The engineer chipped away the plate and stopped the pound, but did not like the looks of things and requested the superintendent to shut down and either have the rod straightened or a new and heavier one put on, but the superintendnt could not see his way clear to stop the canning factory.

Next a click developed in the head end of the cylinder which afterward proved to have been caused by a loose nut on the piston rod. Then there was a smash. The superintendent, who stood near the throttle, closed it, and when the steam cleared away the engine was wrecked as the illustration shows.

The rod let go with a clean square break, and as the crank went over dropped down past the guide plate against the frame, doubled up and bent the crankshaft. The piston went through the end of the cylinder as far as the crosshead and guides would let it, stripping off the lower shoe and breaking the cylinder head into scrap. The frame was cracked all around the cylinder end and the engine was practically a complete loss.



ENGINE WRECKED BY BROKEN CRANK ROD

Engineer of Tests Examination

The United States Civil Service Commission announces an examination on Sept. 21 to secure eligibles to fill a vacancy in the position of engineer of tests at $2000 per annum in the Watertown Arsenal, Massachusetts, and vacancies requiring similar qualifications as they may occur. An applicant must be a graduate in civil or mechanical engineering with not less than three years' experience subsequent to graduation. He must also have had experience in metallurgical work, in the making of investigative tests of engineering materials, in the operation of testing machines, and in the interpretation of results.

Competitors will not be assembled for any of the tests, but will be rated according to their training, experience, and original investigations, as shown in application and examination Form 1312, and corroborative evidence. Applicants must furnish the vouchers of two persons able to testify from their personal acquaintance to the fitness of the applicant. It is desirable that these persons belong to the same profession, or are pursuing the same line of work as the applicant. The age limit is 25 years or over. The applicants must be citizens of the United States.

Applicants should at once apply for Form 1312 either to the United States Civil Service Commission, Washington, D. C., or to the secretary of the board of examiners, in their district. No application will be accepted unless properly executed and filed with the commission at Washington prior to the hour of closing business on Sept. 21, 1912.

Merger of Rhode Island Interests

The formal merger of the four electric and gas companies in Pawtucket and Woonsocket, R. I., into the Blackstone Valley Gas & Electric Co. will take place next month. The stockholders of the Pawtucket Gas Co. and the Pawtucket Electric Co. will meet on Sept. 16 to confirm the action of their directors taken on Aug. 16, on selling the property and franchises to the Blackstone Valley Gas & Electric Co. The Woonsocket Gas Co. and the Woonsocket Electric Machine & Power Co. will take similar action on selling out to the $6,000,000 corporation.

Last spring the Stone & Webster interests obtained a charter for the Blackstone Valley Gas & Electric Co.

Lightning Wrecks Switchboard

On Aug. 15, at 2:15 p.m., lightning entered the switch house of the Claremont station of the Claremont Power Co., Claremont, N. H., and nearly demolished the switchboard containing seven generator and eight feeder panels which supplied current to several local industries and the street railway. The four step-down transformers which were used to step down the current from a water-power station 18 miles distant were not injured to any great extent.

The management by unusual activity was able to get the lighting service on the lines and power for the cars before darkness set in. The shops requiring power were partially supplied by 8:20 the following morning.

It is expected that the damage will approach $20,000. This company operates two stations and is one of the largest in New Hampshire.

Largest Single-phase Locomotives

The most powerful single-unit electric locomotives designed up to the present time are being built for a Swiss railway. These locomotives, ten in number, weigh 108 tons each, and have a weight on their ten drivers (adhesive weight) of 85 tons. They are capable of developing 2500 hp. at a speed of 50 miles per hour, with a possible increase of speed to 75 miles per hour, for an uninterrupted run of 1½ hours, and can exert a drawbar pull of 18,000 kg. from the standstill.

The First Power Plant in India

The first steel and iron power plant ever constructed in India, is in Kalimati, 150 miles from Calcutta. This enterprise was projected by a wealthy Parsee, planned by an American engineer from Pittsburgh, and executed by another Pittsburgh man, with the assistance of native labor only. Among the workmen were Bengalis, Punjabis and Sikhs, for whom a town accommodating 25,000 was built in the midst of a jungle. The equipment of the plant includes three horizontal turbo-generators of the Zoelly type, capable of developing some 4000 hp., which is used to drive the machinery of the Tatia Iron & Steel Co.

BOOKS RECEIVED

METHODS OF ORGANIC ANALYSIS. By Henry C. Sherman. The Macmillan Co., New York. Cloth; 407 pages, 5½x8½ in.; illustrated; tables; indexed. Price, $2.40.

MODERN ILLUMINATION THEORY AND PRACTICE. By H. C. Horstmann and V. H. Tousley. F. J. Drake & Co., Chicago, Ill. Flexible leather; 265 pages, 4½x7 in.; illustrated; tables; indexed. Price, $2.

ELECTRICAL METERMAN'S HANDBOOK. Written and compiled by the Committee on Meters, National Electric Light Association, New York. Cloth; 1076 pages, 4½x6½ in.; 730 illustrations; indexed. Price, $3, and to members, $2.

NEW PUBLICATIONS

A SHORT COURSE IN GRAPHIC STATICS. By William Ledyard Cathcart and J. Irvin Chaffee. D. Van Nostrand Co., New York City. Cloth; 182 pages; 5¼x7½ in.; 58 illustrations, tables, indexed. Price, $1.50 net.

In this book the discussion of principles is largely a summary of a previous book on "The Elements of Statics" by the same authors. Some minor additions and different examples appear in the smaller book, which is intended as a brief course in graphic statics for students of mechanical engineering.

The treatment has been limited mainly to the properties and general uses of the force and equilibrium polygons as these are sufficient for the solution of most mechanical problems met with in practice. Some attention has been given to the design of trusses as such constructions as the Warren girder and the walking beam of an engine fall under this classification. The pages devoted to the various subjects under discussion are as follows: Force and equilibrium polygons, 22; stress diagrams for trusses, 18; shears and moments from stationary loads, 32; shears and moments from live loads, 37; center of gravity and moment of inertia, 24; friction, 41. A list of 35 problems to be solved complete the book, which is clear in its treatment and should admirably serve the purposes of a short term course.

DIFFERENTIAL AND INTEGRAL CALCULUS. By Lorrain S. Hulburt. 6x9. pp. 481 + xviii; illus. $2.25, net. Longmans, Green & Co., New York.

Except an engineer use his calculus constantly, the major portion of it is soon forgotten. When needed, it is, of course, most convenient to refer to the book he used in college. If, however, the book he formerly used is lost, and a new one is to be bought, we recommend this volume for consideration.

To the engineer, it is of chief interest to know that apart from following the ordinary lines of an elementary textbook, there is also a short chapter on differential equations and also something on hyperbolic functions. There are many problems on kinematics, although none on dynamics, as that is more particularly the field of a textbook on mechanics. Of course, the geometric subjects of convexity, concavity, flexes, maxima, and minima, are all treated fully. What is also an advantage to the engineer is the omission of many long rigorous proofs; results are there, which are the things needed, why these results should be so under every possible and impossible condition of $f(x, y)$ can be omitted.

While we have written the above from an engineer's standpoint, we believe the book can still be more highly recommended from the teacher's, especially in engineering schools.

THE ELEMENTS OF STRUCTURES. By George A. Hool. McGraw-Hill Book Co., New York City. Cloth; 188 pages, 6x9 in.; 140 illustrations; indexed. Price, $1.75 net.

This book was published early in 1912 and is the first of a series of home study books to be issued by the Extension Division of the University of Wisconsin. Eventually the series will comprise texts on fuels, boilers, steam and gas engines, electricity, refrigeration and other subjects related to the power plant. The books will be produced in cloth-bound volumes and also as assignment pamphlets with loose-leaf binders.

The present work treats only of the general methods to be followed in design, serving as an introduction to all the regular structural engineering studies to follow. It requires an ordinary knowledge of arithmetic, algebra, geometry, logarithms, trigonometry, mechanical drawing, and strength of materials. The space in pages taken by the various chapters is as follows:

General considerations, 3; loads on structures, 23; principles of statics, 22; reactions, 3; shear and moment, 19; influence lines, 7; algebraic treatment of concentrated load systems, 16; graphical treatment of concentrated load systems, 12; stresses in truss members, 10; design of structural members, 10; use of steel handbook, 40; the structural shop, 3; shop drawing, 5. Considering the subjects under discussion the book is simple and direct in treatment. The numerous illustrative problems and study questions will be of great assistance in home study. If the balance of the series are as good as the first book, they should be greatly appreciated by the home student.

SOCIETY NOTE

Meriden Association No. 1, of Meriden, Conn., would like to receive catalogs and literature from all manufacturers of steam power-plant appliances. They will be filed in the association rooms for the use of the members.

In California 135,000 hp. lifts the water to irrigate 300,000 acres.

Michigan N. A. S. E. Convention

The Michigan State Association of the N. A. S. E. opened its eleventh annual convention in the auditorium of the Masonic Temple in Flint at 10:30 a.m., Thursday, Aug. 1, and continued until Saturday, Aug. 3. Mayor C. S. Mott delivered the welcoming address, State President W. E. Fuller, of Kalamazoo, responding. After addresses by members of the Board of Trade and officials of the Michigan Association, President Fuller called the convention to order.

Secretary Galbraith, of the Flint Board of Commerce, was thanked for arranging the automobile tour of the industrial section of Flint when the delegates on Friday visited the plants of the Buick Motor Co., the Champion Ignition Co., the Flint Varnish Works and the Flint Electric Co.'s power house. An entertainment and addresses by men prominent in N. A. S. E. affairs was thoroughly enjoyed.

The officers elected were, president, W. M. Gomley, Grand Rapids; vice-president, A. Miller, Detroit; secretary, E. E. Smith, Kalmazoo; treasurer, Fred Fisch, Flint; conductor, M. Fredericks, Gold Water; doorkeeper, George Maloney; trustee, W. E. Fuller, Battle Creek.

The exhibit of the Central States Exhibitors' Association was held in the basement of the temple and its 36 booth attracted much attention. The exhibitors were:

Lunkenheimer Co., POWER, Garlock Packing Co., Kenneth Anderson Manufacturing Co., American Steam Gauge & Valve Manufacturing Co., H. W. Johns-Manville Co., Osborne Valve & Joint Co., J. D. Swartout, Standard Oil Co., Wickes Boiler Co., Hawk-Eye Compound Co., Greene, Tweed & Co., France Packing Co., Crandall Packing Co., C. E. Squires Co., Buckeye Chemical Co., Garsten Engineering Sales Co., Hills-McCanna Co., V. D. Anderson Co., Keystone Lubricating Co., G. C. Regulator Co., White Star Refining Co., Diamond Power Specialty Co., Jones Underfeed Stoker Co., Crane Co., Anchor Packing Co., Coon-DeWisser Co., Lagonda Manufacturing Co., Dearborn Drug & Chemical Works, *National Engineer*, Schaeffer & Budenberg Manufacturing Co., Home Rubber Co., William Powell Co.

OBITUARY

FRANK J. MULCAHY

Frank J. Mulcahy, vice-president and general manager of the Crane Valve Co., Bridgeport, Conn., died, Aug. 19, following an operation for appendicitis. He was 59 years of age and a native of Cleveland, Ohio.

HAROLD GAY

Harold Gay, a well known electrical engineer, died at his home in Farrington, Conn., on Aug. 26.

Mr. Gay gained his first electrical experience at the Schenectady plant of the General Electric Co., and recently was a consulting engineer for L. B. Stillwell & Co., New York City. He also did praiseworthy work in the electrification of the Hoosac Tunnel and the Hudson River tubes.

MEMBERS IN ATTENDANCE AT THE FLINT, MICH., N. A. S. E. CONVENTION

Mr. Gay's family is prominent in New England history. He is survived by three brothers and three sisters.

PERSONAL

Milton H. Bronsdon, chief engineer of the Rhode Island Co., which operates practically all the street railways in the state, has resigned and will locate at Los Angeles, Calif.

George F. Murphy has been appointed Eastern sales representative of the Busch-Sulzer Brothers-Diesel Engine Co. The Eastern office will be at 30 Church St., New York, as heretofore.

W. J. Snyder, formerly with the E. M. Moore Co., has been appointed Pittsburgh agent of the Foster Engineering Co., Newark, N. J., with office in room 518, Park Building, Pittsburgh, Penn.

Adolph O. Krieger has resigned as publicity manager of the Busch-Sulzer Bros.-Diesel Engine Co., St. Louis, with whom he had been connected for a number of years, to open an office at 916 Victoria Building, St. Louis, for the sale of the Tacchella oil-burning device.

POWER

Vol. 36 NEW YORK, SEPTEMBER 10, 1912 No. 11

"To Earn More, Learn More!"

A "POWER" READER'S CONCEPTION OF AN "ENGINEER" WHO THINKS HE IS UNDERPAID

Turbo Pumps in German Water Works

By Alfred Gradenwitz

A description of the first German Water Works equipped with turbo pumps having a combined capacity of 7,062,960 cu. ft. of water per 24 hr.

The managers of the Charlottenburg Water Works were the first in Germany to realize the importance of turbo-pumps for water-work service and two years ago they installed a 1000-hp. unit with a rating of 95,348 cu.ft. per hour; three additional sets of the same size will shortly be installed.

Whereas the turbo-pumps in this case only form an extension to an existing plant equipped with plunger pumps, the provincial water-works at Inheiden, are the first German water-works designed from the beginning for exclusive turbo-pump service.

A view of the turbine-pump room is shown in Fig. 1. The principal advantages of turbo-pump service lie in the uniform strain on the pumping plant and the attainment of a practically constant pressure in the main piping. In the pressure piping, which is about 26.7 miles long, there are alternate rises and falls throughout its length. If a pressure valve should be incorrectly closed there is, in the case of plunger pumps, a danger of a sudden rise in pressure, resulting in a bursted pipe. This could not occur with centrifugal pumps, and the same applies to starting the plant.

The economy of a turbo-pump installation is not inferior to that of a water-works operated by plunger pumps. Furthermore, in spite of the somewhat higher coal bill with centrifugal pumps, the lower cost of installation, the small amount of wear, the low expenses for attendance and oil consumption represent such large savings in the sinking funds and running costs not only to cover the extra fuel cost but, under favorable conditions, even allow a considerable margin.

The Inheiden provincial water-works was installed to utilize some of the numerous springs of Upper Hesse and supply water to several small districts, principally to the town of Frankfort, whose rapid development has made it necessary to obtain its drinking water from a greater distance. The pumping station is in the center of the spring district, between Hungen and Trais-Horloff. Those acquainted only with the engine rooms of plunger-pump water-works will hardly realize that in so small a building as much as 7,062,960 cu.ft. of water daily are pumped to a height of about 262 ft.

In its general layout the engine room shows many unusual features. In place of the movement of cranks, reciprocating rods and rotating flywheels in the spacious hall are two low inclosed machines, cylindrical in form, on which, at first glance, only the slowly rotating governor shows that they are working at a high speed.

One attendant supervises the running of the machines; the quietness with which he goes about his work reflects the simplicity of the easily controlled service and its reliability.

One of the two sets runs day and night without interruption for one week when it is relieved by the other which has been kept ready for service as a stand-by unit. Later, when long experience has convinced the management of the absolute reliability of the turbo-pumping service, the working period for each machine will be extended to at least four times its present length.

Each of the two units consists of a steam turbine, the shaft of which is directly coupled to the centrifugal pump mounted on the same base. On the other side of the pump, coupled to the main shaft, is the rotary wet-air pump, but the vertical condenser pump mounted on the floor of the cellar is driven from the main shaft by a worm gear. Besides the large centrifugal pump, a small auxiliary pump with an output of 5½ gal. per second is mounted at the end of the horizontal main shaft, which receives water from the pressure chamber of the large pump, and after a later extension of the works, will be used to pump water 410 ft. to a highly situated district. As an ample supply of excellent cooling water can be obtained at a pressure of about 4-m from the Riedbach, which flows under the station, it was not necessary to install a special circulating pump.

A common base-plate for the turbine and the pumps of great strength insures the necessary stability of the entire machine set and contributes toward its steady running.

All bearing parts are supplied with oil by means of an oil-pressure pump, consisting of two toothed wheels gearing with one another and fitting exactly into the wheel case. The absence of valves, pistons, springs, etc., render the oil pump immune from breakdown. It is driven by the governor spindle to which one of the wheels is keyed. During the starting period, when the pump speed is too low to give the necessary quantity of oil at a sufficient pressure, a small oil pump for hand operation is provided. The temperature of the oil in the separate bearings exceeds 120 deg. F. The oil from the bearings is cooled in the oil cooler, from which it is again drawn by the pump, thus the quantity of the lubricating material required is very limited and requires renewing only at long intervals.

The steam turbines work with a steam pressure of 170 lb. and a temperature of 660 deg. F. measured at the inlet nozzle. The speed is 2800-3000 r.p.m. They are built with high- and low-pressure stages. The turbine casing is divided in the center; after raising the upper

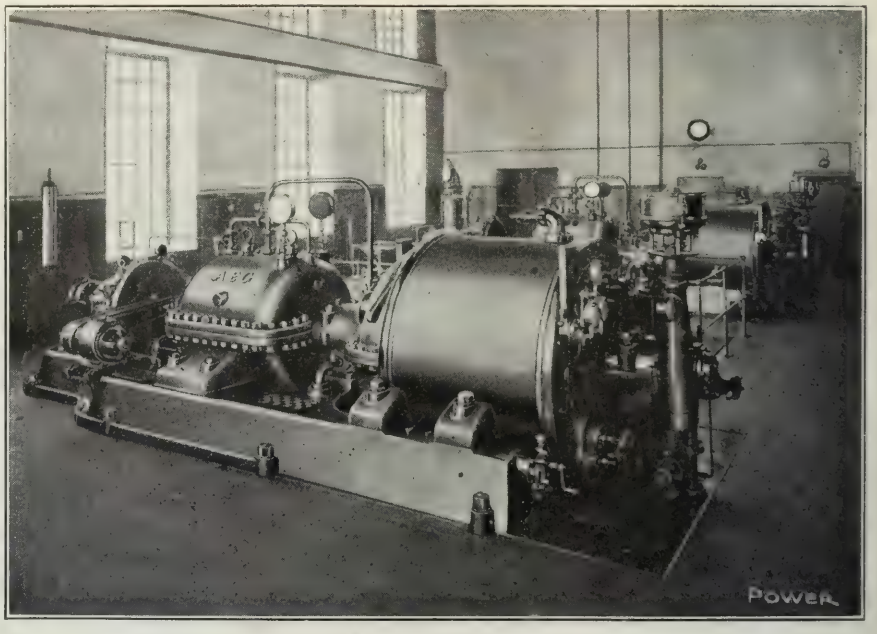


FIG. 1. VIEW OF THE INHEIDEN TURBO-PUMPING STATION

part, the wheel vanes and the guide blades are exposed, and overhauling can be done without much trouble. An undivided cover closes the turbine on the front side, on which the regulator and the inlet nozzles are placed.

The regulating mechanism is of the standard type. The vertical-spring centrifugal governor is driven through a worm gearing from the turbine shaft, and by the movement of the governor sleeve actuates a small relay valve which either admits oil to a pressure cylinder or allows it to flow out. The piston of this cylinder is rigidly connected with the actual regulating valve, which simultaneously controls the quantity and pressure of the live steam coming from the boiler. In addition to this automatic regulation, the most favorable admission for the moment may be obtained by opening or closing separate inlet nozzles by hand.

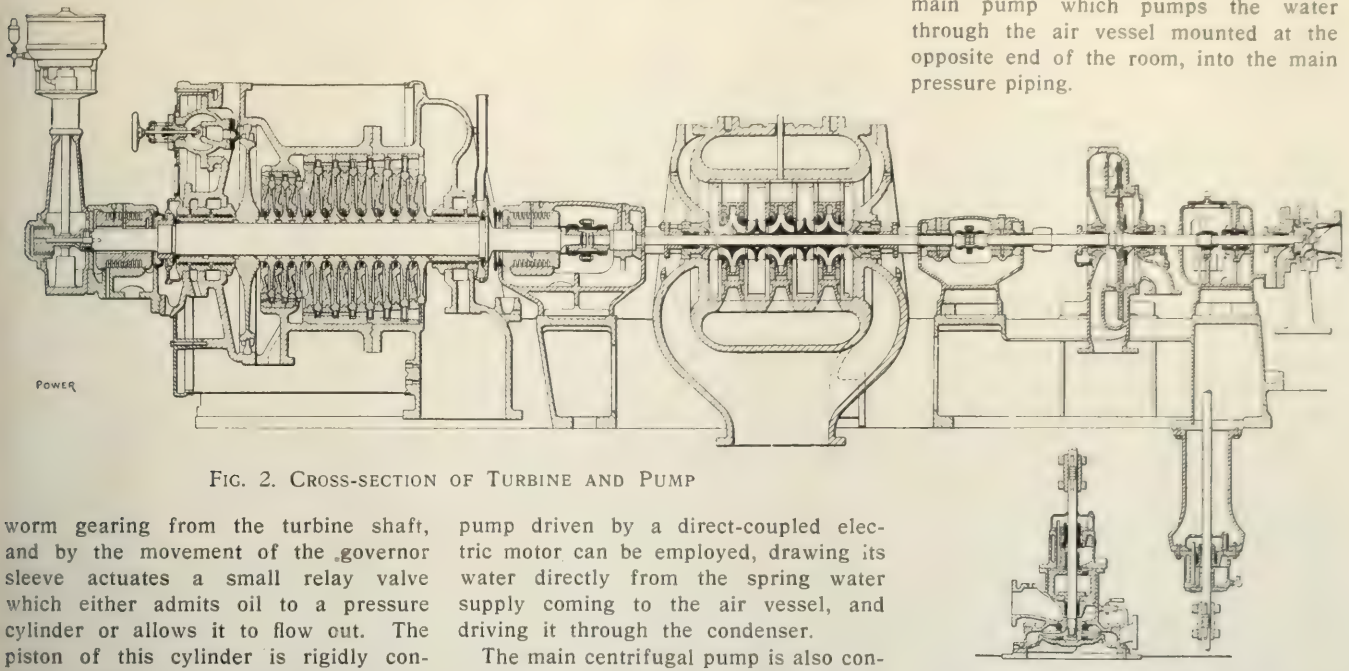


FIG. 2. CROSS-SECTION OF TURBINE AND PUMP

slightly warmed, and is constantly controlled by a thermometer having an electric contact device. If for any reason the temperature exceeds a given limit, an alarm bell indicates the fact to the attendant. If there is insufficient water in the Riedbach, or should it be otherwise impossible to obtain the cooling water from this source, a centrifugal pump driven by a direct-coupled electric motor can be employed, drawing its water directly from the spring water supply coming to the air vessel, and driving it through the condenser.

The main centrifugal pump is also constructed with a two-part casing. During the first period of service the pump, with its two impellers working in parallel, raises water to a height of 263 ft.; in the second period the auxiliary centrifugal pump will probably be employed, and will pump against an additional head of about 148 ft. Later, at a third period, on the main pump, which can be accomplished without altering the shaft or casing. A sectional drawing of the pump set is shown in Fig. 2.

From the six springs from which water is taken at the present time, a collecting pipe leads the drinking water to the air vessels installed inside the building and connected to the section piping of the main pump which pumps the water through the air vessel mounted at the opposite end of the room, into the main pressure piping.

On starting the machine, the air is sucked from the air vessel and the main pump by means of a steam ejector from which the water level of the pump can be accurately observed. When the machine is running, the air collecting in the air vessel can be drawn off by means of a hydraulic air pump, or it can be removed by a water ejector fed from the pressure piping, or from a compressor. The latter also serves to admit air to the air-pressure vessel when the water level glass shows that the water is too high. Directly in front of the air-pressure vessel, between the latter and the pump stop valves, a rigidly anchored nonreturn valve

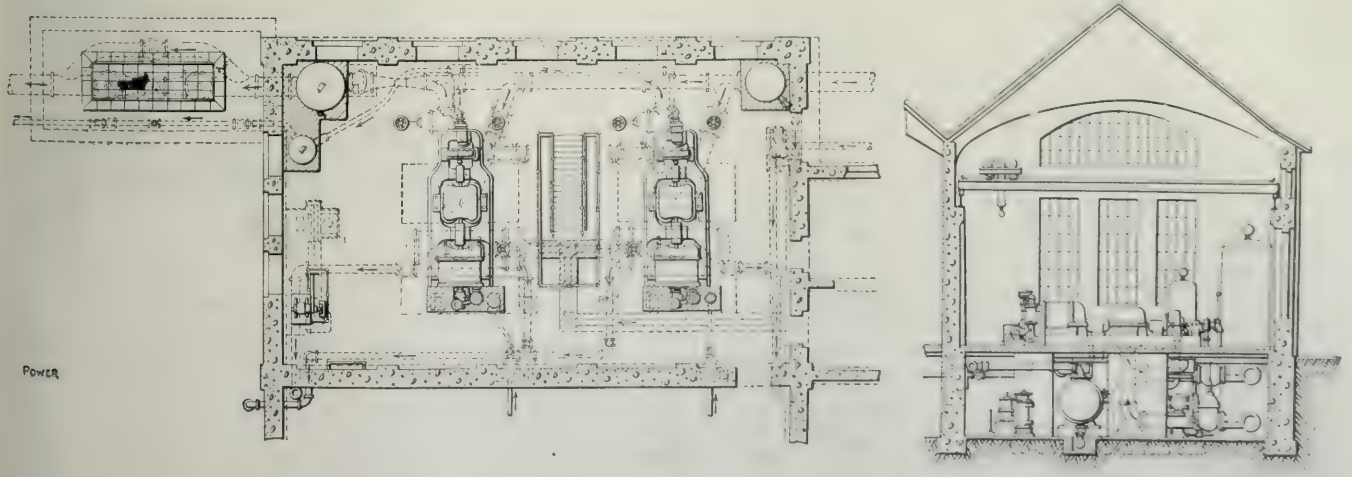


FIG. 3. PLAN AND ELEVATION OF THE PUMPING STATION

The surface condenser is installed below the turbine in the machine cellar through which flows the Riedbach. In consequence of the extensive cooling surface and the large quantity of cooling water available, the condensation is ample. Under normal working conditions the cooling water flows off again in which the quantity of water to be delivered must be increased about one-third, the main pump will work against a head of 328 ft., while the auxiliary pump will deal with a head of but 82 ft., in order to attain the necessary head of 410 ft. The higher output of the third period can be had by mounting a third impeller

is placed, provided with separate valve flaps to protect the pumps against water-hammer in the mains.

In front of this nonreturn valve a measuring nozzle constructed on the venturi system is inserted, is connected with a differential manometer mounted on the machine-room wall, from the scale of which it is possible to ascertain the exact quantity of water pumped. When taking readings regularly at given intervals, the total quantity pumped can also be thus controlled. In addition, this is carried out automatically by a meter provided with a clockwork mechanism and a remote-recording device mounted on a special measuring compartment behind the air-pressure vessel. A special system of piping with the necessary change-over valves allows the meter to be cut out or extended at any time. A second smaller air-pressure vessel having a meter with its system of piping connected to it, intended for use later with the auxiliary centrifugal pump, is already provided with the necessary connecting piping.

The compressor is driven by a belt from a small, vertical steam engine which also drives a continuous-current generator. The latter supplies the current for the exterior and interior lighting of the pumping station, and also supplies the motors operating the stokers, economizer and the repair shop. As a rule, however, the generator only takes over this work in the evening when it has the lighting load, or when the motor for the reserve cooling-water plant is in service. During the day the motors are run from the small dynamo on the base-plate of each turbine near the air pump, and belt driven from the main shaft.

Fig. 3 shows the entire plant in plan and elevation.

In the Inheiden Water Works all the advantages of the turbo-pump system are combined; the low maintenance and running costs, the small space occupied, the easily supervised service and great reliability in working.

Data necessary for estimating the economy of the service are constantly noted by self-recording instruments, as steam and feed-water meters, recording manometers, etc., and are controlled by the management. The boiler plant, the coal-conveying plant and the arrangements for an efficient utilization of the fuel are of the most perfect obtainable, and great care has been given to designing the building and its annexes.

The installation is worthy of imitation by municipal communities, and it may also be regarded as a model installation technically.

One of the recent purchases made at Cairo, Egypt, by the Sultan of Lahej, is an electric dynamo for generating light in his palace at Lahej. An electrical engineer has been engaged to look after the plant.

Massachusetts Workmen's Compensation Act

Engineers employed in the hazardous trades and their employers will watch with interest the working of the new Massachusetts law regulating and fixing the compensation for accidental injuries, which went into effect July 1. Its basic principle is that when accidents occur during industrial labor, the expense of the employee's injuries should be charged, as is that of accidents to machinery, to the cost of production.

The common-law defenses—that the employee was negligent, that he had assumed the risk of injury, or that the injury was caused by a fellow employee's negligence—are abolished by the act as defenses for those employers who refuse to come under the protection of the new law, unless the employee also refuses to accept the act. In this case the employer continues to be protected by the common law and the injured workman must seek damages in the courts as heretofore.

If the employee comes under the act, and the employer elects not to accept, which requires a written notice to the board to that effect, the employee can proceed against the employer and the foregoing three defenses will not be allowed the employer; the sole question to be decided by the court will be the employer's negligence.

That the act shall be of advantage to both employer and employee, the compensation for each kind of injury, so far as possible, is definitely stated, and is based on the probable loss of earnings of the party injured.

For minor injuries, medical and hospital services must be provided by the employer for not over two weeks. If the injury is serious, compensation must be paid, based on half the weekly wage, but not to exceed $10 nor less than $4 per week, beginning the first day of the second week and continuing during the disability not exceeding 500 weeks, or a total compensation of $3000.

If death results from injury, the $3000 is to be paid to the deceased's dependents during 300 weeks, as the employer or insurance company may decide. Additional payments shall be made for loss of both hands or feet, or one hand and one foot, or one-tenth reduction of the normal vision in both eyes with glasses, payments to continue for 100 weeks in addition to the disability compensation; loss of either hand or either foot, or one-tenth reduction of normal vision in either eye with glasses; the additional payments continue for 50 weeks; for the loss of two or more fingers, including thumbs, the period of additional compensation is fixed at 25 weeks. For minor injuries additional payments continue for 12 weeks. The employer is fined $50 for each offense or failure to report to the board, within 48 hours, each accident to an employee.

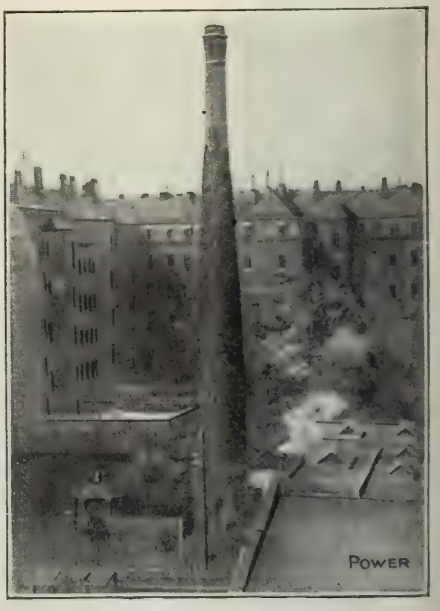
This law was urged by employers and employees, the employer desiring a reasonable and specific compensation for injuries to employees and thus avoiding law suits, and the employee desiring protection for himself and dependents when he may be incapacitated by injury.

The board has charge of the administration of the act, the interpretation of the new law, the supervision of compensation agreements and the settlement of all controversies between employers and employees. There are five members and a secretary, appointed by the governor. The board headquarters is in the Pemberton Building, Boston, where information may be had.

An Ornamental Chimney

By Dr. Robert Grimshaw

The average factory chimney is not decorative to any landscape, but its appearance may be improved by allowing creeping vines to cover its bare bricks



An Ornamental Chimney

and mortar. In Dresden, once famous only for its art collections and beautiful surroundings, factory chimneys are now numerous, except in the residential quarters; and have done much to disfigure the city.

In some instances, however, and especially where the plants are in courts behind flat-houses (as in the built-up sections) the owners have made the chimneys less hideous by the means above mentioned. The illustration is taken from the upper story of a flat building, and shows the result.

The Illinois Central Ry. has adopted a plan for the electrification of its line in Chicago; the work to be done within the next five years.

The Corliss Engine—A Lecture

By F. R. Low

Corliss did not invent the automatic cutoff nor the four-valve engine. What, then, is the distinctive feature of a Corliss engine?

The lecture tells of what Corliss improvements really consisted and gives directions for setting the valves of his engine.

I suppose there is not a man in the room who would balk at the question, "What is a Corliss engine?" and who would not be surprised when he found how difficult it is to tell. Corliss certainly did not invent the four-valve engine. All the earlier engines had separate steam and exhaust valves, and it was only a later stroke of genius which combined the functions of all four in the ingeniously simple D-slide valve. Corliss did not invent the cutoff, as many suppose. Steam was cut off and used expansively in the engines of Watt. He did not originate the idea of regulating the speed of the engine by varying the cutoff instead of the initial pressure. In one of his own patent applications he refers to a description in Tredgold's "Treatise on the Steam Engine," of an engine which had poppet valves operated by a cam under the control of the governor; to the description by the French author Armengaud, in his "Publication Industrielle," of another engine in which the regulator acted upon a cam which was made to act upon the valve rods so as to close the valves at varying portions of the stroke; and to a description in *The Practical Mechanic and Engineers' Magazine*, a Scotch publication, in 1846 of an engine with a riding cutoff under the control of the governor. These were all prior to Corliss' first patent, which was not issued until 1849.

Zachariah Allen, an engineer of Providence, patented, in 1834, an arrangement whereby the governor valve, instead of being opened more or less and throttling the pressure according to the load, was opened fully at the commencement of the stroke by a lever resting on a cam on the main shaft. The cam was wedge-shaped, and when the bearing end of the lever rested on its narrow end the valve was held open but a short fraction of the stroke. When it was changed to the wide end of the cam the valve was held open for the greater part of the stroke, and its position upon the cam was controlled by the governor. An engine of this type was put into practical operation at the Wadsworth Steam Engine Works at Providence, in 1834.

In 1842, F. E. Sickels patented the Sickels valve, so well known in marine practice. Here the *main* valve is made to do the cutting off by means of a trip-motion, the shock of the closing poppet valve being taken up by means of "cushions of yielding water," whence comes the term "dashpot," still applied to the modern substitute, which has nothing to suggest a dash about it.

Sickels was a marine engineer, and marine engines use no governor. He made his detachable or trip cutoff adjustable by hand, so that the speed could be controlled by shortening the cutoff instead of by throttling, but unfortunately for himself, though luckily for Corliss, he did not suggest in his patent specifications that this might be done automatically by the governor.

Under date of February 3, 1870, Zachariah Allen wrote to Mr. Corliss a remonstrance against his acceptance of the Rumford medal which had been awarded to him by the American Academy of Arts and Sciences for inventions that originated "the abolition of the throttle valve of the steam engine and for placing the induction valves under the control of the governor." In the course of his letter he describes a combination of his own and Sickels' invention as follows:

"You have frequently seen the large steam engine in the print works constructed by my brother, Philip Allen, near your office in Providence, and now under my charge, and are aware that it has been operated by the use of a ball regulator applied to produce a variable and detachable cutoff movement, as described in my patent and in the improvements patented by Mr. Sickels independently of the use of a throttle valve, and of any of your claims for steam-engine patents.

"This steam engine has been in operation during the whole term of 14 years of your outstanding patent claims, and is now operating excellently well, with an economy of coal, and a perfection of effective power, as indicated by diagrams, unsurpassed, as I believe, by any engine of your construction."

In the course of the same letter he says further:

"While you have originated a rotary valve, with new and peculiar devices for producing with that valve the same results which had been previously effected by the above described inventions of your predecessors, you must certainly feel that you are not the first person who promulgated or accomplished these inventions, nor the first to reduce them to practical use."

Ah! the Corliss valve, known as such all over the world. Now we have got to something essentially Corliss. But his claim even to this is not undisputed. In the early days of my connection with POWER I called upon William Wright, at Newburg, one day when the old gentleman was in a reminiscent mood. We talked of Corliss and of the time when Wright was superintendent of his shop, and the old man went to the safe, rummaged through some drawers and produced from between the leaves of an old notebook a time-worn sketch, yellow with age, and proceeded to tell me this story of the inception of the Corliss valve.

A few engines had been built in which the releasing gear invented by Mr. Corliss had been applied to different forms of slide valve with more or less satisfaction, mostly less, so far as the practical working of the valve was concerned. Meanwhile the efforts of the inventor were bent in the direction of a valve better adapted to the requirements of his cutoff, which had demonstrated its superiority as a means of regulation.

At this time an order had been taken for the Eddy Street Foundry, of which the father of William A. Harris, of the engine company of the same name, was superintendent. The frame and connections of the beam engine had been made, but the cylinder had been allowed to wait, pending the evolution of an improved valve. Mr. Harris became urgent in his demands for delivery, and on April 17, 1849, Mr. Wright approached Mr. Corliss with the question of the delayed cylinder. In the course of the conversation which followed Mr. Wright took up a piece of paper and made the sketch shown in Fig. 1. The suggestion was greeted by Mr. Corliss with the statement that if he could not have something better than that he would not have anything. Nothing better seems to have offered, however, for thousands of engines have been built in this way, and the appliance shown is universally known as the Corliss valve.

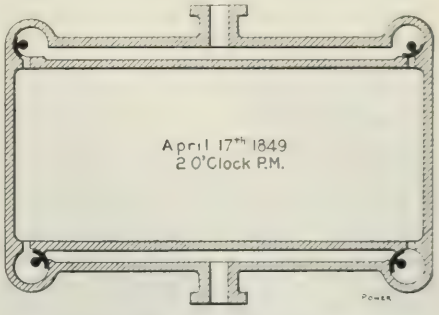


FIG. 1. MR. WRIGHT'S SKETCH

This sketch has recently been presented to me, along with other mementos of her father, by Mr. Wright's daughter, and I am thus able to show the original, creased and worn, held together with tracing cloth, and bearing upon its back, almost undecipherable, the date of its alleged production. I do not know

that any patent was ever taken out upon this valve. It does not seem to have figured in the long and bitter litigation in which the Corliss patents were involved and it was only after the death of Mr. Corliss that this sketch was brought to light, notwithstanding Mr. Wright was an opponent to Corliss in some of the patent suits.

It is probable, if the authenticity of the sketch is admitted, that Corliss found possibilities in it upon examination that failed to impress him at first sight, and that both considered it a legitimate contribution to the development of the engine by a man whose brains, as well as his hands, were in the paid employ of the company.

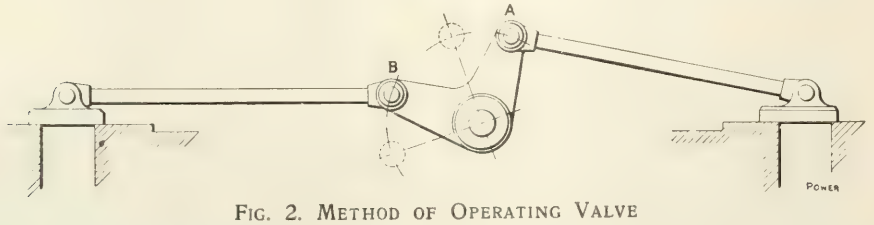


FIG. 2. METHOD OF OPERATING VALVE

What, then, is a Corliss engine, and why does George H. Corliss justly stand, in the popular estimation, by the side of James Watt as a factor in bringing the engine to its present state of efficiency?

George H. Corliss was born on June 2, 1817, at Easton, N. Y. He acquired an academical education at Castleton, Vt., and in 1844, at the age of 27, he went to Providence and associated himself with John Barlow and Edwin J. Nightingale, under the name of Corliss, Nightingale & Co., for the manufacture of steam engines. His only previous manifestation of inventive ability had been the production of a machine for sewing leather, his connection with which is doubtless responsible for the oft-repeated assertion that he was brought up a harness-maker. His senior position in the new firm was probably due more to the means and influence which he had already acquired than to any promise of exceptional engineering talent.

But Corliss was a man who went to the bottom of things. He had not been in the engine business long before he saw the weaknesses of the machine as then built, and in 1848 he applied for the first patent for his improvements. This patent was issued on March 10, 1849, and the first thing it goes into is the bracing of the frames of beam engines.

"The third part of my invention," he says in his application, "relates to the method of regulating the cutoff of the steam in the main slide valves, and consists in effecting this by means of the governor, which operates cams, so that when the velocity of the engine is too great, these cams shall be moved by the centrifugal action of the regulator, that a cam on the valve rods may sooner come in contact with them to liberate the valves and admit of their being closed by the force of springs or weights, and thus cut off the steam in proportion to the velocity of the engine."

His claim relating to this feature is as follows:

"I claim the method of regulating the motion of steam engines by means of the centrifugal regulator, by combining the said regulator with the catches that *liberate* the steam valves by means of movable cams or stops, substantially as described."

How could Corliss get this claim allowed with Allen's patent and the existing descriptions of other automatic cutoffs before the examiners? Because Corliss' was a *liberating* gear. It dropped the valve when the governor said so, and let it get closed as quickly as weights or springs could close it. In Allen's and the other cases, cams opened the valves,

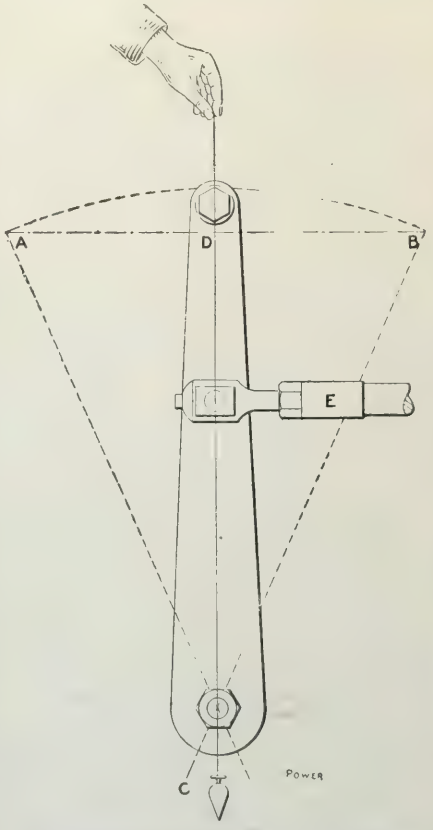


FIG. 3. EQUALIZING TRAVEL OF ROCKER ARM

and without releasing them placed them back in the closed position. In an application for a reissue of one of his patents, in 1859, he said:

"I wish it to be understood that by the term 'liberating valve gear,' as I have used it in this specification, I do not include those previous arrangements for working valves, which consist of cams or various forms to open the valves and then, at certain periods of their rotation, suffer the valves to be closed by the force of weights or springs, but control and graduate the closing movement. Such arrangements of valve gear lack the capacity for varying the periods of closing the valves, combined with the promptness of closing at all speeds of the engine, which characterizes my improved valve gear."

We have, then, as a distinctive feature of the Corliss engine the liberating, or detaching, valve gear under the control of the governor. Sickels had the detaching gear, but he had not specified it as a means of speed control nor combined it with the governor. Allen had the automatic cutoff, but not the detaching mechanism, and yet their combined inventions did not make a Corliss engine.

In the valve gear, the method and mechanism for operating the valves, considered apart from the governing feature and the particular form of the valve, lies the feature of the Corliss engine the originality and ingenuity of which are unchallenged. This invention is described in his first patent application as follows:

"In steam engines operated with slide valves, particularly the large condensing engines used in England, the valves are connected together and move together in pairs, one at each end of the cylinder, and therefore move together over the same extent of surface; and as the power required to move them is due to the friction produced by the pressure of the steam on their surface, and their range of motion under this pressure, it follows that the valves, while closed, require the most power, for much of this friction is relieved the moment the valves are partly opened.

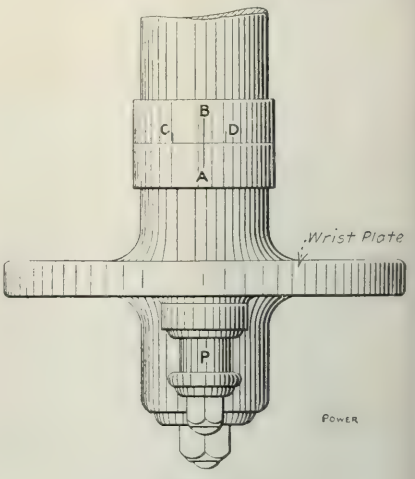


FIG. 4. WRISTPLATE

"One of the valves must always be closed while the other is being opened or closed; hence the closed valve is moved at an entire sacrifice of power.

"To save this several devices have been resorted to, such as cams, the irregular working of which makes too much noise and renders the whole liable to derangement; but by my invention I am enabled to reduce the motion of the closed valve relatively to that of the other valve, and thus greatly to reduce the amount of power hitherto required for this purpose. This, the second part of my invention, therefore consists in communicating motion to the two valves from one rock shaft,

by connecting each valve with a separate arm or crankwrist of the rocker, the two arms making such an angle with each other dependent upon the position of the valves and the rocker, so that the point of connection of the closed valves shall vibrate near the dead point and therefore give but a small amount of motion to that valve, while the other, which is being opened and closed, is moving along that part of its circuit which shall give the greatest longitudinal motion, and therefore giving to that valve the greatest amount of motion."

The meaning of this will be apparent from Fig. 2, where it will be seen that the crank in moving from its present to its dotted position would give a considerable movement to the valve to which it is connected, while the other crank would give little movement to its valve.

It is a singular fact, in view of the widespread impression that Corliss' principal contribution to the engine was the automatic cutoff, that this really distinctive feature of his engine applies with most of its force to the exhaust valves, for the idle steam valve is detached from the operating mechanism, and is not in a position to profit by its advantages. The natural development of the idea, however, was to operate all four valves from a single oscillating member, and the valve gear took the form of the four operating levers attached to the familiar pumpkin-seed-shaped wrist plate, with centers so located that the valves received their quickest movement in opening and closing, and remained practically still when the necessary movement had been accomplished.

On all but the smaller engines the motion is transmitted from the eccentric through a rocker arm, and as this motion will be distorted if this arm is allowed to vibrate to unequal distances to either side of its center of suspension, the first operation in setting an engine in which such an arm is used is to equalize its travel. This is most easily done for a horizontal engine by dropping a plumb line through the center of suspension, as in Fig. 3, turning the engine over slowly or rotating the eccentric and adjusting the stub-end E, or whatever other adjustment is provided, until the distances AD and BD to which the arm travels to either side of the center are the same.

On the stud which carries the wrist plate will be found three marks like BCD in Fig. 4, and upon the wrist plate a single mark like that at A. Hook in the rod which connects the wrist plate with the rocker arm, turn the engine slowly and adjust the stub-end upon the hook rod until the mark A goes out to both C and D, and does not travel beyond either. This does the same thing for the wrist plate which has previously been done for the rocker arm.

A diagrammatic representation of the cylinder and the crank and eccentric is given in Fig. 5. The connecting mechanism can be imagined. The eccentric is set, as you know, something more than 90 degrees ahead of the crank. Disregarding the slight distortion due to the angularity of the eccentric rod, the wrist plate will be in its central position when the eccentric is on the quarter, or in a position at right angles to the axis of the engine, at which time the crank will be approaching the center but some distance away. When the shaft has turned halfway around and the eccentric is again perpendicular to the engine axis, the wrist plate will again be in its central position, and the crank will be in the same position with regard to the other center, so that the position of the valves when the wrist plate is on the center must be the same for both ends.

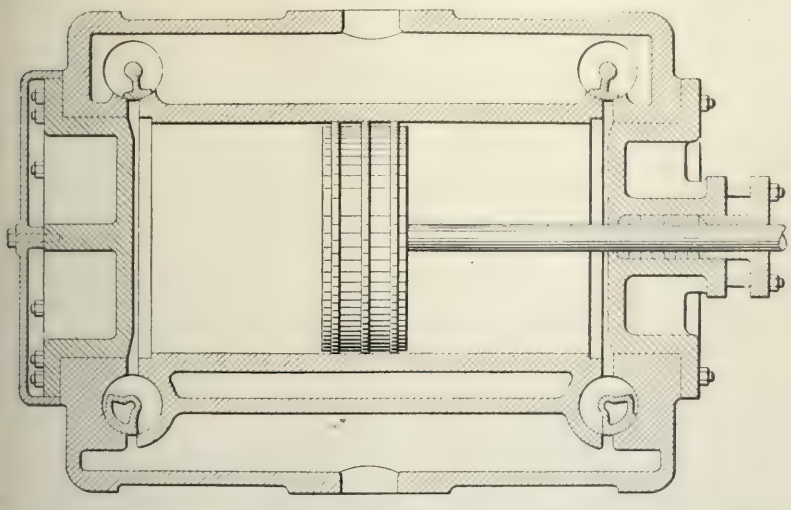


FIG. 5. DIAGRAMMATIC REPRESENTATION OF CYLINDER, CRANK AND ECCENTRIC

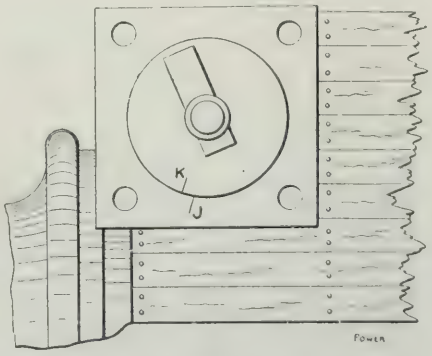
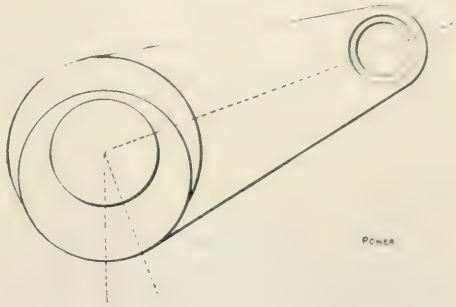


FIG. 6. VALVE WITH BONNET REMOVED

The piston has not yet reached the center; it is too early to open the steam valves, so that for this position of the wrist plate they should have some lap. With the exhaust valves we encounter conflicting conditions. The stroke is uncompleted but nearing its end. In one end of the cylinder the expanding steam must be released if we want a free exhaust; in the other the exhaust valve must be closing if we want compression. We have just seen that in the central position of the wrist plate the valves must be set alike on both ends. Now if we set this exhaust valve closed to get more compression, we must set the other one with an equal amount of lap, and may hamper the release upon that end, and if we do not set it the same it will be at the same disadvantage with regard to compression that we are trying to correct in this valve. Further, the very lap which we are putting onto this valve to get compression will be in the way when we are trying to open this valve early for release.



The best that can be done with an engine in which the motion for all four valves is taken form a single wrist plate is to compromise between the release and the compression. Set it for as little compression as you can get along with, which will be doing the best you can for the release. With a noncondensing engine a very good distribution can be gotten in this way, as neither the release nor the closure for compression has to be so early as in the condensing engine, where there is a considerable drop from the terminal to the vacuum pressure and an early release is imperative if the toe of the diagram is to be kept down and the full vacuum area realized; and where it is necessary to close the exhaust valve early in order to compress the thin vapor of the vacuum to a pressure which will arrest without shock the momentum of the moving parts. For the ordinary condition the exhaust valves of such an en-

gine should be open from a thirty-second to three-sixteenths, for engines up to 42 inches in diameter, when the wrist plate is in its central position. Tables of lap for the steam valves and of opening for the exhaust valves are published in various books, or can usually be obtained from the builder for the particular engine in hand.

If you take off the back bonnets you will find something like Fig. 6. The little marks at *J* and *K* show the positions of the working edges of the port and valve, as shown in Fig. 7. With the steam valves hooked on put the mark on the wrist plate in line with the central mark on the stud, and if your previous work has been accurate, the wrist plate will be in its central position. Now lengthen or shorten the connections between the steam valves and the wrist plate by means of the right and left connections until the distance between the marks *J* and *K* is equal to the desired lap. *K* should stand with reference to *J* so that it will move toward it on the opening movement of the valve. Some builders make their valves open toward and others away from the cylinder. Adjust the exhaust valves with the desired opening in the same way. For every material change in the length of the steam-valve connections there must be a corresponding change in the length of the dashpot rods. If they are left too short the valves will not hook up, and if they are too long the closing shoulder on the moving member of the steam-valve operating mechanism will bring up against the block before the wrist plate has reached the end of its travel, and something will have to bend or break. Make the length such that when the dashpot piston is thoroughly bottomed the valve will hook on easily and still so that there is plenty of clearance for the closing block, dividing the leeway substantially equally between the two. Now hook on the eccentric, put the engine on either center and turn the eccentric ahead of the crank in the direction in which the engine is to run until the steam valve has the desired amount of lead. If the work has been accurate the lead will be the same on the other end.

The position of the eccentric ahead of a right angle with the crank is called the "angular advance," and it is desirable to keep this as small as possible, first because the motion of the eccentric available for opening the valves becomes sluggish as it gets away from the quarter, and it is desirable to have the valve opened sharply, and secondly because by increasing the angular advance you decrease the range through which the engine can cut off. The cutoff is effected, as you know, by the contact of the releasing mechanism with the little cam, the position of which is controlled by the governor. The releasing mechanism is pushed toward this cam by the wrist plate as it moves away from its central position. If the contact does not occur on this outward movement it cannot occur at all during that stroke for when the wrist plate starts back it is drawing the mechanism away from the cam.

It is evident that this outward movement of the wrist plate in either direction occurs only during one-quarter of the revolution of the eccentric, and if we use up a large proportion of this quadrant in angular advance we shall have so much less of a range through which the governing mechanism is effective, and the earlier in the stroke will the engine commence to hang on.

All of the above applies to the admission valves. For the exhaust valves, on the contrary, considerable angular advance would often settle an awkward conflict between the release and the compression. I have pointed out before that if you add lap here to make the valve close earlier and get more compression, the lap will be in the way when you want to open the valve early and get a free release. But moving the eccentric ahead makes everything earlier and would quicken both release and closure for compression. So here is another compromise which we must make, this time between admission and exhaust. The position of the eccentric is determined, in the directions which I have just given you, by the lap which is given to the steam valves, because you set the eccentric ahead until with that lap the engine has the desired amount of lead, and this is the controlling factor in making up the tables to which I have referred.

In 1884, W. E. Crane, at that time in charge of the steam plant of Benedict & Burnham, at Waterbury, Conn., hit upon the idea of using a separate eccentric for the exhaust valves. This results in but little complication, and gives one a large latitude in adjustment, wiping out both of the compromises to which I have referred, for it divorces the steam valves from the exhaust valves altogether, and as adding lap makes the compression early and the release late while setting the eccentric ahead makes both earlier, you can get all the compression you want by lap, and set the eccentric so as to still open the valve in time for the release. In writing of this experience Mr. Crane says:

"I placed the two wrist plates in a vertical position and connected the steam valves so that each had $\frac{1}{16}$-inch lead. The exhaust valves were given $\frac{1}{8}$-inch lead. The eccentrics were placed at right angles to the crank and connected up. The steam eccentric was fastened to the shaft in that position, while the exhaust eccentric was advanced about 30 degrees.

"The engine had a cylinder 28 inches in diameter, and an attached condenser with a vacuum of 28 inches. Setting the steam valves in the above manner enabled me to get a range of cutting off from 0 to $\frac{3}{4}$-stroke, which made regulation very close, and very slight speed change in irregular work. The exhaust valves opened sufficiently early to get full vacuum at the commencement of the stroke and also had sufficient compression.

"With a noncondensing engine it is not necessary to advance the exhaust eccentric more than about 20 degrees. On a compound engine I have given the low-pressure steam valves $\frac{1}{2}$-inch lead and not found it too much. With valve stems, as usually splined, it will not be possible to set them as indicated above. If you have an engine just building, the valve stems can be splined that way in the first place.

"With stems already in use it will be necessary to take the stems out, turn them over and make new keyways. With the old style, where the blade was laid in a slot in the valve, it will require an offset key, or some other method of arranging the key, to get the above results."

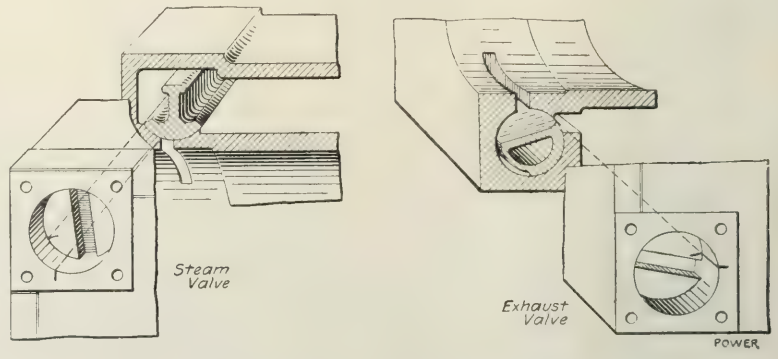


FIG. 7. STEAM AND EXHAUST VALVES

It now remains to adjust the governing mechanism. The three esentials are that the valves shall not be able to hang on long enough to give the engine steam when the governor is in its highest position in order that the governor may be able to prevent the engine from running away when there is no load, that the valves may not hook on at all when the governor is in its lowest position in order that the engine may not get steam

and run away in case the governor belt breaks or the governor becomes otherwise deranged, and that the cutoffs be equalized between the two ends on the average load. It is not possible, with the usual arrangement to so adjust them that they will be equal for all loads, for the movement of the piston, due to the angularity of the connecting rod, is different in the two ends of the cylinder, while the motion of the wrist plate, due to the much smaller angularity of the eccentric rod, is much more nearly equal on either side of its central position. The governor in descending a certain distance moves the knock-off blocks back an equal distance on both ends, and the piston has to move further in one end of the cylinder to turn the crank and the eccentric far enough to make the wrist plate push the releasing mechanism up against the cam in one end than in the other. This difference varies for different parts of the stroke, so that if it is corrected for one point of cutoff the correction will not be right for another.

The different angularity of the governor rods is also a feature. It would seem as though some simple mechanism might be interposed to make the movement of the cams vary in the same way as that of the piston, and I recommend it to you as a field for the exercise of your ingenuity. In the meantime the best we can do is, as I have said, to equalize the cutoffs for the average load. Perhaps the best way to go about it is to block the governor in its highest position, and adjust the length of the governor rods so that when the wrist plate is worked with the bar the hook will let go of the valve before the steam port gets open, which will be when the wrist plate has passed the central position a very little, for you will remember that the tables call for the valve to be lapped some when the plate is at the center of its travel.

Now if you cut off usually, as you should, at about quarter stroke, place the engine in that position, block the governor up to its normal running position, hook the wrist plate in and adjust the governor rod of the end which should just be cutting off so that the cam will be in contact with the releasing mechanism. Turn the engine over until it has made one-quarter of its stroke upon the other end, and repeat the operation. Now block the governor in its highest position and see that you have not allowed the engine to get steam under that condition. If you have, you must take care of that point. Perhaps you can do so by raising the collar which confines the upward movement of the balls. If not you must choose a higher position for the normal. If everything is all right, adjust the safety cams so that they will prevent the valve from hooking on at all when the governor is in its lowest position.

If the engine is a large one and difficult to turn over, the equalization of the cutoff can be done while running, all the better if you have an indicator. Here you simply shorten one rod and lengthen the other until the diagrams are alike. If you get either of them too short to cut off before the engine gets steam in the highest position of the governor, you must lengthen both, and look out for those safety cams.

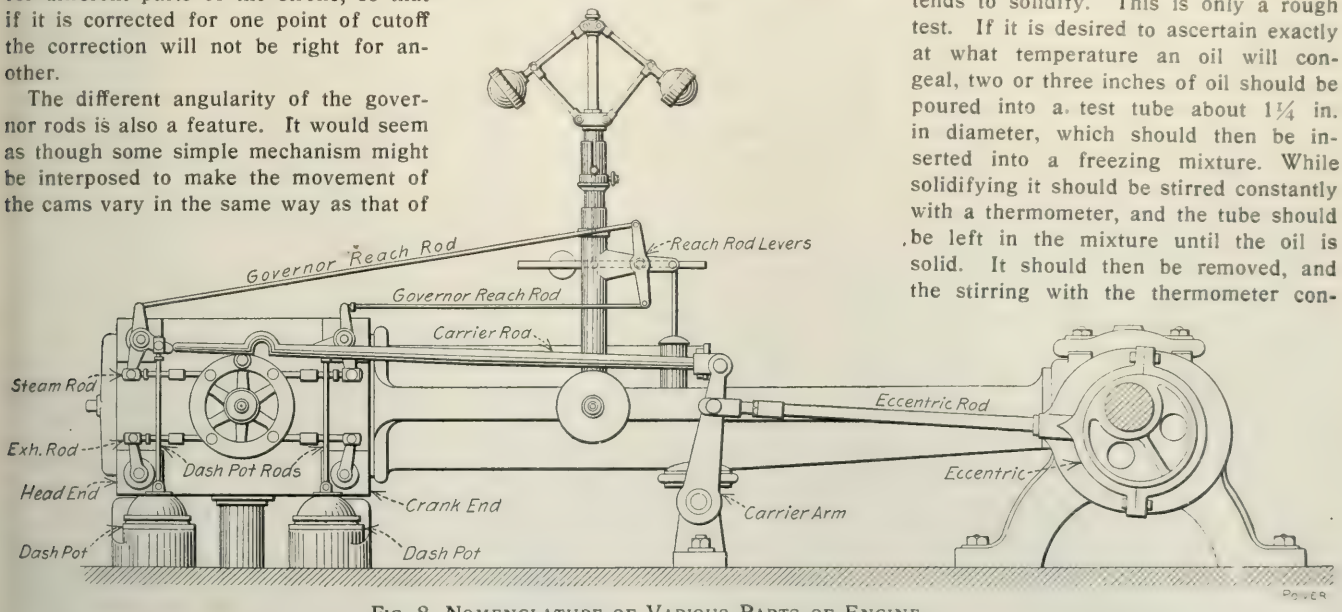


FIG. 8. NOMENCLATURE OF VARIOUS PARTS OF ENGINE

Testing Lubricating Oils

BY FRANCIS H. DAVIES

Scientific testing of lubricating oils is outside the province of the average engineer, needing as it does elaborate apparatus and special chemical training. Still, it is very desirable that the man responsible for the running of a plant should be able to make at least a few elementary tests, as oil has directly and indirectly a material bearing upon operating costs.

A simple method of ascertaining whether an oil contains acid or not is to immerse a polished copper plate in it for a few minutes. If no acid is present there will be no change in the appearance of the plate, but a trace of acid will dim the surface.

To determine whether a mineral oil is pure and well refined, a small quantity of strong sulphuric acid should be added to a sample. This will have no effect on the color of the oil if it is pure, but if it contains fatty material of any sort, discoloration will result. The addition of sulphuric acid has the effect of raising the temperature of an oil. An increase of temperature of about 20 deg. C. is the maximum that should occur with a good oil.

The solidification temperature of a lubricating oil is an important matter, and an approximate idea of the nature of any particular sample may be obtained by smearing it over the surfaces of two polished iron plates and laying them together. They should then be left for some days. If they do not move as freely over one another as at first it is evident that the oil used is one which tends to solidify. This is only a rough test. If it is desired to ascertain exactly at what temperature an oil will congeal, two or three inches of oil should be poured into a test tube about 1¼ in. in diameter, which should then be inserted into a freezing mixture. While solidifying it should be stirred constantly with a thermometer, and the tube should be left in the mixture until the oil is solid. It should then be removed, and the stirring with the thermometer continued until the oil becomes liquid and clear. The temperature of the oil when the last portion becomes fluid, is the temperature at which solidification begins to take place. For accuracy, this test should be repeated several times and an average result obtained

When testing mixed oils having both mineral and fatty constituents, an approximate idea of the proportions of the two for purposes of comparison may be obtained by mixing a small quantity of the oil with a strong solution of caustic potash. If the mixture is well shaken, an emulsion will be formed varying in proportion to the quantity of fatty matter present. This test is useful for making a rough comparison between several oils, but it does not give any indication of the relative proportions of the two ingredients in any one of the oils. With a pure mineral oil there will be at most a slight cloudiness, but saponification will be marked if either animal or vegetable fats are present.

Flue Gas Analysis for Beginners—II

By J. E. Terman

Directions for operating Orsat apparatus, adjusting level of reagents, measuring a sample and testing for carbon dioxide, oxygen and carbon monoxide.

Manipulation of the Orsat Apparatus

To become familiar with the manipulation of this apparatus, instead of a gas sample from the stack, air may be analyzed, or, to have CO_2 present in measurable quantity, a sample of the operator's breath may be used. The first operation to understand is the adjustment of the reagents in the pipettes. A line on the neck of each pipette below the glass stopcock, as at *A*, Fig. 1, indicates the proper level of the reagent after passing the gas sample into and out of the pipette. Approximating this level is sufficient, for the bore of the tube is so small that a trifle higher or lower level is practically without effect on the measurement of a sample.

For adjusting the reagents to these lines, the leveling bottle *L*, Fig. 2, is filled with water and connected to the bottom of the burette by about 30 in. of rubber tubing, then, with cocks *K* and *G*, Fig. 2, open, raising the bottle above the apparatus will fill the burette with water, when the cock *G* should be closed. Now, after opening the cock connecting the first pipette to the burette *N*, Fig. 2, the reagent may be caused to rise in the pipette to the guide line by lowering the bottle sufficiently; then the cock should be closed. Many place the leveling bottle at some fixed low point, and depend on shutting off the cocks at the

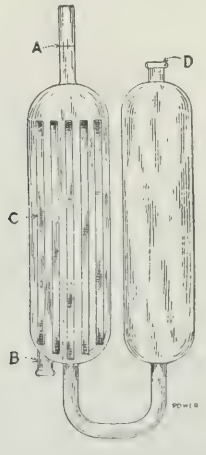


Fig. 1. Pipette with Glass Tubes in Place

right instant to prevent the reagents from running up too far in the tube, but the writer prefers the other procedure.

With the reagents adjusted in all the pipettes, the rubber bags protecting the outlets of the rear legs of the pipettes, connecting *D*, Fig. 1, should be removed for an instant to restore the atmospheric pressure within this portion of this pipette where a partial vacuum was produced during the adjusting of the reagents. For the last two pipettes, the rubber bags will have to be removed several times, as the reagents in them absorb oxygen from the air. This persistent reducing of the pressure will interfere with subsequent manipulation of the gas sample unless the pressure is repeatedly equalized.

Measuring a Sample of Gas

The next step is to learn to measure a sample correctly. A sample of breath to be analyzed may be collected in one of the sample bottles, Fig. 3, by opening the pinch-cock *Q* and holding the end of the tubing at *P* in the mouth, then opening the pinch-cock *P* and blowing into the bottle to replace the water flowing out of the bottom connection *Q*.

After filling the burette with air, both cocks *P* and *Q* are closed and the bottle placed near the left end of the apparatus. The end of the tubing which was held in the mouth should be connected with the end of the main connecting pipe at *H*, Fig. 2, and a second bottle, similar to that in Fig. 4, containing water, connected to the bottom of the sample bottle at *Q*, Fig. 3, as illustrated in Fig. 4. This last bottle should be elevated slightly above the sample bottle to put the sample under a light pressure. This is always advisable to prevent infiltration of air if there is any leak.

For measuring a sample, the method originated by Prof. Gill is doubtless the most simple and effective. He compares it to measuring a bushel of grain by heaping up the measure and then striking off the top with a straight-edge. More gas than required is drawn into the measuring burette and compressed until it occupies just the volume desired; then opening the cock *G*, Fig. 4, permits the excess gas to escape.

Before drawing the sample of gas into the burette, sufficient water to fill the bottle and its connecting tube up to the bottom of the burette, should be placed in the leveling bottle *L*. During the filling of the bottle, the cock *G* should be open to the air; then, with *G* and *K* still open, the bottle should be raised until the burette and the main connecting pipe to the pipettes are filled with water, after which *G* should be closed. This cock *G*, Fig. 4, is a three-way cock, and can close the end of the main connecting pipe, or open it to the atmosphere, or connect it with the end *H*, Fig. 2. Opening cocks *K* and *H*, Fig. 4, and lowering the leveling bottle, will transfer the sample from the bottle to the burette. When the sample reaches a point 1 or 2 c.c. below the zero mark in the burette, the pinch-cock *K*, Fig. 4, and the cock *G*, should be closed.

Examining the burette will show drops of water trickling down the sides. As these occupy space used to measure the gas sample, they should be eliminated before actual measuring begins, although this is not as important as to have the amount left at each measurement the same. One per cent. of the space could be occupied by moisture at every meas-

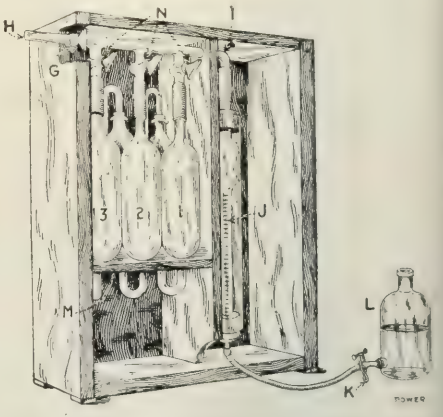


Fig. 2. Orsat Apparatus Assembled

urement of gas without appreciably affecting the result of the volumetric analysis, but if, at one measurement, the sides of the burette were practically dry, and at the next, 1 per cent. of water was clinging to the sides, serious errors would result.

To obtain uniform conditions, a fixed time must elapse after the gas has been run into the burette so that the water can drain down the sides before measuring the volume of the gas. One minute is enough. A sample should be measured at the same pressure all through an analysis. Any pressure would do, but atmospheric pressure is most convenient as it can be obtained by simply bringing the water in the bottle and the burette to the same level, having the cock *K*, Fig. 4, open, for the mouth of the leveling bottle is open to the atmosphere. Keeping the temperature of the gas sample constant during the analysis is also important and is the reason the burette is jacketed with water. Advisable also is keeping the apparatus unexposed to drafts.

After one minute has elapsed for draining the water from the sides of the

burette, the leveling bottle is raised, with cock *K* open, until the sample in the burette is compressed to exactly 100 c.c. Now, closing the pinch-cock *K* and opening cock *G* to the air, the excess of gas escapes, after which cock *G* should be closed to retain the sample intact. Next cock *K* should be opened and the water in the bottle brought level with that in the burette, as a test of the measurement, which should show 100 c.c. at atmospheric pressure. Sometimes a little moisture will obstruct the opening through *G*, thus retaining a small excess of the sample, in which case a little larger excess of gas should be drawn into the burette from the sample bottle and the operations described repeated.

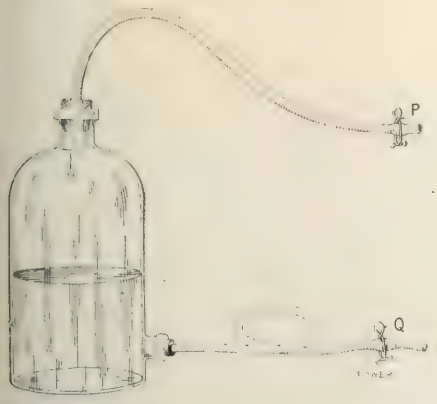


FIG. 3. BOTTLE FOR COLLECTING GAS SAMPLE

ANALYSIS OF SAMPLE

As explained in the first article, the pipette next to the burette, No. 1, Fig. 2, contains the caustic potash solution, the middle one, No. 2, potassium pyrogallate and No. 3, acid cuprous chloride. Most writers on flue-gas analysis fail to impress on the beginner that speed in using the Orsat apparatus can be attained only by keeping the gas agitated, while it is in the pipettes, and in contact with the surfaces of the glass tubes wetted by the reagents. The only ways to produce this agitation are to shift the gas sample back and forth between the burette and the pipette, or agitate the reagent by pressing on the rubber bag connected to the outlet *D*, Fig. 1, of the rear leg.

If the entire sample of gas is shifted between the burette and the pipette containing the caustic potash solution, about four or five times, the absorption of CO_2 is usually complete, but more passes may be needed, as several things tend to vary the rate of absorption, such as the condition of the reagents, the temperature and, possibly, the composition of the gas. If measurements of the volume of the gas after consecutive passes, tally to within 0.01 per cent. for CO_2 or O, the absorption may be considered complete, but for CO, two consecutive readings should tally as nearly as can be determined by the eye.

To hasten an analysis these comparative readings may be made immediately after the sample has been run into the burette instead of waiting a minute for drainage. If the two consecutive readings compare favorably, the final reading may be taken after a minute has elapsed from the time the gas is returned to the burette, including the time occupied by the comparative reading.

While shifting the gas back and forth between the pipette and the burette, it is most convenient to leave the cock connecting the pipette open, and control the flow of the gas entirely by the position of the leveling bottle, until time to measure the volume. Then the reagent must be brought back to about the level of the mark on the pipette stem, the cock closed and the water levels in the bottle and the burette made the same.

With a breath sample, the CO_2 content will probably be from 2 to 4 per cent. of the total volume and may be noted on the paper pasted on the inside of the frame. To determine the O, the apparatus is manipulated as before, but more time is required for absorption, generally about twice as much as for CO_2, or say, eight or ten passes, and complete absorption is again judged by comparing consecutive volume readings until they agree within reasonable limits. The O absorbed is also noted on the paper slip, and from the breath will be about 17 to 18 per cent., if the CO_2 is 3 per cent., for the total oxygen in the air is about 21 per cent. by volume.

If the gas sample had been drawn from a furnace, the determination of CO would be the next step, but, as none of this is present in the breath, it would be useless to test for it. The process of testing, however, is identical with the other two, except that the reagent is far less active than those used before, and consequently more time and more care are necessary in making the comparative readings between the passes of the sample into the burette. In regular flue-gas analysis, usually little or no CO is present, and small inaccuracies in the determination are relatively more important than for CO_2 or O. The small quantity of CO, usually found, and the uncertain action of the reagent, make this part of the test very unsatisfactory.

The remainder of the gas is assumed to be nitrogen. In obtaining each successive percentage the amounts previously absorbed are deducted from that shown by the measurement in the burette. Starting with 100 c.c. of a sample and having 10 c.c. absorbed in the first pipette, indicates 10 per cent. of CO_2; if after analyzing for oxygen 19 c.c. has been absorbed 19 — 10 = 9 c.c., or 9 per cent. of O was present, and if finally the total volume absorbed is 19.6, then 19.6 — 19 = 0.6 per cent. of CO was present.

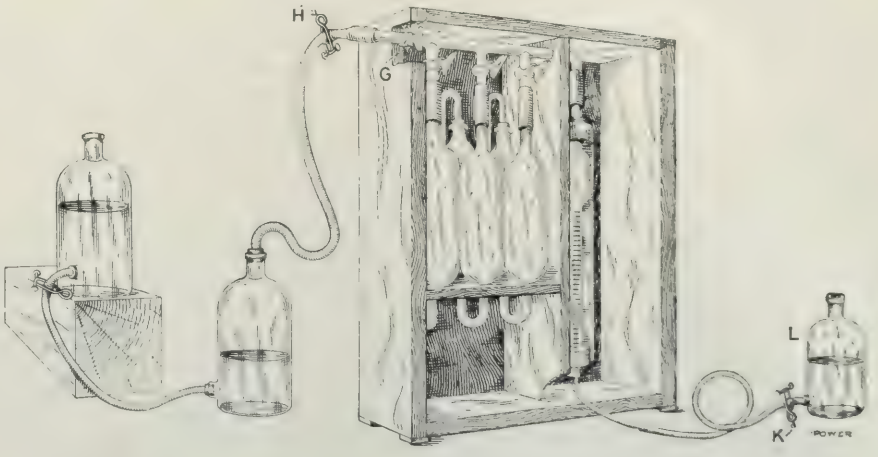


FIG. 4. GAS TRANSFERRED FROM SAMPLE BOTTLE TO BURETTE

An Oriental Lighting Plant

The installation of a 500-lamp electric-lighting plant in the Sultan's palace in Arabia has just been completed. This is the first instance of the operation of any sort of electric machinery in that country.

Permission has been granted to operate a commercial lighting and power plant employing two 30-kw. dynamos for supplying current for several thousand lamps. One set of machines will be run at night and the other in the day, principally for the purpose of driving fans which, on account of the great heat and the difficulty of securing energetic punkah pullers, are much needed. Power will be furnished by oil engines and direct current will be used.

Putting steam boilers in charge of incompetent men is as foolhardy as sitting with a lighted cigar on an open can of gunpowder. The chances are good for an explosion in either case. Two instances of boiler explosion have recently been reported and in each the boiler attendant was killed.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Inspecting and Testing Electrical Apparatus IV

By A. L. Cook

Transformers

There are three general classes of transformers used for power supply:

1. The air-blast type in which all the insulation is of a solid nature and cooling is accomplished by forcing a current of air through the windings and laminations. This type is made for voltages up to 33,000 and in capacities up to 3000 kv.-a.

2. The oil-insulated self-cooling type, in which part of the insulation is of solid character, the remainder consisting of the insulating oil in which the windings and core are immersed. This type is cooled by radiation from the containing tank, the oil serving to conduct the heat from the windings. They are made for any commercial voltage and in capacities up to about 3000 kv.-a.

3. The oil-insulated, water-cooled type is similar to the preceding type with the addition of coils of pipe immersed in the oil through which cooling water is circulated. These transformers are made for any voltage and in any capacity up to 10,000 kv.-a.

Oil-insulated transformers are frequently shipped completely assembled, with the oil in the tank. In this case the transformer has only to be put in place and connected to the circuit. Air-blast and oil-insulated transformers shipped out of the oil generally require drying out after installation; the necessity depending upon the voltage and the condition of the windings as regards moisture. Sometimes the windings of oil-insulated transformers are shipped in sealed tin cases; hence do not require drying unless the case has become damaged.

Transformers of small capacity can be dried out by passing current through the windings. The transformer should be left in the packing case, with openings at the top and bottom to allow air circulation and the low-voltage windings short-circuited. Alternating current at normal frequency should be passed through the high-voltage windings and should be so regulated that the temperature of the windings in the hottest parts will not exceed 160 deg. F. To obtain the proper temperature will generally require about one-fifth full load current, and a voltage about 2 per cent. of normal; in no case should 150 per cent. of full load current be exceeded.

For large transformers it is also desirable to apply external heat, to assist in the drying-out process. This can be furnished by incandescent lamps or iron-wire resistance coils placed at the bottom of the case. If the latter are used, the windings near the resistances should be shielded from the direct heat and the resistances must be installed so as to avoid danger of setting fire to the packing case or windings. Where it is impracticable to use current for drying out, heated air may be passed into the bottom of the case by a flue. Care should be taken, however, that the gases of combustion are not carried into the transformer. Drying out a transformer may take a week or more of continuous operation, depending upon its size. If current is used, a switch should be provided so that it can be cut off readily if the temperature rises too high. Where impossible to get exactly the right voltage to hold the temperature steady the current can be cut off at intervals.

When drying out transformers, they should always be inclosed to assist in retaining the heat and only a slow circulation of air allowed; sometimes it is convenient to place the windings in the tank for this purpose. Transformers for voltages of 40,000 and above, require special drying methods, one of the best of which employs a vacuum.

It is impossible to tell when a transformer is entirely dry, but measurement of the insulation resistance between the windings and the core by means of a suitable voltmeter is sufficient for practical purposes. To get satisfactory results, use either a 600- or a 750-volt standard voltmeter with a 500-volt circuit, or a special high-resistance voltmeter may be used with 125 volts. The ordinary voltmeter used on 125- or 250-volt circuits would not, in general, be satisfactory, as it could not measure high enough insulation resistance. When using a voltmeter on a grounded circuit, the meter must be connected in the side which is not grounded, and the lead between the voltmeter and the apparatus tested should have good insulation so as not to affect the readings. It is also well to test the switches connected to the windings when drying out, to see that they have a high insulation resistance. When testing the leads or switches alone, there should be no reading of the meter. Tests of the insulation resistance should be made at intervals of 2 or 3 hr. and the drying process continued until the resistance remains constant for several readings, with the temperature of the coils remaining the same; the transformer can then be filled with oil if this is used.

It is important that the oil be free from moisture. If water is found in the bottom of the barrel, or there is reason to suspect its presence, the oil should be tested. Various tests are recommended for detecting moisture in oil; none of these, however, except the high-voltage test, gives conclusive results, although the use of dry copper sulphate is fairly satisfactory. To prepare the copper sulphate, ordinary blue vitriol may be heated to a moderate temperature until all the water is driven off and a fine white powder remains, a small quantity of which can then be shaken up in a bottle containing a sample of the oil. If the powder takes on a bluish tinge it is an indication of moisture. The trouble with this test, as with others suggested, is that considerable experience on the part of the tester is required to detect the presence of small quantities of moisture. The only reliable method is to determine the voltage required to break down a definite gap between terminals immersed in a sample of the oil. If the approximate methods of test indicate moisture or there is reason to believe that such exists, the voltage test should be made, particularly on oil used in high-voltage transformers.

If moisture is found, it must be removed before the oil is used. When there is a large amount of water present, it will settle to the bottom of the barrel and can be drawn off from the bottom. This, however, does not leave the oil suitable for use, as it will still hold enough water in suspension to make its dielectric strength very low. To dry out the oil, it may be heated in an open tank, either by external heat or by resistance coils immersed in the oil, the temperature being maintained at about 220 deg. F. The oil should be well stirred and the temperature maintained for about 10 hr. or until the moisture has been removed. The objection to this method is that long continued heating of the oil changes its character, making it unsatisfactory for use in the transformers. If a vacuum outfit is used for drying the transformer windings, this can also be employed to advantage for drying the oil, since the temperature need then only be raised to about 160 deg. F. Oil can also be dried by passing it over unslacked

lime, for which purpose good clean lime should be used and the oil slowly circulated through it. Before being treated by the lime, the oil should be filtered through a piece of chamois skin, which will remove part of the water. It is well to filter the oil in this manner, irrespective of the final method of treatment.

When applying the voltage for the first time to a bank of transformers it is well to bring up the voltage gradually, using one of the generators for the purpose. The ammeter should be watched to detect any excessive current. This method not only tests the windings of the transformer but also tests the connections between transformers. When the voltage has been brought up to normal that on each phase of the high-tension side should be checked if there is a voltmeter in this part of the circuit. The apparatus is then ready for service.

Lightning Arresters

Lightning arresters having a number of metal gaps in series require only careful cleaning and inspection before applying the voltage. The electrolytic type, consisting of a number of aluminum trays nested together with the electrolyte forming a conducting path between adjacent trays, require a preliminary charge, however. To charge these for the first time, gradually build up the voltage on the arrester after it has been completely assembled, the voltage being applied through a 10-amp. fuse wire and gradually increased, either by regulating the generator voltage or by using the transformer with a water resistance. The voltage should be increased slowly, taking about 5 min. to reach normal and noting any noise in the arrester which would indicate that it was not in proper condition. A better method is to apply a low voltage to each pair of trays after they have been filled with the electrolyte but before being put into the tank. To do this properly, 220-volt alternating current is necessary, although 110 volts can be used to give a partial charge and 550 volts if two cells are charged at a time. A lamp bank giving about 2 amp. when the arrester is not in circuit, should be provided, and the charging leads should then be applied to two adjacent trays. At the first instant, the lamps will burn at full candlepower but in a few seconds they will dim considerably. If there is no change, the two trays are short-circuited, or the film has not formed. If the lamps are dim at first, this indicates either that the trays have not been properly filled, or that the tray is already charged. The indication with 110 volts will not be as definite as with 220 volts since the voltage is not sufficient to completely charge the trays. After the trays have been charged in this manner, they can be put in the tanks and the oil poured in; the arrester is then ready for service.

Feeders

Methods of testing feeders depend largely upon the normal voltage and the importance of the particular circuit. Underground feeders are generally more thoroughly tested than overhead feeders, since the condition of the latter can be partly determined by inspection. Low-voltage feeders, both overhead and underground, can be tested in the manner described for direct-current feeders. (See Article I of this series, Aug. 6 issue.) Medium- and high-voltage feeders should have a voltage test applied before putting in service. With overhead feeders, this is necessary to detect defective insulators; for underground cables it is necessary in order to test the splices as well as to make certain that the cable has not been damaged during installation.

Overhead lines for 11,000 volts and over, should, if possible, be tested at double voltage, applied for 15 min. between each wire and ground. This would require a high-voltage testing transformer, and suitable regulating and voltage measuring devices, and as such tests are justified only for long lines, they will not be considered here. If only the normal transmission voltage is available, the line may be tested by gradually raising the voltage on the line, using a generator for the purpose. Generally, this can be done by grounding one terminal of the testing circuit through a resistance and attaching the other terminal to all the conductors of the feeder. When the voltage is built up there will be practically full voltage applied between the conductors and the ground which gives them a test to ground about double the normal potential difference for a single-phase or two-phase line and about 70 per cent. more for a three-phase line. This test should be applied for about 15 min. The test also subjects the windings of the generator or the transformer to the same voltage. After making the test to ground, the line can be connected to a spare generator and the voltage gradually built up to normal; this will test for short-circuits between conductors.

High-voltage underground cables require careful testing during installation, and before putting in regular service. While the cable is being spliced it is customary to measure the insulation resistance at frequent intervals as a check on the splicers. These measurements have to be taken with special apparatus, however, as the voltmeter method is not sensitive enough. When the cable is completely spliced and ready, measurements of insulation resistance are first made and then a high-voltage test is applied. The testing voltage varies with the conditions, but generally double the normal voltage, applied for 5 min., is sufficient. Testing underground cables in this manner requires special apparatus and considerable care, and is of such a character that it can generally be made most satisfactorily by the manufacturers; hence no attempt will be made to describe the test here.

Relatively low-voltage cables, such as 2200 volts, or even short lengths of cable for higher voltages may not justify the expense of elaborate testing methods, but a test at normal voltage should be made, employing the method described for overhead feeders.

After the feeders have been tested, they can be connected up for regular service. If two or more connect to a set of busbars at each end, as in a power station and a substation, it is necessary to "phase out" before making permanent connections. To do this, connect one terminal of a test-lamp circuit to ground and the other to one lead of the feeder. The leads at the other end should then be connected to ground through a lamp, testing one lead at a time until the lamps burn. The lead can then be tagged at each end and the test repeated on the other leads. A diagram of connections for this test is given in Fig. 10.

Substations

After the feeders entering the substation have been phased out, the main wiring should be tested; the extent of these tests depending upon the size of the station and the voltage.

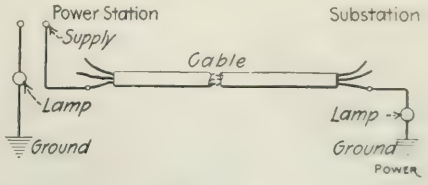


Fig. 10

The procedure to be followed for the inspection of the switching and metering equipment is the same as that described for power stations.

The adjustment and fitting of brushes on rotary converters should be made in the same manner as for direct-current generators, the brushes being given a slight forward lead. The transformers should be inspected and, if necessary, dried out as previously described. The alternating-current side of the converter should be disconnected from the transformers, either by raising the brushes from the slip rings, or by removing the leads, and the voltage gradually raised on the transformers to test the connections. When the applied voltage is normal, the voltage at the alternating-current brushes should be checked. For a three-phase converter, with three slip rings, the voltage between any two leads or sets of brushes should be identical, and about 61 per cent. of the direct-current voltage. For a two-phase converter with four slip rings, the voltage for each phase is about 71 per cent. of the direct-current voltage. Six-phase rotary

converters are frequently used on three-phase systems. The two common arrangements of the connections are "double delta" in which the secondary of the three transformers are divided into two parts with two independent deltas; and the "diametrical" where the secondary windings of the transformers are not connected together but are each run independently to a set of brushes. The alternating-current voltage is about 61 per cent. of the direct-current voltage for the double-delta connection, and about 71 per cent. for the diametrical.

Converters which are started from the alternating-current side can be phased out by closing the starting switch and noting the direction of rotation. Before doing this, the field switch provided on the machine should be opened. The correct direction of rotation is determined by the arrangement of the direct-current brushes, and if the machine starts in the wrong direction the rotation must be changed by reversing one phase of the alternating-current circuit either on the primary or secondary side of the transformers. Generally it is easier to reverse the primary circuit. In the case of a three-phase system, this should be accomplished by interchanging any two of the leads which connect to the delta of the transformers. It is not correct to interchange the connections of one of the transformers, but the change should be made in the circuit feeding the group. Therefore, it is good practice to make temporary connections between the high-tension circuit and the transformer delta so that this change can be made easily if necessary. A two-phase converter may be reversed by interchanging the leads connected to one of the transformers.

When converters are started from the direct-current side, or by means of induction motors, they must be phased out in a different manner. First, the starting devices must be tried out to see that the machine starts in the correct direction. If a spare generator is available, the converter can be easily phased out by gradually raising the voltage on the alternating-current side until the machine starts. Before doing this, the shunt-field circuit should be opened in at least two places by removing the connections between adjacent field coils, leaving about the same number of coils in each part of the circuit. This precaution is necessary to reduce the voltage in the field coils during the starting operation. Then the voltage can be gradually applied to the alternating-current side by slowly raising the generator voltage. The machine should start before half the normal voltage is reached; if it starts in the reverse direction, the connections must be changed as described.

After this test is completed, the field coils should be reconnected and the synchronizing connections checked.

Where it is not possible to raise the voltage gradually, the machine can be phased out with lamps. If there are switches in the leads between the slip rings and the transformers, these should be opened; otherwise, the leads must be removed from the brush-holder studs. Lamps should then be connected between each lead and the corresponding stud, or across the corresponding clips of the switch, using enough lamps in each group so that the sum of their voltages will about equal the voltage between phases. The proper connections of the lamps for three-phase and two-phase converters are shown in Figs. 11 and 12. The converter can now be started from the direct-current side, or by means of the starting motor. If a starting motor is used, a voltmeter should be placed across the direct-current terminals of the converter and its reading noted as the machine speeds up. The shunt-field rheostat should be partly cut out so that the voltage of the machine will build up in the same manner as a direct-current generator. If the machine fails to generate, or builds up in the wrong direction, the same procedure should be followed as for direct-current generators. If the machine builds up in the correct direction, the alternating current may be applied to the transformers. The lamps in the secondary circuits will vary at intervals from bright to dark, the rate of pulsation depending upon the deviation from normal speed. If the connections are correct, all the lamps will be dark at the same time. If the sets of lamps are dark at different times, either the primary or secondary connections should be reversed, as previously described, and the test repeated. If the change is correctly made, all lamps should be dark at the same instant, showing that the phases have the correct rotation.

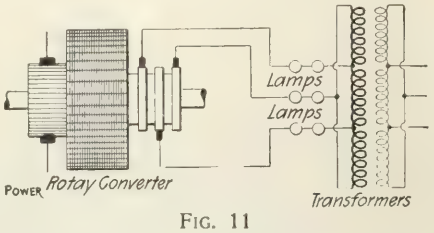


FIG. 11

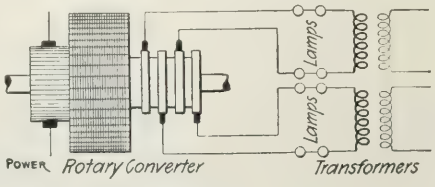


FIG. 12

The regular synchronizing connections should now be checked. To do this, all the alternating-current brushes should be raised clear of the slip rings, taking care that brushes on adjacent arms do not touch each other, or ground on the frame of the machine. If there is a switch in the secondary circuit, it should be closed and then the switch in the primary circuit of the transformers should be closed. The synchronizing lamp and the synchroscope, if there is one, should then indicate synchronism; if they do not, the synchronizing connections should be corrected. Having checked these connections, the transformer switch should be opened, the brushes replaced, and the machine synchronized in the regular manner. The converter is then ready to be connected to the direct-current busbars, the procedure to be followed being the same as for a direct-current generator.

If a synchronous motor is to be put in service it can be phased out in the same manner as a converter.

CORRESPONDENCE

Motors Will Not Carry the Load

Referring to Mr. Shrode's inquiry in the June 18 issue, it would appear that his 7½-hp. induction motor has been run with the belt too tight, thus causing excessive wear on the pulley bearing. If this has been taken up, and the belt runs with normal tension, I see no reason why the motor should slow down very much below normal speed unless too small for the load. It is assumed the applied voltage is that at which the motor is rated.

I would recommend that the rotor bars of the second motor of the soldered-disk type be tested for grounds. If found free from grounds, be sure to solder the disks securely to each rotor bar. It is essential that they be soldered well, since heavy currents flow in the bars during starting and during periods of heavy load.

BEN. DAWSON.

Cedar Rapids, Iowa.

Reversed Polarity

In the July 2 issue, Mr. Mitchel writes how he reversed the polarity of his direct-current generator by running two wires from the No. 1 machine to the field connections of No. 2. This is not necessary either to change the polarity or to charge the fields for lost residual magnetism.

To change polarity or to charge, all the brushes on the machine to be changed or charged should be raised and a piece of cardboard or heavy paper placed beneath to insulate them from the commutator. This can be done without loosening the brush-holder springs. If no paper is handy, the spring can be loosened, the brush raised and the clamp then tightened to hold the brush up. One should be sure that no brush touches the commutator. Finally opening the rheostat to running position and throwing in the machine or paralleling switch for a moment will reverse the polarity, or charge the field magnets as the case may be.

C. A. SCOTT.

Wales, Wis.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Operating Costs at Willard Plant

By A. D. Williams

The plant at which these operating statistics were obtained is a part of the factory equipment supplying power for use in forming storage-battery plates and charging storage batteries. The plant operates upon a 24-hr. schedule, 7 days a week, as the process of forming storage-battery plates requires a continuous supply of direct current, which cannot be interrupted without interferring with the electro-chemical reactions. For this reason it was deemed inadvisable to depend upon central-station service. The original plant comprised the two 80-hp. units, one of which acted as a reserve; the 135-hp. unit was next added, then the 300-hp. machine. The later machines were selected so that sufficient reserve capacity would be available at all times to carry the full load on the plant.

The engineering force is divided into three watches of 8 hr. each; the chief taking the day run from 7 a.m. to 3 p.m. The other two watches, taken by the assistant engineers, are from 3 to 11 p.m. and from 11 p.m. to 7 a.m., except upon Saturday nights, when they change shifts with a dog watch of from 3 p.m. to 6 p.m. and a 13-hr. shift to 7 a.m. The load conditions are such that runs of from 7 to 20 days without a stop are frequent with occasional longer runs. In 1911 the 135-hp. unit was operated continuously for three months. The 300-hp. unit has eclipsed this, however, with a continuous run which started Dec. 26, 1911, and ended Apr. 6, 1912—102 days, the unit being started with only a 24-hr. shutdown intervening between it and a previous 19-day run. On Apr. 6, the machine was shut down to replace one of the exhaust valves and aside from this was in condition to have run much longer.

The operating cost statistic herein given cover the first four months of 1912. During this period the crank cases of all the engines were refilled with oil so that this item is higher, except in February, than the normal. The oil item also includes that used outside of the power plant for lubrication; this, however, is a small amount as comparatively little machinery is required in making storage batteries. The rental charge is based upon the floor space occupied and the insurance upon the plant valuation. The depreciation charge is upon a plant life for the machinery of 7 years. The labor charge includes, beside the engine-room force, part time of a laborer in handling the coal and ashes for the factory-heating plant. The water rate includes that used in heating as well as the engine cooling water. City water is used with a meter charge of 40c. per 1000 cu.ft. The engine cooling water passes through a home-made cooling tower, which has reduced the water bill between $150 and $200 per month, so that the tower pays

OPERATING DATA, WILLARD STORAGE BATTERY PLANT

	Jan.	Feb.	March	April
Rent of space occupied	$25 00	$25 00	$25 00	$25 00
Depreciation	303 86	303 86	303 86	303 86
Insurance	49 05	49 05	49 05	49 05
Water at $0 40 per 1000 cu.ft.	20 77	20 77	20 77	21 91
Gas at $0 30 per 1000 cu.ft.	511 80	501 60	552 60	615 20
Repairs	70 54	58 07	39 41	61 14
Oil	101 88	78 13	116 13	143 51
Miscellaneous	2 34	26 32	2 00	14 18
Labor	396 00	394 70	417 98	412 86
Totals	$1,481 24	$1,457 50	$1,526 80	$1,647 71
Deduct Credit items	9 05	387 38	364 59	541 02
Net total	$1,472 19	$1,070 12	$1,162 21	$1,106 69
Heating factory	314 01	328 43	169 89	78 00
Total, Power and Heating	$1,786 20	$1,398 55	$1,332 10	$1,184 69
Credit: Old oil barrels	$ 9 05	$ 3 55	$ 9 00	$
Sale of energy		383 83	355 59	541 02
Total Credit	$ 9 05	$ 387 38	$ 364 59	$ 541 02
Heating: Coal and freight	$ 308 01	$ 319 43	$ 154 89	$ 72 00
Use of side track	6 00	9 00	15 00	6 00
Total	$ 314 01	$ 328 43	$ 169 89	$ 78 00
Kw. hours generated	122,428	119,856	135,690	135,572
Gross cost per kw.hr. generated	0 01210	0 01216	0 01125	0 01215
Net cost per kw.hr. generated	0 01202	0 01124	0 00857	0 00816
Cost of power and heating per kw.hr. generated	0 01459	0 01167	0 00982	0 00874
Load factor of generating plant	41 8	44 0	46 3	47 27
Total 1000 cu.ft. units of gas	1,706	1,672	1,842	2,054
Total cu.ft. gas per kw.hr. generated	13 935	13 950	13 575	15 153

The last items include all gas used in the power plant, for starting, when engines are running light and under partial loads. It is the average consumption of all the engines under the commercial loads they carry.

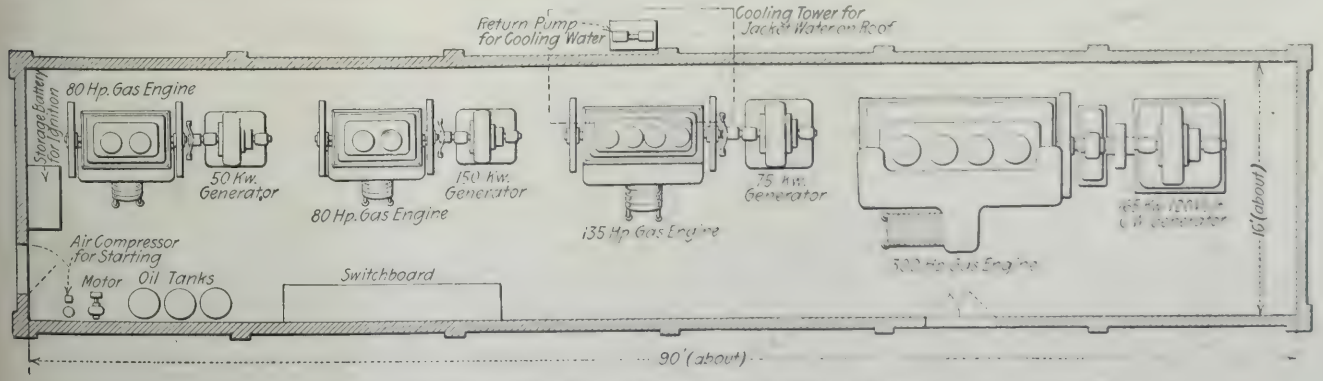


Fig. 1. Plan of Engine Room at Willard Plant

for itself from two to three times per year.

Only one item seems to be omitted, namely, the interest upon the money invested in the plant. The inclusion of this item, unless it has been lumped with the depreciation, would add about $0.00125 to the rate per kilowatt-hour.

For much of the foregoing data the author is indebted to J. F. Kalb, chief engineer of the plant, and to the Bruce-Macbeth Engine Co., who installed the engines.

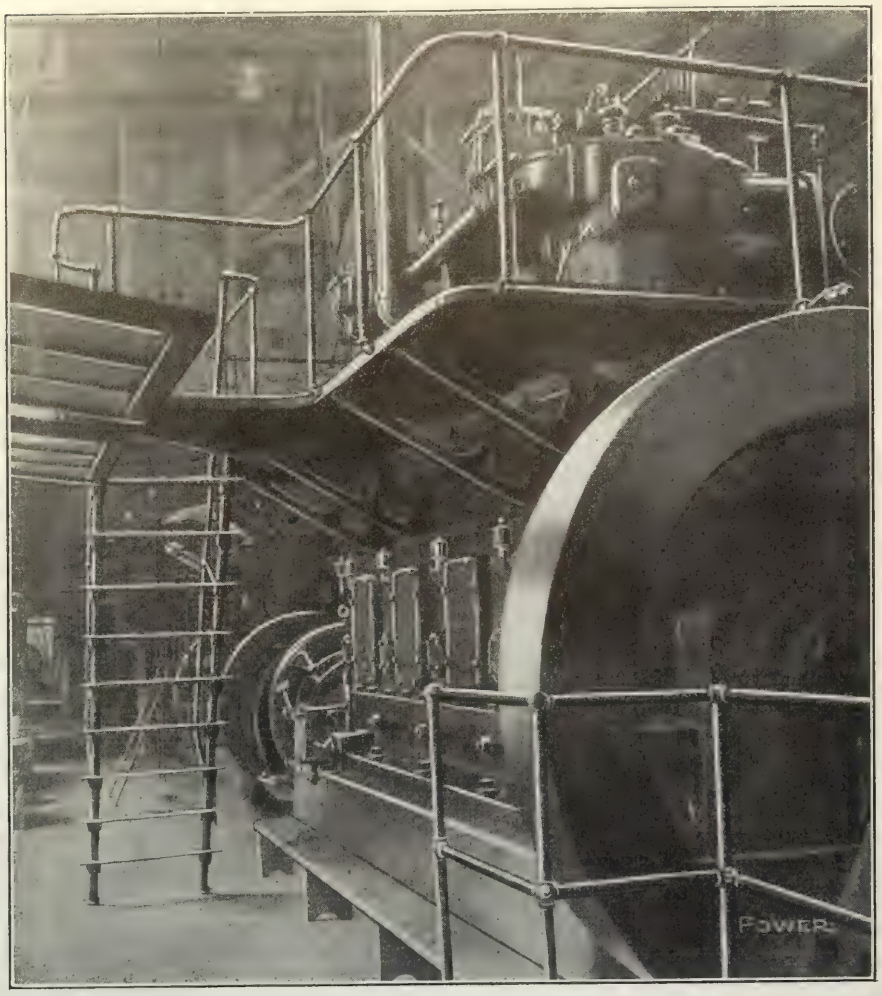


FIG. 2. VIEW OF ENGINE ROOM; 300-HP. UNIT IN FOREGROUND

Preheating Gas Engine Charge

BY P. A. TANNER AND A. C. EDWARDS

Acting on the suggestion of the chief engineer of a well known gas-engine concern, the authors conducted a series of tests extending over two or three months to determine the effect on a gas engine's economy by preheating the charge.

It was thought that by preheating the charge not only could the compression pressure be raised, at light loads without preignition, but heat from the exhaust gases could be returned for utilization in the engine cylinders.

It can be shown by thermodynamic analysis that if the charge is preheated so that the temperature T_1 at A (Fig. 1) is, say, 100 deg. F. higher than normal, the temperature T_2 at B will be more than 100 deg. F. higher than if the charge had not been heated 100 deg. F. at point A. Assuming, then, the same number of heat units added on the combustion line BC, both with and without preheating, the temperature T_3 at point C will be more than 100 deg. F. higher than if the charge had not been preheated. Evidently, then, while 100 deg. F. has been added to the lowest temperature of the cycle, more than that has been added to the highest; consequently the Carnot efficiency has been increased. This may not mean, however, that the actual efficiency of the engine is improved, but merely the theoretical limit has been raised, giving a better chance for increase in the actual efficiency. From a practical standpoint, apparently, assuming the same energy entering the engine per charge otherwise than sensible heat, both with and without preheating, the only gain in actual efficiency will be due to better combustion because of the higher temperature.

THE ENGINE

The engine used for the tests was of the vertical two-cylinder, four-stroke-cycle type, built by the Bruce-Macbeth Engine Co., and was rated at 27 b.hp., with a guaranteed continuous overload capacity of 20 per cent. It is shown in Fig. 2 and operated on city illuminating gas. The governing is effected by a fly-ball governor actuating the mixing-valve stem. This valve is so constructed that the proportions of the mixture remain constant while the quantity is varied.

The ignition current was supplied by a high-tension magneto, the timing and distribution of the spark being contained within the magneto. The timing could be varied throughout a range of 30 deg. by a shift lever at the front of the magneto.

Following are a few of the important particulars of the engine: revolutions per minute (normal), 325; cylinder bore, 8½ in.; stroke, 10 in.; clearance, computed, 22.32 per cent.

AIR HEATING APPARATUS

The air heater consisted of a 12x12x36-in. wooden box containing a 2x4x34-in. asbestos core upon which was closely wound a spiral steel spring. This spring was over 1 in. in diameter and when wound on the core formed a large heating surface over which the air could circulate freely. Referring to Fig. 3, the core was held in place by the bolts AA. The binding posts B, B' and B'' were for attaching the electric-supply wires. With connection at B and B'', current was sent through the entire coil; when at B' and B'', current passed through about three-fifths of the coil; or at B' and B, through about two-fifths of the coil. By connecting the sections BB' and $B'B''$ in parallel, a very high temperature was attainable. Through rheostats placed in parallel the temperature could be regulated within a few degrees.

FIG. 1

Cold air was sucked into the heater through 1-in. holes at the bottom of the box; the hot air coming out at the top and passing through an orifice connected with a new style Meriam gas economy indicator used for measuring the air. The temperature of the air was taken just above the orifice and also near the engine mixing valve (see Fig. 3).

Measuring the Gas

The gas was measured by a calibrated meter and then passed through the casting of an old type Meriam gas economy indicator. This casting was simply a chamber, in two halves, joined in the middle by flanges and bolts and divided into two compartments by a copper diaphragm having an orifice ¾ in. in diam-

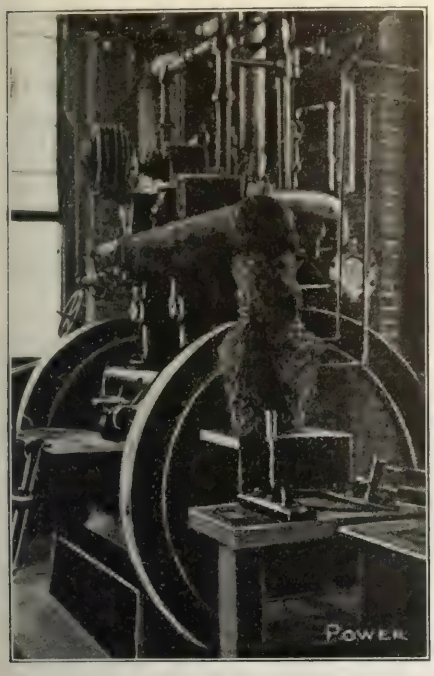


Fig. 2. Engine Rigged for Testing

eter. There is a slight difference in pressure in the gas before and after passing through this orifice, which is proportional to the amount of gas flowing through in a unit of time. This principle was used, then, as a direct-reading flow meter for the gas. For this purpose small tubes were led from the chamber on each side of the diaphragm to a sensitive manometer showing when the least gas was flowing. Very slight differences in pressure and consequently, very slight changes in the rate of flow were thus detected. After passing through the orifice, the gas passed into a small gasometer and thence to the mixing chamber.

Arrangement of Apparatus

Water manometers were placed at the outlet of the meter and on the gasometer. The pressure at which the gas was metered and that at which it was supplied to the engine were thus noted.

The indicators were fitted to the engine and driven through a specially constructed reducing rig, the motion being derived from an eccentric placed on the crankshaft between the flywheel hub and the end housing.

A prony brake was used to measure the delivered power. Thermometer cups were inserted in the gas pipe just above the mixing valve, in the inlet and discharge cooling water pipes, in each exhaust pipe leading to the manifold, and two in the air line, one just above the orifice casting and the other just outside the mixing valve. A water meter was placed on the cooling water discharge line.

Methods of Making Tests

Each test of the series was run at one load, and was divided into runs, each run at a different temperature of the entering air. Only the air was heated, the gas being mixed with it before entering the cylinder. The runs reported were from 1 to 2 hr. long, the longer runs being at light loads. Except the second set of runs, those at a given load were all made the same day, with the engine run continuously throughout the several runs.

The engine was allowed to run from ½ to 1 hr. at the desired conditions before the data were taken. On some of the runs, data were taken for 3 hr. and only those for the 2 last hours of the run were preserved. This was because during the first part of the run trouble was experienced from backfiring, which was accompanied by a greater gas consumption. The cooling water was always at about 66 deg. F. at the inlet and the valve on the water line was adjusted to keep the outgoing water at 120 deg. F.

Before starting each run, the gas and air throttle was adjusted to give the best possible economy. An old style Meriam gas economy indicator was used as already described, to indicate the position of the mixing valve and spark advance which gave the best economy under the different conditions. This meter was not used to give the actual flow of gas, but only to indicate when the minimum flow was obtained. The new style Meriam gas economy indicator, which was used to measure the air, was carefully adjusted to the zero point before each run.

Heat Value of the Gas

The calorific value of the gas was obtained by a Junker's calorimeter; the method of sampling the gas was as follows: Where the gas pipe enters the building, it divided, one part going to the engine and the other supplying other services. To get a continuous sample, a bunsen burner was allowed to burn on the line serving the building, while the engine was in operation. Almost an hour before the last run of the day was finished, the calorimeter was set up and by the time the tests were finished, the observers were ready to use the calorimeter; several determinations were made and the average taken. The lower heat value of the gas thus obtained was used.

Conclusions

From these tests only one conclusion is possible. The efficiency of this engine under these circumstances was not materially increased. Some interesting things will be noted, however, by referring to the summarized data sheet. Among these is the decreased speed of the engine, resulting, perhaps, from a change in the length of the mixing-valve stem, due to expansion on account of the change in the temperatures of the mixture around it. Noticeable also is the increased volumetric efficiency. This, however, is to be expected because of the increased volume of the charge due to the preheating. Another interesting thing, which also was to be expected, was the decreased power of the engine when the charge was heated. This is explained in that the temperature of the charge was increased, making the weight of the charge per stroke less; hence the heat added per unit of time and the amount transformed into work were less.

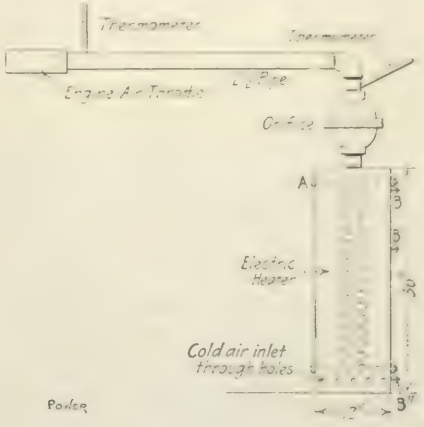


Fig. 3. Air-heating Apparatus

SUMMARIZED DATA

Run No.	Air Temperature Deg. F.	Average volumetric eff. from Indicator Diagrams	Air used cu.ft.	R.p.m.	Gas per hour cu.ft	B.t.u. per B.hp. hr.	B.hp.	Efficiency
1-a	101.6	77.5	5,175	325	331.5	9,564	20.00	27.1
1-b	152	85.9	5,775	318	327.0	9,641	19.57	26.9
1-c	197.1	91.3	6,150	313	323.3	9,690	19.25	26.8
2-a	103.4	46.4	2,055	325	232.5	11,963	10.65	21.7
2-b	175.2	50.7	2,415	323.5	231.8	11,862	10.65	21.84
2-c	244.4	47.5	2,610	325	281	11,942	10.60	21.7
3-a	101.5	28.8	945	327	174	18,884	5.04	13.75
3-b	176	31.6	1,050	324	168	18,379	5.00	14.1
3-c	252	31.9	1,260	324	162.5	17,778	5.00	14.6
4-a	101	59.8	4,215	322	284.3	10,472	14.85	24.8
4-b	153.3	64.5	3,960	322	282.5	10,435	14.85	24.8
4-c	206	64.2	3,900	322	281.5	10,368	14.85	25.0
5-a	103	92.4	6,645	305	380	8,894	23.5	29.2
5-b	192	94.8	6,473	295	373	9,037	22.7	28.2

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Troubles in an Absorption System and Remedies—III

By W. S. Luckenbach

Why Coils Show Different Frost

When one of the expansion coils fails to show the same frost as the others, the cause is likely to be due to using dull pipe cutters, when making the connections, thus not only leaving ragged edges internally for any substance to cling to and obstructing the passage of gas, but reducing the area of the pipe also. It is very important, throughout the entire ammonia system, that every pipe and nipple be examined internally before being used, as this may save the engineer much trouble later on.

If the expansion coils are all connected to one manifold with the expansion valve at one end, and the coil that fails is near the valve, the explanation may be that insufficient ammonia passes through the valve; hence it should be opened a trifle more, but if the front of the manifold or the pipes get moist it should be closed again, as the limit has been reached. It is then evident that either the gas is not evenly distributed, or the internal passage of the coil is obstructed. All liquids, gas, steam, etc., under pressure tend to travel in a straight line unless diverted by resistance. In this case nothing would divert the gas until the last fitting was reached. Under the best of management, some irregularity will exist unless a valve is placed at the top and bottom of each coil where it can be regulated to a nicety, or the coils are divided into sections (as they should be) of from two to four, according to the size of freezing tank.

If the coil still does not frost properly, it becomes a question of deciding whether to go on and get out of it all possible until the busy season is over, or to cut it out and remove the obstruction.

Irregular Pulling of Ice Cans

Regular draws should be made for numerous reasons, especially when distilled water is used. The cans cannot be put back until enough water is distilled to fill them, and lifting them out may lower the brine enough to expose the top row of expansion coils, thus shortening the life of the coils by excessive corrosion; also, by reducing the height of the brine in the entire bath below the top of the water in the cans, the freezing is retarded. Even with enough distilled water to fill the cans at once, the excessive draw will give water at a higher temperature, with accompanying ill effects.

Pulling Ice When Rich Liquor Requires Strengthening

Under these circumstances ice can be pulled without much, if any, loss if anhydrous or liquid ammonia is injected. Connection should be made directly into the expansion coils. The drum should be a trifle higher than the coils and the ammonia admitted at intervals until the amount required has been obtained. Knowing the exact amount needed is often important, especially toward the end of the season. Suppose the rich liquor has weakened from 26 deg. Baumé to 24, the necessary amount may be determined as follows: Referring to the standard table giving percentages of anhydrous in aqua ammonia will show that at 26 deg., 28.6 per cent. is anhydrous and at 24 deg., 24.2 per cent., a loss of 4.4 per cent. by weight. With a charge of 15,000 lb. at 26 deg., 28.6 per cent. or 4290 lb. is pure anhydrous. A loss of 4.4 per cent. would require that 188.76 lb. of liquid ammonia be added to bring the charge to 26 deg., if that amount can be got in the machine without distilling.

Sometimes using aqua ammonia instead of anhydrous is preferred. As 28.6 per cent. is anhydrous, 71.4 per cent., or 2.5 times as much is water or

$$188.76 \times 2.5 = 471.9 \textit{ lb.}$$

making

$$188.76 + 471.9 = 660.66 \textit{ lb.}$$

of 26-deg. aqua ammonia, or practically one drum to be added.

Presuming the machine has its full charge at least as much water must be drawn out as will be pumped in with the aqua ammonia, which in this case would be 472 lb. at 10 or 12 deg. Baumé. This is done by distillation and will cause scarcely any decrease in the pulling per hour. First the generator should be filled as full as possible without causing a boiler-over. Then the ammonia pump should be stopped and the poor liquor valve at the generator closed. When the generator pressure is down to, say, 40 lb., some may be drawn and tested; if it is not down to 11 or 12 deg. the steam should be continued until that point has been reached. The valve on top of the analyser or rectifier should be closed and the amount required drawn off, then from the drum a charge should be pumped until the liquor is up to its usual height, and as soon as the ammonia gage registers enough to prevent the poor liquor in the cooling coils running back into the generator, the poor liquor and gas valves should be opened and the machine started. The maximum time required should not be more than four hours. With this method only the amount of liquor in the generator is distilled.

When pumping from a drum on the floor, the pump frequently fails to lift the liquor after the drum is half empty. This may be caused by too much space between the valves and plunger and the warm ammonia, but, anyhow, it is better to elevate the drum a few feet above the pump. The pipe leading from the top of the drum to the pump is thus longer than the internal pipe and when once the air is excluded in the vertical pipe it becomes a siphon and will empty the drum.

Traces of Ammonia in Ice

This will rarely occur if the engineer daily inspects the plant, not neglecting the distilled and raw water. The first move is to test the water at the can filler with the standard solution for that purpose. Next, if a forecooler is between it and the distilled-water tank and the water is free between them, it shows that the return gas coils or pipes passing through the forecooler leak. If there is no bypass for such an emergency the machine must be shut down until the proper connections have been made. This will increase the temperature of the water in the forecooler; hence these repairs must be made as speedily as possible. An increase of 5 deg. F. is equivalent to losing one ton of ice per 24 hr. on a 60-ton machine. The addition or abstraction of 1 deg. in temperature is approximately equivalent to 1 B.t.u. per lb. of water. Hence an increase of 5 deg. would mean the loss of

$$60 \times 2000 \times 5 = 600{,}000 \textit{ B.t.u.}$$

The standard for one ton of refrigeration per 24 hr. is 284,000 B.t.u. and for one ton of ice approximately double that amount, or 568,000 B.t.u. This leaves 32,000 B.t.u. for losses by radiation in the freezing tank and meltage when removing ice. The importance of attending to these seemingly unimportant details is thus apparent.

National Ammonia Compressor

As arranged for belt drive, the National duplex ammonia compressor is herewith illustrated; direct drive can readily be substituted. The design departs from the usual inclosed-crank type in being horizontal, for which the following advantage is claimed: All important strains being horizontal, the compressor may be placed on any floor without imparting vertical vibration to it; hence a concrete foundation is not necessary. The pistons, piston pins, crankpins and main bearings are splash lubricated. The discharge valve is near the bottom of the cylinder, facilitating the discharge of foreign matter and the liquid ahead of the gas, a factor in favor of long life and smooth operation at high speed. The machine being duplex further assures smooth operation; one piston always acting against compression keeps the crankshaft against the bearing opposite the cylinder and prevents any reciprocating motion of the shaft and the consequent strain on the packing when the bearings become worn.

The shaft is a single forging of large proportions, turned down smaller where the wheel fits, thus avoiding the necessity of bushing the wheel should it ever become necessary to take a cutoff where the shaft runs in the bearings or stuffing-box. The bearing heads have interchangeable babbitt linings, oil wells and oil-return channels from the outer end and below the bushings back to the crank case.

The connecting-rods are crucible-steel castings of I-section with a strap crank end and a solid piston end. The piston-pin bearing, fitted with wedge adjustment, is cast-iron ground and lapped to fit the case-hardened and ground piston pin. The crankpin bearing, also fitted with wedge adjustment, is cast steel and babbitted.

The piston is trunk type, with three snap rings near the closed end. The clearance is adjusted by screwing the steel follower either on or off the piston proper; it is locked by a screw in the center. A coil spring actuated pin, inserted in both the follower and center screw, engages with any one of a series of countersinks so that neither the follower or screw can turn except by force.

The cylinder head is bored to fit the follower accurately. Its hemispherical form allows using valves of such area that should the piston speed be doubled, they would not be required to open to their full capacity with only a few ounces difference in pressure inside and outside the valve.

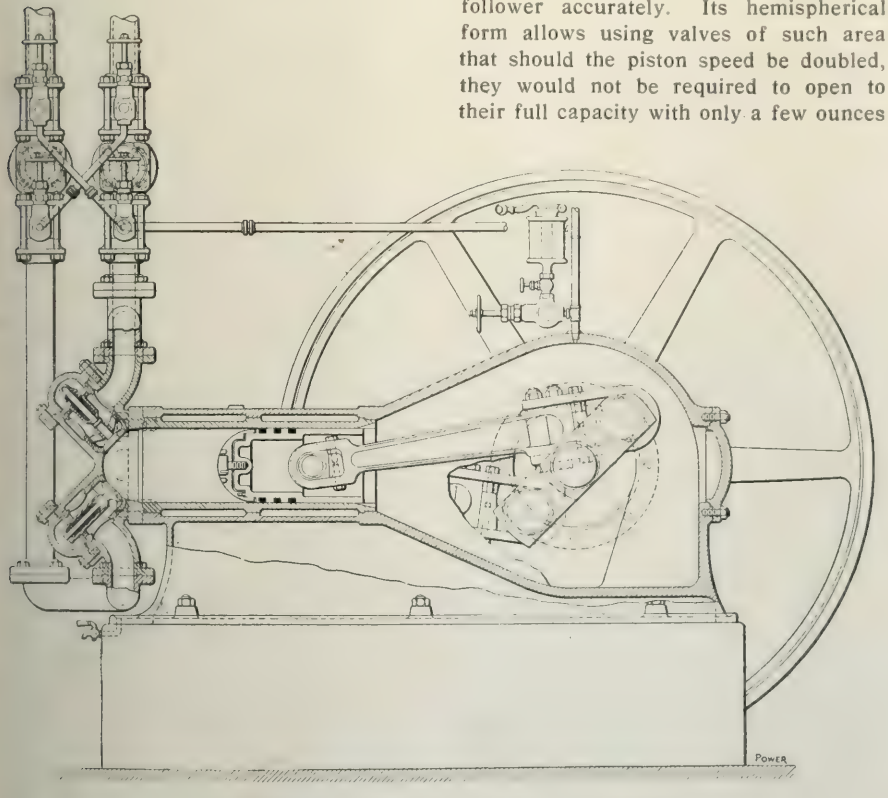
The suction valves have cast-steel safety collars, between the steel seat and cages, which also act as bearings for the lower ends of the valve stems. The weakest part of a stem is above the collar, but if it failed it could not fall into the cylinder. The valve cages are ground into the heads and held in place by the valve caps.

The cylinders are cast separately of hard close-grained iron, machined, tested to 500 lb. hydraulic pressure and pressed into the frame. The frame complete with all heads in place is again tested to 400 lb. hydraulic pressure and 300 lb. air pressure.

The water jacket surrounds the cylinder for a length sufficient to cool the compressed gases but not to warm the forward end of the cylinder. All heads and flanged joints are made male and female with lead gaskets. Studs and through bolts are used wherever possible. The suction and discharge pipes have cross-connections; the crank case is connected with the suction pipe and also with the oil trap, so the oil that works through can easily be returned to the crank case.

The ball-and-socket ring-oiling type outboard bearing rests on a substantial pedestal and is adjustable with setscrews to take up wear and maintain alignment.

These machines are built in sizes of from 3 to 25 tons capacity by the National Machine Co., Milwaukee, Wis.

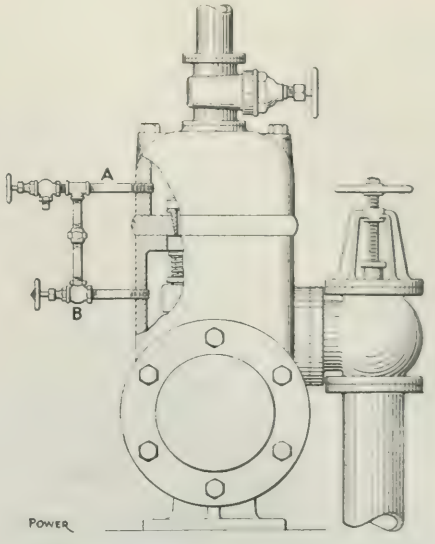


LONGITUDINAL SECTION THROUGH COMPRESSOR

CORRESPONDENCE

Arrangement of Priming Pipes

The illustration shows an arrangement of priming pipes put on two brine pumps that had an 18-ft. lift and used brine of 1.240 sp.gr. It works satisfactorily and has saved much annoyance to the engineers in charge when priming, after having had the pumps shut down for examination, or packing.



PRIMING PIPES ON BRINE PUMPS

The priming pipes *A* and angle valve *B* connect the space below the suction valves with the discharge chamber. The top valve is used for purging out the air, and the lower one for filling the suction chamber to the inlet valve with brine when starting. The pump should be started with the lower valve open. The suction valve should then be opened slowly and the pump will pick up its brine at once. When the pump is working all right the lower valve is closed.

R. T. REZNIEM.

Brooklyn, N. Y.

Measuring the Flow of Air in Pipes

By G. G. Crewson

Some time ago, in a large mill where it was desired to make a proper cost distribution for the compressed air supplied to the various departments, the problem arose of measuring the air velocities in the different pipes. These velocities varied from 500 to 5000 ft. per minute with the air under a pressure of about 80 lb. per square inch and as dry as can be obtained in a large compressed-air system.

The pitot tube was considered to be the most convenient device for the purpose, and in order to determine the accuracy of the various arrangements, an apparatus was fitted up as shown in Fig. 1. The air was admitted to the large tank *A* through the pipe *B* and escaped through an orifice at *C*. Pressure gages showed the pressures at the pitot-tube connection *E* and at the discharge orifice. Different-sized orifices were used in each case, these having previously been calibrated and the flow per minute under a tank pressure of 80 lb. per square inch was determined as follows:

Diameter of orifice, inch	Cu.ft. of air per min.
⅜	191
½	340
⅝	532
¾	765

The pitot tube was inserted in the supply pipe at a point not less than 20 diameters away from the tank or from any bend or fitting in the pipe that would tend to cause eddy currents. Various sizes of supply pipes were also used.

The two sides of the pitot tube led to an ordinary form of U-tube, and at first this feature was troublesome as difficulty was experienced in maintaining tight connections. This was finally overcome to a large extent by making a special form of U-tube with well packed connections and leads of ⅛-in. pipe.

The form of pitot tube first employed was but a piece of ⅛-in. brass pipe bent as shown in Fig. 2. The pressure side extended slightly inside the wall of the supply pipe. This form proved very unsatisfactory, giving varying results for the same conditions.

The next change was to make the velocity tube a straight pipe instead of a bent one. This projected directly into the supply pipe and was plugged at the end with a ⅛-in. hole drilled in the side facing the flow. The results in this case were but little better than before. The opening was then changed to $\frac{1}{16}$ in. without materially affecting the observations.

Many modifications were made in the form of pitot tube used until the one shown in Fig. 3 was adopted. This tube was of brass, turned and threaded to correspond to a ⅛-in. standard pipe to the point where it screws inside of the air pipe. Within the air pipe it has the form of a wedge with a knife-edge, the edge facing the flow, hence offering little resistance. The opening through the tube and the outlet facing the flow were $\frac{1}{16}$ in. in diameter.

These tubes were made in a number of different lengths for different sizes of pipe in order that the opening at the edge might always be near the center of the pipe. The pressure side of the pitot tube was also found to give better results if it did not extend within the pipe but was just flush with the inside.

With this arrangement very accurate results were obtained. At one time, with velocities ranging from 500 to 5000 ft. per minute and in pipes from ¾ to 2 in. in diameter, an average of 50 observations showed, when the velocity was calculated by the formula $V = \sqrt{2gh}$, an average error of less than one-half of 1 per cent. and a maximum error in a single observation of less than 2 per cent.

As explained, this arrangement was first used with an ordinary U-tube but as this can give instantaneous readings only, and in many cases the flow to be measured was very irregular, a recording differential-pressure gage was substituted. By this means records could be kept of the flow of air to any department over any length of time. Line leakages were also quickly detected by means of this record.

For pipes of 1¼ in. diameter and larger, the brass velocity tube was screwed directly into the wall of the pipe, but for smaller sizes standard ties were used, the upper branch being plugged and the plug tapped to admit the pressure and velocity tubes. The tee was then filled with lead and drilled to correspond with the diameter of the pipe with which it was to be used.

In applying this arrangement to an air line, two conditions must be met in the connections leading to the U-tube: First, all connections must be perfectly tight. Any leakage, particularly on the velocity side, due to the small opening will introduce a large error. Every joint should be carefully tested with a soap solution for leakage and if a bubble is observed the joint should be made tighter. The second condition is to have the connections come from the top of the air pipe or, if from the side, have them pitched up at first so as to drain any entrapped water back into the air pipe. If these two conditions are observed, the U-tube or differential-pressure gage may be placed at some distance from the tube connections.

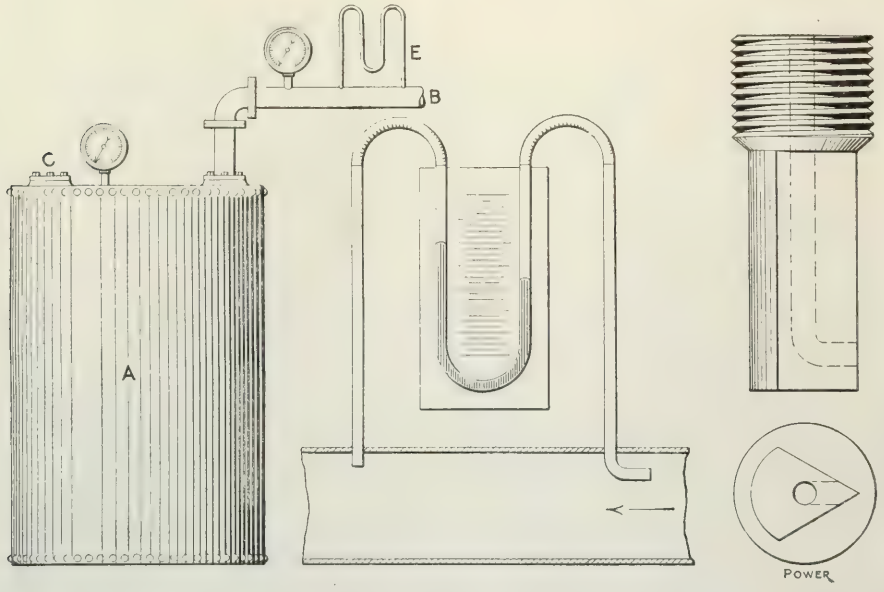


Fig. 1. Arrangement of Apparatus

Fig. 2. First Form of Pitot Tube Used

Fig. 3. Final Form of Velocity Tube

Transmission of Small Water Powers

The present day is one of very large hydro-electric transmission schemes, but in France small water powers have been successfully utilized.

A typical rural hydro-electric scheme is that of Cotentin, on the banks of the River Saire. In this installation the energy in the fall, only 15 to 20 hp., is transmitted over a maximum distance of 4½ miles. During the day the output of the generating plant is utilized to run dairy machinery and at night the current is switched on to lighting circuits.—*Scientific American.*

Recently the city council of Chicago passed an ordinance requiring certain electrical equipment to prevent damage by electrolysis.

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POWER

Technical Museums

To an industrial people the history, development and present state of their industries is of more than sentimental interest. Inspiration is derived, directness of thought and effort fostered, and duplication of research and endeavor avoided by well preserved records and the intelligent exposition and exhibition of processes and products.

Across the water, industrial museums are not uncommon. Scotland has one at Edinburg, England one at London, France one at Paris, Bavaria one at Munich.

In general, the exhibits are historical and educational. The full-sized object may be displayed, or a model to some desirable scale. These are frequently arranged to be operated electrically or pneumatically or by hand. The models, working or stationary, are usually shown sectioned in their essential features, so that their movement is made plain. A brief description, with or without references to standard works on the subject, accompanies the exhibit. Good photographs and drawings are used, generally when the object itself cannot be shown or is not yet illustrated by a model.

We Americans are proud of the progress we have made in a technical way, and we have had world leaders in such lines. But we cannot afford to rest upon our laurels. We must equip our country with additional means of study and information, aiding our engineering and technical schools and presenting to a larger number of our citizens an interesting and concrete opportunity for acquiring knowledge.

Our National Museum is in Washington, and has many visitors each year; but it has not technical exhibits to compare with those abroad. It is not at all what it should be, and is available to only a very small proportion of our people.

England has a fine technical museum; the area of the country is about 58,000 square miles. Scotland has its technical museum; its area is about 30,000 square miles; Ireland has a similar museum and its land is about 33,000 square miles.

The United States comprises 2,870,000 square miles. Except nine of the Eastern states, each one contains 30,000 square miles or more. Many of the states do not have the density of population of the three foreign countires named, but are growing rapidly. In these forty-eight states are universities, supported more or less by the state. All of these can use to advantage such technical museums. Museums in connection with them would be nearer to the people of the state than the national one, and would soon become centers of attraction.

Some universities may not feel the need of such museums at present, but will in the near future. Others already desire them, but lack funds for proper displays. Many have now small beginnings of such exhibits scattered about the different buildings. With a fine technical museum at hand, such objects as are frequently desired in the class or lecture room could be duplicated; others could be visited in the museum.

The English, Scotch and Irish museums are run independently. Each has its mechanicians who make models to scale, and usually only one of each kind. Very seldom is a duplicate made and sent to another museum. If these museums cooperated, each set of mechanicians might make three models, reducing the cost *pro rata* and making the exhibits in all three more complete.

Avoiding this condition, our National Museum could take the lead for this country and help each state acquire a fine technical museum. For example, its director could have his mechanicians make working models say of Newcomen's engine, in lots of 50, 40, 30, 20 or 10, according to how many other museums might wish them, so reducing the cost to each. Similarly the mechanicians already employed in some of the universities peculiarly qualified to make some certain model could be engaged to make them for the various museums. The cost of the models would be paid to the National

Museum and its running expense not increased much by the plan.

If power to follow some such plan is not already in the hands of the director of the National Museum, Congress can pass a bill giving him the necessary power.

In fact, Congress might be willing to grant financial assistance to each state in the building of a proper, fireproof museum for technical purposes. It already allows aid in certain lines, so a precedent already exists. Such museums could be built for the cost of one battleship, and it might be that the interest aroused and stimulated in some of our future citizens by these museums would prove in an emergency of more practical value for offense and defense than the present battleships.

We would suggest this matter of technical museums to our citizens generally, our technical workers, our manufacturers, our farmers, our engineers, our university faculties, our state and national legislators. It would be an excellent thing for the organization of university presidents, already in existence, to urge the necessary measures for carrying out such plans.

Law and Common Sense

It is well to be law-abiding, but one can be punctilious. The Massachusetts law accrediting boiler inspectors employed by insurance companies, provides that they shall be examined and approved by a board of examiners. If a man fails, he may appeal and be examined by another board. Except in cases of such appeal he must wait ninety days before he can try again. If he ceases to be employed by the same company his license lapses; he cannot work for anybody else as a boiler inspector in Massachusetts without a license, and the law says "ninety days must elapse between examinations" so that he cannot get another for three months.

The ninety-day provision in the law, especially as taken in connection with the similar but more specific provision for examining engineers, was obviously intended for the applicant who had tried and failed. An inspector who is stickler enough to apply it to the case of an insurance inspector who has lost his job because his company has gone out of business, is capable of some of the fool questions and decisions, to charges of which we have listened with a reluctant ear.

Low Pressure Steam Tables

For many purposes the steam tables as ordinarily printed are not divided finely enough in the low-pressure section where the values change much more rapidly than in the higher pressure ranges. Some idea of this may be gathered from the following table:

DIFFERENCES IN VALUES BETWEEN

		1 and 2 lb	100 and 101 lb
Temp. deg		24.32	0.8
Sp. vol., cu. ft. lb		59.5.1	0.041
Density, lb cu.ft		0.00276	0.0021
Heat of liquid, B.t.u.		24.2	0.8
Heat of evap., B.t.u.		13.6	0.6
Total heat, B. t.u.		10.6	0.2
Internal energy	evap. B.t.u.	16.2	0.6
	steam B.t.u.	.8	0.2
	water.	0.0422	0.0009
Entropy	evap.	0.0996	1.0017
	total.	0.0574	1.0008

The values differ much less per degree of temperature between the lower and higher temperature ranges, but a difference of one degree makes a difference of nearly 0.2 in. of mercury at 150 deg. and of nearly 0.06 in. at 100 deg. In condenser work the lower range is usually given, not in even degrees, but in inches of vacuum or pounds of absolute pressure, which seldom coincide with the pressures corresponding to the even temperatures for which values are given in the tables. Interpolation is tedious, difficult and unsatisfactory. A table was demanded which would give the properties of steam for each tenth of a pound pressure, and even tenths, at least, of inches of mercury. These, with the intermediate values given by the even degrees of temperature, make a table from which one can find a value so nearly corresponding to that for the conditions in hand that interpolation will rarely be necessary.

Such a table, prepared especially for POWER by Prof. Lionel S. Marks, one of the compilers of the Marks-Davis steam tables, appeared in the issue of July 16, and we have received many pleasant recognitions of the service rendered by its publication. So many of those who have written have expressed a desire for these tables in a form that can be inserted in the volume of steam tables that we are forced to comply with their desires, and the tables will shortly be available in the shape of an eight-page pamphlet of the size of the Marks-Davis book.

An Eight Hour Day

A law recently passed by the Louisiana legislature limits the working day of stationary firemen to eight hours.

It has been the practice in large power plants in all states to operate with three eight-hour shifts, largely influenced by the engineers' and firemen's organizations rather than by the power-plant manager's desire to benefit their men.

It may be contended that the legislature is meddling with the rights of individuals in limiting the hours a fireman shall be employed, and that the law will work a hardship on the owner of the small steam plants who operates his factory ten hours daily. Under this law, he must either reduce the working hours in the factory or employ two firemen for the day's run.

But there is another way of looking at this question. Some states and cities have passed laws prohibiting all who cannot pass a satisfactory examination from operating steam boilers and engines. The object is not to protect them or the plant owner, but the public, which, as a whole, had nothing to do with this legislation.

For instance, a fireman having a proper license secures a position in a power and lighting plant, say on the night shift, which in thousands of small plants is from six p.m. to six a.m., twelve long hours. The twelve-hour day shift is longer than most men can withstand and give efficient service even under the most favorable conditions. When the heat, foul air and oily odors, dust, etc., are considered, let alone the nervous strain attendant on the care of the fillers, the objections to a twelve-hour shift are multiplied.

Such long hours will eventually overtax a man's vitality and mental endurance and unfit him to meet emergencies. On the night shift conditions are often worse because the fireman is deprived of his natural rest, and therefore is working under unnatural conditions. Even under the most favorable surroundings there comes a time when the brain will not act promptly, and men in such a state are not fit to cope with sudden accidents.

Why, then, should not laws be passed preventing employers from compelling their engineers or firemen to work beyond a certain safe limit?

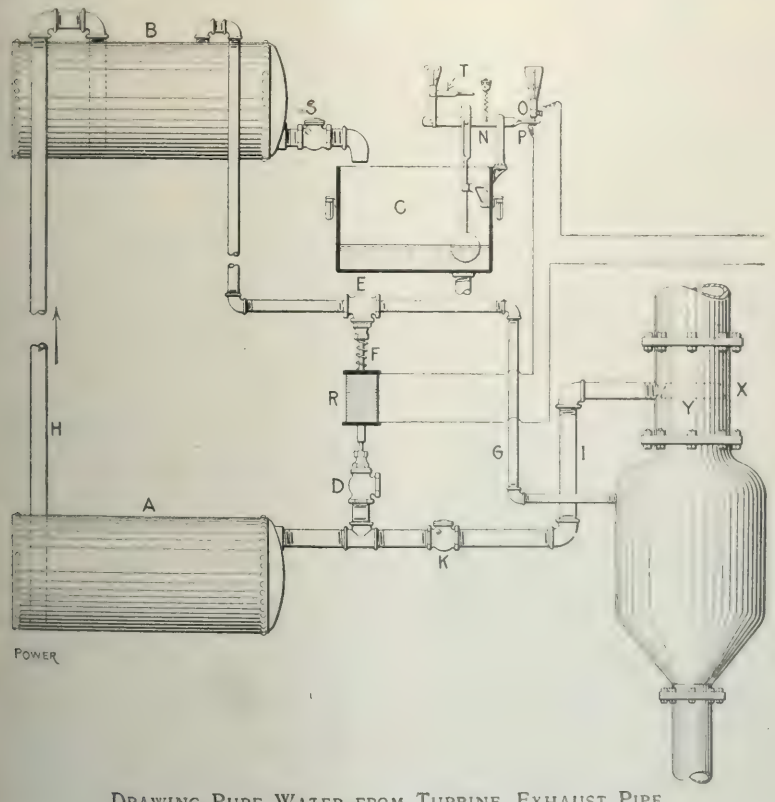
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Pure Water from a Jet Condenser

In a turbine plant equipped with jet condensers considerable trouble was experienced with the bearing glands, due to scale-forming matter contained in the sealing water. To obtain a supply of pure water for this purpose, as well as for house use, the apparatus illustrated herewith was devised.

In the exhaust from the turbine it was thought that there would be considerable moisture running in a thin sheet along the inside wall of the exhaust pipe. To test this a strip of sheet copper was rolled into a ring of ogee section and bolted to



DRAWING PURE WATER FROM TURBINE EXHAUST PIPE

the inside of fitting *X*, so as to form a projecting ring ½ in. wide to catch the water. A hole was tapped in the fitting at the bottom of this ring, and a pipe containing a valve was connected to a small tank. Upon opening the valve and equalizing pressures in the tank and condenser the former was soon filled with water. The apparatus indicated in the illustration was then constructed.

Tanks *A* and *B* have capacities of 60 gal. each, and tank *C*, made of galvanized sheet steel, is open. Pipe *G* is ¼ in. diameter and choked to $\frac{3}{16}$ in. to prevent air entering the condenser rapidly enough to impair the vacuum. At the plant this pipe leads directly to the air pump. The valve in this line is a ¼-in. globe with a loose disk, the collar nut being bored deeper to give the disk $\frac{3}{32}$ in. lateral movement on the stem. The latter can pull the disk from its seat, but cannot push it back, as the travel of the stem is limited to $\frac{1}{16}$ in. by a collar inside the bonnet of valve *D*. The disk is provided with wings working in guides in the valve body to insure proper seating of the valve. Pipes *H* and *I* are 1 in. in diameter. The operation is as follows:

With the condenser in working order, tank *C* full, valve *E* held open and *D* closed by spring *F*, air will be drawn through pipe *G* until a vacuum is established in tanks *A* and *B*. Water will then flow through pipe *I* and check valve *K*, filling tank *A* and pipe *H* to the level of the ogee. When the water in tank *C* reaches its lowest level, the float through lever *N* brings contacts *O* and *P* together, thus closing the circuit through magnet *R*, overpowering spring *F* and opening valve *D* to the atmosphere. Atmospheric pressure in tank *A* forces the water up pipe *H* into tank *B* until the bottom of pipe *H* in tank *A* is uncovered, when air rushes up pipe *H*, breaks the vacuum in tank *B* and continuing through pipe *G* forces the disk of valve *E* to its seat, thus cutting off communication to the condenser.

The weight of water in tank *B*, which is now under atmospheric pressure, opens the check valve *S* and water flows into tank *C*, raising the float until the end of the slotted float rod strikes the latch arm *T*. This releases contacts *O* and *P* so that magnet *R* is deënergized and spring *F* opens valve *E* and closes valve *D*. A vacuum is then created as before, and the operation is repeated.

The capacity of the system is limited by the time taken to fill tank *A*. The installation described furnishes about 200 gal. per hr. Tank *B* was placed about 20 ft. above tank *A* and a pressure of 4 lb. is obtained at the turbine. This seems to be sufficient to seal the bearings perfectly, although the manufacturers recommend a pressure of 15 lb. or more.

The only attention required by the apparatus is an occasional grinding of the check valves, which are of the globe variety, and the removal of rust from the piping, which forms rapidly, due to the frequent admission of air. Brass pipe would, of course, eliminate this trouble.

JAMES L. PIVENS.

Albuquerque, N. M.

Accidents Due to Water in Cylinders

In accounts of engines going to pieces they are usually either being started or shut down; but the explanation as to the cause or precautions taken to prevent a similar accident is regrettably meager.

The first engine I saw wrecked was a 500-hp. cross-compound condensing, direct-connected to a 400-kw. generator, running at 150 r.p.m. It was the engineer's second year on a Chicago license, and this was his first engine. The steam pressure was 160 lb. and the exhaust went to a main exhaust line of about 3000 hp. capacity and was about half loaded. The vacuum on the engine was about 26 in.

There was a horizontal receiver between the two cylinders having a 1-in. drain cock near the bottom with an extension stem running up through the floor. There was no floor in the basement, and water would rise up around the receiver foundation after a rain. It had been raining considerably prior to the accident, and the water was up even

with the top of the drain. As the engine had no atmospheric exhaust, the stop valve leading into the main exhaust line had to be opened before the engine could be started.

The engineer opened this valve and then cracked the throttle and started to close the drain. Just as he stooped down there was a clap in the low-pressure cylinder on the head end and the connecting-rod doubled up like a jackknife. The solution was simple. The water flowed through the drain into the receiver and then into the cylinder; the exhaust valve closed and the entrapped water did the damage.

I have seen locomotives stalled by running into a flock of sheep and getting the wool wound into the crankpin boxes. Years ago I made a stop one night to replace a broken crankpin-wedge bolt in a triple-expansion 1000-hp. pumping engine. After the bolt was tightened, the chief opened the throttle, and as the low-pressure engine came to the crank center there was a loud report and the low-pressure crosshead parted just back of the wristpin.

I concluded the cause was this: He closed the throttle and dropped the knock-off cams on both the intermediate- and low-pressure cylinders (the high pressure was the only one hooked to the governor), and the flywheel and what steam was left in the cylinders and receivers carried the engine around a few more revolutions. This allowed the vacuum in the low-pressure receiver to drop considerably. The pressure when running was about 18 lb. absolute.

He closed the river cock and put on the forced injection so as to have the jet condenser cool when he made the start. The condenser was 10 ft. above the river. It filled and, as the cylinder pressure was less than atmospheric, the water backed up and when the engine was started the exhaust valve closed for compression and the water was "squeezed."

Some years later, the connecting-rod on the same engine was bent to an angle of about 15 deg. when making a start from a dead stop. After the pumps had gained their suction and the engine came up to speed, the reach rod loosened from the wristplate, due perhaps to its not being properly latched. This let the receiver pressure drop and, as it had some water in it, the cylinder "took a drink."

I was oiler on a lake steamboat when we got a hot crankpin shortly after we cleared. Everybody was more or less excited, as we were in a bad place to stop and the wind was blowing a gale. The majority of the passengers were sick and nervous and when the engine stopped they thought they were going straight down. Water backed up into the low-pressure cylinder, and when we started the valve was forced back from its seat, bending the stem so badly that the cylinder was useless until the stem was removed and straightened. The second engineer was handling the engine while the chief and fourth engineer stood by to put the shims in between the crankpin brasses. He left the forced injection water on too long and it backed up, doing much damage. We had to put back to port.

No engine can be made foolproof, but an engineer should take every precaution, if he can get the necessary money, which is about half the work.

I have put atmospheric checks on all low-pressure receivers and checks in their drain pipes, a good trap on all intermediate- and low-pressure receivers and a tee in place of the ell running to the jet condensers. I have a hinged valve on this tee that can be worked from the operating stand. As soon as the engine comes to a stop I open this valve and close it as soon as the engine turns over a few times.

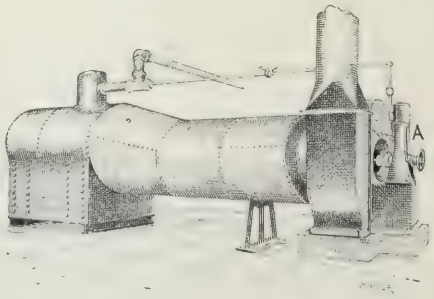
SAM H. FARNSWORTH.

Chicago, Ill.

Stationary Locomotive Boiler Test

Railroads, when they need a stationary boiler, are accustomed to use an old locomotive boiler stripped of its running gear, and with a steel stack 40 to 60 ft. high.

These boilers do not prove very satisfactory, as the horsepower developed is much below that obtained on the road. At a certain railroad roundhouse, a water-softening plant was installed and my job was to set the boilers. I long had a notion that by proper erection these boilers could do better, and proceeded to put my idea in practice.



LOCOMOTIVE BOILER FITTED WITH INDUCED DRAFT

I concluded to set this boiler somewhat as it was designed to be set, and operate it as it was intended to be, by leaving in the baffle-plates, exhaust nozzles and brick firebox arch. I took down the stack, closed the opening and cut one in the bottom of the extension front, and connected an induced draft fan, as shown at *A* in the illustration. Then a baffle-plate was put in to equalize the draft through the flues (the draft being from the bottom, the baffle was the reverse of what it would be when the draft was at the top). A brick arch was put in and the boiler fired up. The improvement was so marked that a test was run to determine what the boiler would do.

The efficiency was highest when the output was greatest and the uptake temperature highest. The uptake temperature was lower when the draft was 1.73 in. than when it was 0.59 in. This was probably due to the scouring action of the gases when passing through the flues and that the gas was swirled, and had no hot core. Note, 255 hp. with 16 sq.ft. of grate and 1 hp. per 3.42 sq.ft. of heating surface.

TEST RESULTS OF LOCOMOTIVE BOILER

HEATING SURFACE			
Firebox	87.7 sq.ft		
Tubes	792. sq.ft		
Total	879.7 sq.ft		
Grate surface	16. sq.ft		
Duration of test	3 hr. 59 m.	2hr. 51m.	1 hr. 33m
Draft in water	0.59	1.09	1.7
Steam pressure lb.	99	97.9	101.
Temperature feed water deg.F.	57.6	57.5	58.
Temperature flue gas, deg. F.	830.	756.	802.
Total coal burned, lb.	2,400.	2,800.	2,000.
Coal burned per hour, lb.	602.	984.	1,290.
Combustible burned per hour, lb.	454.	742.	973.
Total water evaporated lb.	12,418.	15,812.	11,374.
Water evaporated per hour, lb.	3,117.	5,548.	7,338.
Factor of evaporation.	1.2		
Water evaporated per pound coal, lb.	5.17	5.65	5.6
Water evaporated per lb. coal at 212 deg.F.	6.20	6.78	6.8
Water evaporated per lb. combustible at 212 deg.F. lb.	8.22	9.00	9.0
Coal burned per sq.ft. grate per hour	37.4	61.5	80.6
Horsepower developed	108.	193.	255.
Sq.ft. heating surface per hp.			3.4
B.t.u. per lb. coal	9,550.		
Percentage of ash	13.7		
Efficiency	62.8	68.5	68.9

I have installed induced draft on about a dozen such boilers, with highly satisfactory results.

E. J. SAXE.

Christopher, Ill.

Results in CO_2

You have urged the engineer to purchase a flue-gas analysis instrument and keep pace with modern engineering practice and you print, from time to time "highbrow" articles about it, and then let the matter drop. You offer indicator diagrams for criticism, and I venture to say that 60 per cent. of your readers examine them and all unconsciously figure how they could be improved.

Why cannot the flue-gas analysis be worked from the same standpoint? Having given a certain size boiler, grate and load, with gas analysis, draft and flue-gas temperatures and firings, and a certain grade of coal, criticize the results obtained, suggest how they can be improved with ideas as to how to fire, and how the draft and air supply should be regulated. To start the ball rolling I send herewith, some results taken from a han

fired 350-hp. Babcock & Wilcox boiler for criticism.

HARRY A. COX.

Middletown, N. Y.

350 H.P. B. & W. Boiler — 76.5 sq.ft. grate area
Coal No. 1 Buckwheat — Avg. Load 225 Boiler H.P.
Temp air - 80° F. — Avg. Steam 145 lb. gage.

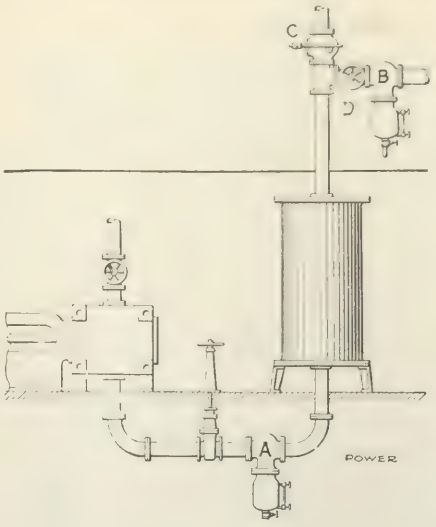
Time	CO_2	Draft at breeching	over fire	Flue gas Temp	Remarks
9 a.m.	11.8	0.39	0.33	380	Damper ¼ open
9:05	11.2	0.40	0.34	385	
10	14.0	0.29	0.22	395	Damper nearly closed
15	11.6	0.38	0.32	395	
20	13.9	0.39	0.33	395	
25	14.0	0.39	0.33	410	
30	15.1	0.22	0.15	405	
35	13.0	0.37	0.31	405	
40	14.6	0.36	0.30	408	
45	7.0	0.52	0.46	420	Damper wide open. Cleaning fire.
50	13.4	0.51	0.45	415	
55	5.4	0.50	0.45	425	Cleaning fire.
10—	11.0	0.50	0.45	425	
05	13.8	0.43	0.37	417	
10	11.6	0.41	0.35	405	
15	15.8	0.40	0.34	402	
20	11.8	0.38	0.31	397	
25	13.1	0.20	0.13	390	
30	11.0	0.41	0.35	402	
p.m.					
7:15	14.0	0.36	0.30	397	Avg. Load 375 Boiler H.P. Steam pressure 151 lb.
20	14.0	0.40	0.34	410	
25	5.0	0.34	0.28	400	Fire door open Temp. air 73° F.
30	12.0	0.39	0.33	415	
35	12.6	0.52	0.46	415	
40	16.1	0.51	0.46	415	
45	16.4	0.51	0.46	430	
50	15.1	0.51	0.46	440	
55	15.8	0.44	0.38	445	
8—	12.1	0.56	0.50	420	
05	16.6	0.53	0.47	440	
10	13.6	0.53	0.47	445	
15	15.6	0.50	0.44	436	
20	12.8	0.48	0.42	425	
25	12.4	0.51	0.45	435	
30	13.4	0.51	0.45	437	
35	12.2	0.56	0.50	430	
40	14.6	0.51	0.45	435	
45	13.0	0.52	0.46	430	

A Misplaced Separator

A recent inspection of the plans and specifications for a large addition to a power plant showed the oil separator between the engine and the heater. One reason for this design is that the exhaust steam will be purified before it goes into the heater, and the latter, if of the closed type, will not be made inefficient by the accumulation of sediment from cylinder oil. Another point in its favor is that steam condensed in a closed heater may be returned to the boilers. A separator, however, will do better work if placed beyond the heater. Drops of water in the exhaust steam after it passes the heater assist in the process of eliminating oil.

The illustration shows the heater and part of the exhaust-steam piping in another power plant, in which a change has recently been made. The separator *A* was located near the engine under the floor, but when put into service it did not purify the exhaust steam. An expert steam engineer reported that the separator should be located beyond the heater, and as there was not enough head room for it in the engine room, it was located in the room above, as shown at *B*. When the engine is exhausting to atmosphere, steam does not go through the separator, but if it did there would be less grease on the roof.

During cool weather the back-pressure



LOCATION OF OIL SEPARATOR

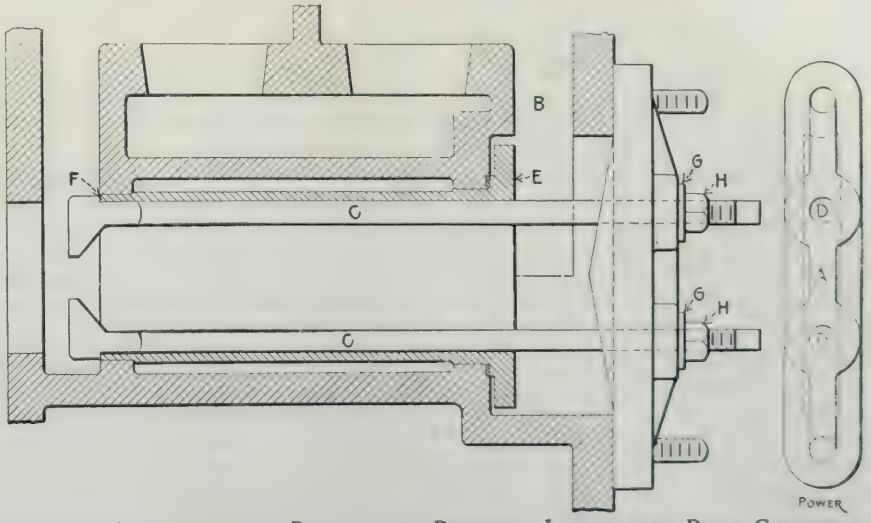
valve *C* is closed, and *D* opened, so that all exhaust steam goes through the separator, is delivered to the heating system free from oil and can be safely pumped back into the boilers.

W. H. WAKEMAN.

New Haven, Conn.

Device for Removing Water Cylinder Liners

The removal of liners from the water cylinders of pumps is usually a troublesome job, especially if the liner is stuck fast as it often is.



SHOWING APPLICATION OF DEVICE FROM REMOVING LINERS FROM PUMP CYLINDERS

This trouble can be avoided and their removal made easy by the use of the device shown. The casting *A* is slipped over two studs after the cylinder head has been removed as at *B* and then the two pull bolts *C* are pushed through holes *D* in the casting and made to engage the back end of the liner *E* as at *F*. The washers *G* and nuts *H* are then tightened to hold the bolts in place. By screwing up on the nuts the liner can be started and thereafter easily removed. The device may also be used to drive the new liner in place.

W. H. WOLFGANG.

Toledo, Ohio.

Care of Electric Elevators

In the plant under my charge there are three electric passenger elevators having motors operating under 220 volts. As I am not very familiar with electric elevators, will some interested reader tell me, through POWER, the best way to manage and care for them?

What I particularly desire is an explanation of the counter-balancing of elevators in general, a brief enumeration of the chief sources of trouble, and the proper way to avoid them in electric elevator operation.

L. A. TROWBRIDGE.

Cambridge, Mass.

Doubts Economy of Uniflow Engine

It seems impossible to me that this type of engine can run with economy at any load either condensing or noncondensing. To those who understand its economical features, I would ask a few questions:

Why does excessive compression give more economy with this engine than with others? Why does this engine show better results with overloads than others? What good is a condenser except to reduce the exhaust pressure, and if it is of advantage to reduce this pressure, why is it not the opposite to increase it?

L. JOHNSON.

Exeter, N. H.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Cotton Rope Drive

A. Rauch has been particularly unfortunate in his experience with cotton rope drives, as related in the July 2 issue.

From the figures he gives the conditions should be favorable to a very satisfactory drive. Surely something must be abnormal when his ropes will not run three weeks without breaking. It would be a help in diagnosing the trouble to know what horsepower is being transmitted by each rope.

I once had a cotton rope drive which gave out in 12 months. The velocity was about 5000 ft. per min., and the ropes were 1½ in. in diameter. These ropes were put on during structural alterations and became filled with lime and cement dust. After 12 months we removed them and put on another complete set of 1½-in. four-strand cotton ropes. The last time I saw this drive it had been running nine years and the ropes were still in good condition.

I am with a firm having a number of mills, one of which is driven with 112 cotton ropes which have been running four years. Fifty of these ropes are now overloaded and are being replaced by cotton ropes of slightly larger diameter, as the pulleys will permit of this change without overcrowding the grooves.

I once had trouble with one 1500-hp. drive which had 1½-in. cotton ropes running at 7000 ft. per min. These ropes used to jump the grooves, apparently because their period of vibration synchronized with the beat of the engine. We eventually corrected this by putting on 1¾-in. ropes. These were rather large for the grooves but pulled down after a few weeks' work.

I have never had the misfortune to have to stop after six hours' run to tighten ropes. We have had to tighten them after a week's run but that tightening lasted for years. Mr. Rauch should look into his drive again, because, from his figures, there is nothing to prevent his getting a satisfactory life out of cotton ropes. We have many thousand horsepower transmitted by cotton ropes and I have not come across a similar experience to Mr. Rauch's. If he can give any further information it might help us to avoid such trouble as he encountered.

My experience is that 10 to 15 years' service can be had from ordinary cotton ropes under normal conditions.

I have had no experience with manilla ropes nor with the American continuous cotton-rope drive; our drives are all parallel.

JOHN GILLEAN.

Paisley, Scotland.

Another Experience

The foreword in the July 9 issue reminded me of my own experience.

I was only about 18 years old at the time I took charge of my first plant, which consisted of an old 60-in. by 16-ft. boiler, and an old rattle trap of an engine about 14x20 in. The whole plant was very dilapidated, and, in addition to burning all the shavings and refuse from the mill, sawdust and wood from an adjacent sawmill were used.

I first cleaned the boiler, which was very badly scaled from using feed water carrying a high percentage of lime. All the other plants in the neighborhood used some boiler compound, but the owner of this one would not furnish any. "It cost too much and was no good anyway," he said. After knocking off all the scale I could by hand, I made a feeder, and began using a weak solution of lye made from wood ashes, as a water softener.

Next, I reset the valves on the engine and packed nearly every joint, valve and rod on the place; cleaned the boiler tubes and combustion chamber, as well as the heater, and then began to see some results.

Some three weeks later, after we had cut off the extra fuel from the sawmill and were running on our own refuse, the owner saw me sitting in the doorway reading POWER. I was apparently at ease with the world. "Well!" said he, "You sit around more than any other man on the job; haven't you anything to do?" "While I am sitting here I am not shoveling fuel into the furnace, and you have that much less to pay for," I replied. The conversation immediately shifted to the state of the weather, and I spent a good many pleasant hours' reading and studying in that same doorway without any further interference.

The next plant had a somewhat larger engine equipped with a Meyer cutoff valve. The proprietor, in showing me around, paused in front of the mysterious little handwheel at the back end of the steam chest, and assuming an awed air, told me to "never, never change that wheel; it is set just right and must never be changed." However, I found out in a few days that several turns of that same wheel in the proper direction would save lots of work with the scoop, so I made what changes I wanted without consulting him. But even then, the work was so hard that I turned it over to another engineer with more muscle.

There are many owners of this type in charge of small plants and I have found that the only way of dealing with them is to be independent, and insist on having absolute charge of the plant, or else leave it to others willing to put up with that kind of treatment.

F. P. READ.

Memphis, Tenn.

Remedy for Unequal Cutoff

LeRoy Whitney's trouble cited in the July 9 issue is likely caused by a bent or worn governor connection.

The eccentric also appears to be too far back, causing a late admission and wire drawing the steam even at slow speeds. I would suggest that he set the valves all over again, checking each part, and fitting on a new block and catch if necessary to hold the head-end valve open during the stroke with the governor on the block. When this is done it will be simple to apply the indicator and equalize the cutoff.

J. M. COLEMAN.

Ipswich, Mass.

Oiling a Prony Brake

The account of a prony brake test, given by Alan A. Slade in the Apr. 16 issue, reminds me of a similar experience with a 60-hp. tractor.

The brake pulley was hollow, 34 in. in diameter, with an 11-in. face. The shaft was mounted in roller bearings and a 6-ft. beam was used. As Mr. Slade suggests, the parts were made as light as was consistent with safety. The brake band was made of hard maple blocks set so that the wear came parallel to the grain. The pulley had a water-feeding device and a scoop for removing the water as it heated.

Although the brake was water-cooled it heated excessively, evidently being overloaded. A one-pint glass-body sight-feed oil cup was arranged to deliver oil directly to the face of the pulley. In addition a ¾-in. hole was bored in the frame and a mixture of lard oil and graphite dropped on the pulley face at

regular intervals. With this regular oil and graphite lubrication very satisfactory tests were made, though we got a great deal of steam from the interior of the brake pulley. It was much easier to keep the beam balanced when oil was used.

This brake has since been supplanted by an electric dynamometer which we find is a very successful testing machine.

C. V. HULL.

Charles City, Iowa.

Valve Motion and Rods of Duplex Pump

If N. H. J., who asks questions on page 69 of July 9 issue, will turn to POWER for Apr. 5, 1910, page 646, he will find those questions answered fully. If he will then draw a diagram of a duplex valve motion having both valves driven by a lever of the same class, he will see how impossible it is for a pump to work with the form of valve movement of which he inquiries.

When pump rods wear and turning them down seems necessary, do not forget that these rods must be packed. Do not turn them so that a 5/8-in. packing is just too large and a 7/8-in. just too small; turn to a definite size. If the pump is a duplex do not turn the rods different sizes as is sometimes done. It means continual trouble for the man who does the packing. Just use a little forethought and find the smallest place on each rod. Take the smaller of the two measurements and fix upon a diameter that it will be possible to fit with packing. Then turn both rods alike.

WILLIAM E. DIXON.

Malden, Mass.

Shorter Cutoff Greater Load

Concerning the article on the above subject on page 180 of the issue of Aug. 6, my idea is as follows:

Assume an engine to be running with an average load that is constant, with the receiver pressure just equal to the terminal pressure in the high-pressure cylinder.

If the cutoff in the low-pressure cylinder is shortened, the receiver pressure rises, because the low-pressure cylinder is not taking from the receiver as great a volume of steam as is being delivered to it by the high. The receiver pressure is then higher than the terminal in the high and a diagram from this cylinder will loop. The loop represents negative work, and to make up for this, the engine will slow down to a certain extent, thus lengthening the cutoff in both cylinders, with a tendency to still further raise the receiver pressure. On shortening the cutoff again practically the same results would occur, and so on.

To make the argument more clear, we will assume the high-pressure cylinder capable of taking steam full stroke. Then if we keep shortening the cutoff in the low, the receiver pressure will continue to rise, and the cutoff on the high continue to lengthen until full stroke is reached. Boiler pressure will then exist in the receiver, or will when the volume at cutoff in the low is equal to the total volume of the high. Then with a short cutoff in the low-pressure cylinder, with boiler pressure for the initial pressure, we can see that this cylinder could carry practically all the load.

While we have been shortening the cutoff in the low, the governor has been lengthening the cutoff in both cylinders, and the low pressure would have all the load with the governor running in a lower plane. If we had an engine designed for a cutoff up to 3/4 or 7/8 of the stroke, as most long-range cutoff engines are, and desired to put all the load possible on the low-pressure cylinder, we could shorten the cutoff on it until the high-pressure cutoff neared its limit. The governor would then revolve in a lower plane, having less range in admitting more steam and a wider range to reduce the cutoff in both cylinders. This would probably not be a satisfactory manner in which to operate an engine of this kind, but is intended only to show that the load can be shifted from one cylinder to the other in a very marked degree, by altering the cutoff in the low-pressure cylinder.

I may be wrong in my belief on this matter, and no one will be better pleased than myself to be shown wherein lies the error or errors in my argument, as this is a question on which many engineers are hazy and which they are anxious to understand thoroughly.

JOSEPH STEWART.

Hamilton, Ohio.

Hoarding Knowledge Hinders Progress

I think the foreword under this title in the June 11 issue was very timely and contained good advice. I have often heard people criticized for "giving their ideas or brains away."

If an engineer or mechanic knows a little kink or improvement that will help someone, it is more to his credit to give it out freely than to keep it to himself, thus "hiding his own light under a bushel," so to speak. But if he feels he ought to be paid for it, he does well to publish it.

He at least has the satisfaction of feeling that he has not had his trouble for nothing and it may do some other fellow good. By being generous in small things he gains later on in large ones.

GEORGE J. LITTLE.

Passaic, N. J.

Exhaust Steam Turbine

The possibility of increasing the electrical output of a small- and medium-sized power plant, without correspondingly increasing the steam-generating equipment, by installing exhaust-steam turbines, is of great general interest to all operating engineers.

POWER'S editorial of July 2 suggests a golden opportunity for wide-awake, progressive engineers to prove to their employers that they are worthy of the consideration and remuneration due the man of ideas and initiative.

Were you to be called upon tomorrow by the "old man" to advise him as to how your power plant might best be increased in capacity; could you discuss the subject intelligently? Could a low-pressure turbine be used advantageously in the plant under your charge? What definite, specific facts, backed up by actual cost figures have you to present to substantiate your arguments? If you have not gone into the subject, you may be missing one grand chance for advancement.

The low-pressure turbine is not a cure for all power-plant ills. It may not work out as worth while in your case. While you may convince yourself that to use a turbine is advisable, you will probably be unable to induce the boss to agree with you in your conclusions. Take a chance. Work it out. Your assumptions, figures, and conclusions, and even perhaps the reception accorded your suggestions by the boss, might be of sufficient interest and educational value to warrant publication in POWER. In which event your labors will have a triple value. First and foremost, the careful consideration of an engineering problem of this character will be of value to any engineer. Your findings may prove of value to other engineers, and finally you will receive a monetary remuneration for your effort, even though, perhaps, not directly through the boss.

A paper mill in New England has just installed a low-pressure turbine to more fully utilize the exhaust from an overloaded Corliss engine. The layout of the plant made it possible to substitute several motor drives for considerable shafting and belting. The results obtained are most gratifying to the owners. A very considerable increase in power was made with no increase in steam-generating capacity. Regrettably, installing the low-pressure turbine was suggested, not by the engineer, but a consulting engineer called in to prepare plans for extending the boiler plant.

A remarkably inexpensive and apparently efficient condensing plant is used in this installation. The condenser is of the so-called ejector type. Injection water is supplied by a steam turbine-driven volute pump. No dry vacuum pump, or vacuum pump of any kind is used, yet

a steady vacuum of 28 in., as indicated by a mercury column, is being regularly maintained. As a simple test showed that the ejector will raise its water supply 20 ft. automatically, the power required for operation must be very low.

The use of this simple type of condenser for high vacuum is proving a source of comment to many engineers who have visited the plant.

JOHN W. BRUCE.

Holyoke, Mass.

Taking Chances with Boilers

F. J. Naylor's experience with a defective boiler, as related in the July 23 issue, reveals how some owners will take chances with defective boilers, thereby endangering the lives of their employees, to say nothing of the loss of property that might result from an explosion.

Some inspectors are very slack, or worse in making inspections. In a large mill where I was employed as assistant engineer, I noticed one morning an unusual activity on the part of the chief engineer and his helpers. I inquired into the cause, and found that the boiler inspector was coming the next day. The fire pump which had been out of commission for four days was repaired, and a month's accumulation of soot was cleaned from the boilers and everything fixed up as presentable as it could be. The next morning, which was Sunday, three of the five boilers were cut out and cooled down. The inspector came into the boiler room dressed for entering the boilers, asked the chief a few questions, felt of one of the boilers, said they were too hot to enter and went out. On the following Monday night a gage-glass broke on one of the inspected boilers. They had to send out to another mill to get one. The fireman tried to locate the water with the gage-cocks, and on finding them plugged with scale, cut the boilers out and pulled the fire. One machine had to be shut down. He was promptly fired the next morning.

The boilers were inspected that way for about two years, when one Sunday morning a stranger stepped into the boiler room and stated that he was an inspector. The chief called up the owner and he came down, but the inspector refused to be bluffed, and things began to happen right away. He said it was the worst boiler room he had visited in two years. The steam gages were tested, two of which were defective. He refused to let them run the boiler until perfect ones were obtained. He forbade them running one of the boilers until retubed. When the old tubes were removed there was a mass of a sort of porous scale from the second row of tubes from the bottom up to the third row down from the top. He stayed in town three days, and when he went away the chief said he hoped he never would come again. But he did, and always without any warning, too. The result was things were kept in better condition. The chief or owners were not far-sighted enough to see the saving which resulted, and were always complaining about how strict the inspector was.

V. C. WOOD.

Copenhagen, N. Y.

Explosion of Air Receiver Due to Oil

The writer of the article which appeared under the above caption in the July 30 issue is evidently firmly convinced in his theory that the receiver failure was due entirely to the ignition of the oily coating on the inner surface of one head and the consequent weakening of that head. In presenting this theory, however, he neglects to furnish the details of construction that would permit calculations showing whether the receiver was safe for the pressure carried.

That thousands of air-pressure receivers are in use under the same conditions as stated makes it imperative that before entering into any theorizing on oil ignition, the safe working pressure of the received should be carefully ascertained.

The statement that rivets ⅜ in. in diameter were used to fasten the head to the shell seems sufficient to arouse suspicion that the receiver was structurally weak for the pressure carried, and the photographed head in Fig. 4 has the appearance of an ordinary flat head, or, of being bumped to a much larger radius than the pressure would warrant. If this is true, and the head that failed was similar and flanged inward, the cause is apparent.

If this oil-ignition and combustion theory has any tangible basis every air reservoir carrying 100 lb. pressure or more is likely to explode at any moment, no matter how strongly it may be constructed. Therefore, I would urgently request Mr. Richards to send in the following data on the receiver's construction: material, thickness of shell plates, type of joint, pitch of rivets, thickness of heads, and radius to which the heads were bumped.

The article states that the receiver was constructed "with dished heads; the convex side of the head was outward at one end, and the concave side outward at the other end, which is the too common practice, just for convenience in riveting in the last head, and for no other good reason." This leads me to ask the writer what other way he can suggest for fastening the heads of small receivers, also what objection he has to this form of construction, if the heads are made of proper material and thickness, and bumped to the proper radius, or stayed as the conditions may require.

JOSEPH KING

Lynn, Mass.

Plant Economy

W. J. Bailey's letter under this heading in the issue of May 21 interests me. While the most economical plan for his plant cannot be determined without more data, it would seem that the procedure contemplated would be a move in the wrong direction for the desired results.

I would suggest that the whole plant be run at 150 lb., and that the high-speed units be run condensing and also other steam-driven equipment, the exhaust of which is not needed for heating. The exact economy in coal depends on a number of conditions; in fact, the cost of water may make my suggestion entirely impracticable, but in any case I should not attempt to run the engines with 75 lb. back pressure, or the pumps at 75 lb. pressure if more pressure is available.

L. JOHNSON.

Exeter, N. H.

While Mr. Bailey's intended experiment appears interesting, I cannot see how any saving could result or the plant be any more economically operated. By keeping 85 to 90 lb. steam pressure and operating the engines, air compressor, and pumps from the same header (which I am sure is of ample size, being 14 in. in this case) he can get full service from all the units, at a minimum cost for packing, upkeep, lubrication, fuel and a hundred and one other things that will give absolute satisfaction under 100 lb. pressure, but certainly would give trouble under 150 lb. pressure. Moreover, reducing valves would then be unnecessary.

Under his plan considerable condensation can occur between the power units and the air compressor and pumps which, by the way, are the meanest of all engines I have ever had anything to do with, to operate with wet steam. This condensed steam could be handled very well by installing separators of sufficient sizes, but a steam separator is a device to take wasted energy from that which may be utilized; hence another waste. Then, I would not think it wise to put an engine designed to operate at 100 lb. pressure where it would have to operate at 150 lb. If I tried such an experiment I would expect results not very pleasing to the boss.

As Mr. Bailey does not say how he intends to operate the air compressor and pumps when the power units are not running, and as he says he will have a specially designed back-pressure valve to relieve No. 3 boiler if the pop valve proves too small in case of overloads, there is evidently no intention of firing this boiler after it is converted into a "receiver."

P. T. BARKER.

Stonega, Va.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Percentage of Slip in Motors

Why must an induction motor operate with slip, and is there the same percentage of slip in all sizes of motors?

H. E. H.

In an induction motor, the current is induced in the rotor by the latter cutting the rotating field flux. This would not be so if they were in exact synchronism, as no current would be induced; therefore no torque would be exerted and the motor would not run. The amount of slip of induction motors varies with their size. In large motors of 200 to 300 hp., the slip may be as low as 1 per cent., while in 1-hp. motors it may be as high as 20 per cent.

Free Air Capacity

What is meant by the free air capacity of an air compressor or a blower?

J. E.

As commonly used the term refers to the number of cubic feet of air at atmospheric pressure and ordinary temperature received per minute for the operation of compression or delivery in form of forced draft.

Width of Keyways

What is considered good practice for the width of keyways for shafting?

E. J.

Make the width of the keyway equal to one-fourth the shaft's diameter.

Instruments Required For Boiler Trial

What instruments are required for making an evaporative trial of a boiler?

Scales for weighing coal and ashes; tanks or "liquid weighers" for gaging or weighing the feed water; thermometers and pyrometers for taking temperatures of air, steam, feed water, waste gases, etc.; steam-pressure gages and draft gages; and a colorimeter for determining the percentage of moisture in the steam. Local barometric pressure of the atmosphere at the time of making an evaporative trial should be known. In addition to the instruments mentioned a hydrometer for determining the amount of moisture in the atmosphere and apparatus for analysis of flue gases are highly desirable for studying the character of combustion, although these are not as essential for conducting an evaporative trial as they are to determine whether conditions were favorable or unfavorable for obtaining best results.

Glass Gages on Vertical Boilers

What is the proper height to set a water gage-glass on a vertical boiler?

H. H.

Vertical fire-tube boilers of ordinary proportions should have their water columns or glass-gage fixtures set at such a height that the lowest visible part of the glass gage will be not less than one-third the length of the fire tubes above the lower tube sheet.

What Pressure for Receiver

What pressure should be carried in the receiver of a cross-compound condensing engine 12x22½x18 in. (i.e., cylinder ratio 3.51), the initial pressure being 110 lb. and the vacuum 26 in.?

F. A. F.

In addition to the data given it is necessary to consider the desired distribution of load between the two cylinders (generally equally distributed), the point of cutoff and back pressure of the high-pressure cylinder. See "Equal Work in Compound Engines," by F. R. Low, in POWER of July 18, 1911, pages 88 to 91.

Cast Iron Fittings for High Pressure Steam

Are cold-water pressure tests sufficient to warrant the safety of cast-iron fittings and appliances against rupture when used for high-pressure steam?

M. D.

No, such tests may be sufficient to reveal blow-holes or cold-shut castings, but will not determine whether rupture is liable to result from changes of form due to changes of temperature as well as from pressure, nor afford assurance of strength to resist distorting reactions of expansion or contraction of connected piping and other appliances. As a material of construction, cast-iron is proverbially treacherous, and when fittings and appliances made from it are employed in high-pressure work they should be regarded as hazardous until thoroughly tried out for their especial purpose and situation.

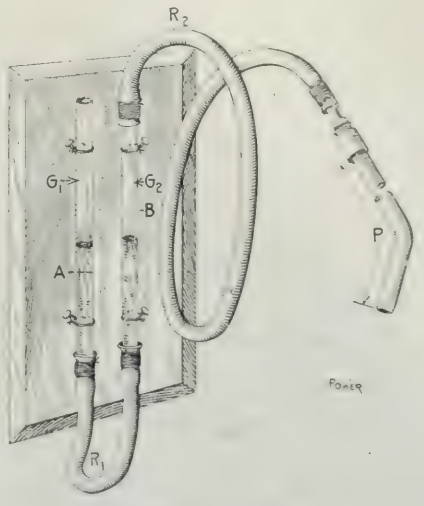
Measurement of Force of Draft

How is the force of a flue or chimney draft measured?

W. B.

Force of draft is usually expressed in inches of water, meaning pressure equivalent to that of a column of water which would be supported by the difference between the pressure of the flue or chimney gas and the pressure of the atmosphere. It is generally measured by the difference in levels of water columns of a U-tube manometer having one end open to the atmosphere and the other in communication with the chimney flue or passage in which the force of draft is to be measured. The U-tube may be made of glass tubing bent as the letter U or improvised as shown in the illustration.

G_1 and G_2 are two ordinary boiler gage-glasses joined at their lower ends by the rubber tubing R_1. One of the columns of the U-tube as G_1 has its upper end open to the atmosphere, while the upper end of G_2 is connected by a rubber tube R_2, and a nipple to a piece of ½-in. pipe P about 4 ft. long, bent over at its open end. Water is poured into the open end of G_1 until it stands about half way up the length of both glass tubes. The open end of P being inserted in a flue or passage where the force of draft is to be measured, then as the atmospheric pressure is greater than that of the flue gas, the water is depressed in the tube G_1, as to A, and rises in G_2, as to B, the difference in levels A and B thus assumed being the height of water column due to the unbalanced force of draft. By turning the pipe P in different directions its bent open end will be presented for measuring the force of draft in different directions or parts of the flues and passages.



DRAFT GAGE

Study Questions

This Week's Questions
Last Week's Answers

(86) If a gas under constant pressure occupies a volume of 10 cu.ft. at 32 deg. F., what volume will it occupy at 212 deg. F.?

(87) How many pounds of ⅛-in. sheet lead will it take to line the sides and bottom of a rectangular tank 6 ft. long by 4½ ft. wide, holding two tons of water when filled within 6 in. of the top?

(88) A belt-driven triplex pump delivers 3.2 gal. per revolution of its driving pulley, which is 18 in. in diameter. It is to be belted to a line shaft running at 80 r.p.m. What size pulley must be put on the shaft to cause the pump to discharge 400 gal. per min.?

(89) In heating 1 lb. of water from 50 to 120 deg. F., how many heat units are absorbed by the water and what is the equivalent in units of work?

(90) An engine develops 100 hp. at 80 r.p.m. with a mean effective pressure of 36 lb. Required the diameter of the cylinder if the stroke is 3½ ft.

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(81) One pound of the condensing water contains 50 — 32 = 18 B.t.u., then 36.71 lb. contains

$$18 \times 36.71 = 660.78 \text{ B.t.u.}$$

The total heat in 1 lb. of steam at 216 deg. F. (from steam tables) is 1151.9 B.t.u. Then 36.71 + 1 = 37.71 lb. of the mixture contains

$$1151.9 + 660.78 = 1812.68 \text{ B.t.u.}$$

and each pound

$$\frac{1812.68}{37.71} = 48.07 \text{ B.t.u.}$$

Then its temperature must be 32 + 48.07 = 80.07 deg. F.

(82) Atmospheric pressure at sea level is 14.7 lb. per sq.in. abs. A piston 3 ft. in diameter has an area of

$$0.7854 \times (3 \text{ ft.})^2 \times 144 \text{ sq.in.} = 1017.9 \text{ sq.in.}$$

Then the total pressure acted against is

$$14.7 \times 1017.9 = 14{,}963.13 \text{ lb.}$$

and acting through a distance of 5 ft. the work done is

$$14{,}963.13 \times 5 = 74{,}815.65 \text{ ft.-lb.}$$

(83) The diameters of the gears will be inversely as the speeds. Let

D = Diameter of the larger gear;
d = Diameter of the smaller gear.

Then

$$\frac{D}{d} = \frac{240}{192} \qquad (1)$$

Since the pitch circles will be tangent, the sum of the radii of the two gears will equal the distance between the shaft centers, or

$$\frac{D}{2} + \frac{d}{2} = 18 \text{ in.}$$

From this

$$D = 36 - d \qquad (2)$$

Substituting in (1)

$$\frac{36 - d}{d} = \frac{240}{192}$$

Simplifying

$$6912 - 192\,d = 240\,d$$
$$432\,d = 6912$$
$$d = 16 \text{ in.}$$

and

$$D = 36 - 16 = 20 \text{ in.}$$

(84) The coefficient of expansion for mild steel is 0.0000067. Then

$$0.0000067 \times 60 = 0.000402 \text{ ft.}$$

or

$$0.000402 \times 12 = 0.004824 \text{ in.}$$

is the amount each rail will expand per degree rise of temperature, and for 98 — 32 = 66 deg., the expansion would be

$$0.004824 \times 66 = 0.318384 \text{ in.}$$

which is the space that must be left between the rail ends.

(85) The unit stress developed will be

$$f = c\,t\,E$$

where

c = Coefficient of expansion;
t = Degrees change of temperature;
E = Modulus of elasticity.

For steel E is generally taken as 30,000,000. In the answer to the preceding problem, c was given as 0.0000067. Then

$$f = 0.0000067 \times 66 \times 30{,}000{,}000 = 13{,}266 \text{ lb. per sq.in.}$$

Pitot Gas and Air Meter

In order to get close readings and accurate determination the Pitot meter for measuring air or gas, shown herewith, has been brought out. It can be attached to any pipe from ½ in. up, and can be used as a portable or permanent instrument.

The meter consists of a brass tube or well, around which is wound a transparent celluloid tube, the lower end of which connects to the well; the upper end to a passage through the yoke and cock to the tube in the pipe whose opening is perpendicular to the flow of gas or air.

The well is connected by a similar opening through the yoke and cock to the pitot tube having one or more openings facing the flow of the air or gas.

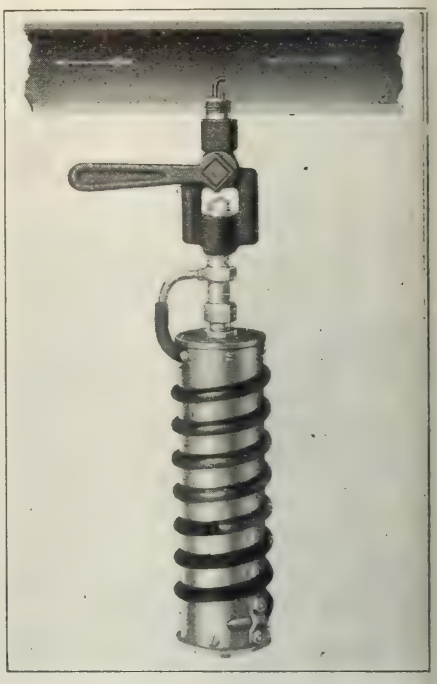
Water or oil is put into the well through an opening made for the purpose until it reaches a zero mark on the helix.

When the cock handle is vertical the space above the fluid in the well and above the fluid in the helix are both open to the atmosphere, and if the fluid stands at zero under these conditions, accurate results will be obtained.

By turning the cock handle 90 deg., as shown, the well is connected with the air or gas pipe through the pitot tube and the space in the helix is connected to the same pipe through the straight tube.

If there is no velocity to the air or gas in the pipe, the liquid in the helix will remain at zero irrespective of the pressure, as the statical head does not change with the pressure where the pitot tubes terminate.

The pressure in the pitot tube when the air or gas flows through the pipe is transmitted to the well, and the liquid in the helix rises in proportion to the velocity of the air or gas.



GAS AND AIR METER

For every inch of vertical rise the liquid travels about 10 in. around the well, and as this distance is graduated in 100 parts, the head can be read to $^1/_{100}$ in.; consequently the actual velocity of the air or gas is readily determined.

The well and helix are revolvable on the yoke so that the head of liquid in the helix can always be read, yet the connection to the pipe is not broken.

Any error caused by evaporation or condensation is readily detected by turning the cock handle for an instant to a vertical position, when the liquid in the helix should return to zero if conditions have not changed.

As the helical tube is of transparent celluloid it cannot break. The meters are tested to 100 lb. pressure. They are heavily nickelplated and are manufactured by the Sargent Steam Meter Co., 136 West Lake St., Chicago, Ill.

Test of Pipe Covering

The following results were obtained on an acceptance test made by D. J. Lewis, Jr., to determine the effectiveness with which a 10-in. pipe partially exposed to the weather at the Matteawan, N. Y., plant of the New York Rubber Co. had been insulated. The pipe, which leads from the boiler house to a distributing receiver located in the mill is 346 ft. in length, 198 ft. being exposed to the outside temperature at an average height of about 25 ft. above the ground, as shown in the reproduced photograph herewith, and the specifications required that it should be so thoroughly insulated that the condensation should not exceed 0.17 lb. per square foot of interior pipe surface per hour with a temperature of 65 deg. inside the building and zero outside. The exposed portion of the line was covered with 2½-in. Nonpareil high-pressure blocks and ½ in. of Nonpareil medium-pressure cement wrapped with canvas and waterproofed.

Inside the building 1½-in. blocks and one-half an inch of cement were used.

The condensation from this pipe and receiver was collected in a trap which discharged into a receptacle located upon a pair of scales. The receptacle was half filled with cold water into which the discharge pipe from the trap extended 8 in. When the water became so hot as to show a vapor it was discharged and replaced by cold water.

Simultaneous readings were taken every ten minutes of the pressure and temperature in the boiler house and at the receiver, and of the temperature of the inside and outside air. All instruments were calibrated and the test run for 4½ hours while the factory was in operation, extending from 10:15 a.m. to 2:45 p.m.

RESULTS

Square feet of surface exposed to outside temperature	517.364
Square feet of surface exposed to inside temperature	181.725
Total surface exposed	999.089 sq.ft.
Average steam pressure in boiler house	112 lb.
Average steam pressure at receiver in mill	111.64 lb.
Average difference in steam pressure	0.36 lb.
Maximum difference in steam pressure at any one reading	1 lb.
*Average steam temperature in boiler house	341.51 deg.
Average steam temperature at receiver in mill	341.03 deg.
Average difference in temperature	0.48 deg.
Maximum difference in steam temperature any one reading	2 deg.
Average temperature of outside air	27.66 deg.
Average temperature in mill	73.8 deg.
Total amount of condensation discharged by trap	568 lb.
Average amount of condensation discharged per hour	126 lb.
Average condensation discharged per sq.ft. of surface per hour	0.1261 lb.
Average condensation discharged per sq.ft. of surface per hour corrected for a temperature of 61 deg. F. in mill and zero outside	0.1361 lb.

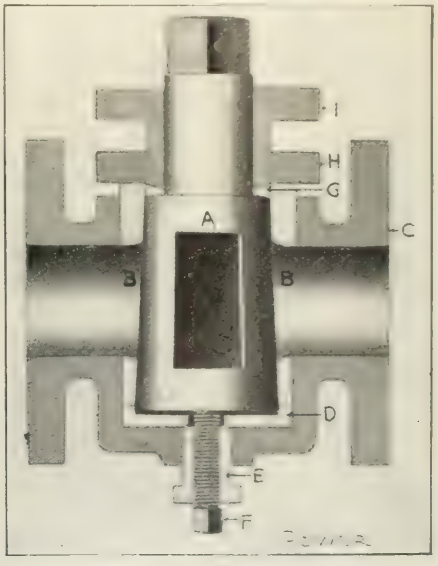
*The discrepancy between the average temperature given and that corresponding to saturated steam is doubtless due to the lag of the thermometer.

10-IN. PIPE TESTED AT MATTEAWAN PLANT OF N. Y. RUBBER CO.

A bronze bust of Joseph Henry, the distinguished scientist, has been presented to the American Institute of Electrical Engineers by Past-President Gano Dunn.

Turneasy Blow-off Cock

This plug cock consists of a body, renewable bushing and a plug *A*, which has a reversed taper with the bushing *B*. The plug and bushing are both removed from the body after taking out the glands *H* and *I*, and forcing the disk *D* up with the screw *E*. The parts are forced to a seat with the retaining gland *H*. The plug *A* is adjusted and held in position with the screw *F* to any degree of tightness. If the plug seems unnecessarily tight, the screw *F* is loosened one-quarter to one-half turn and the plug lightly driven down.



SECTION OF BLOW-OFF COCK

Wearing action takes place on the outside of the plug and on the inside of the bushing. These cocks are made interchangeable for every size, and of brass, bronze, iron or steel, as conditions require. The cock is manufactured by the Easton Tool & Machine Co., Easton, Penn.

McLeod-Mueller Turbine Check Valves

In the McLeod-Mueller nonfouling turbine check valves the disk is cast on a stem with inclined turbine wings below the disk and straight wings above, as shown in Fig. 1.

The fluid entering the inlet of the valve acts on the turbine wings and causes the stem to revolve and unseat the disk. The stem continues to revolve while the fluid is passing through, causing the straight wings to agitate the fluid above and expel all dirt. It is claimed that this valve cannot foul as the dirt is all expelled from the upper chamber by the straight wings. Another claim is that the valve cannot hang or stick because a cup bearing at the bottom, which is slotted so it cannot become packed with fluid, and a cone-shaped bearing at the top, which

conforms to the shape of the stem, act as guides to the stem and cause the disk always to seat true, and also, because the disk seats with a whirl and expels all dirt from the seat. The valve is self-grinding, since the disk rotates in seating and constantly changes its position. This rotation also has a cushioning effect and tends to prevent hammering and maintain a water-tight surface between the disk and the seat.

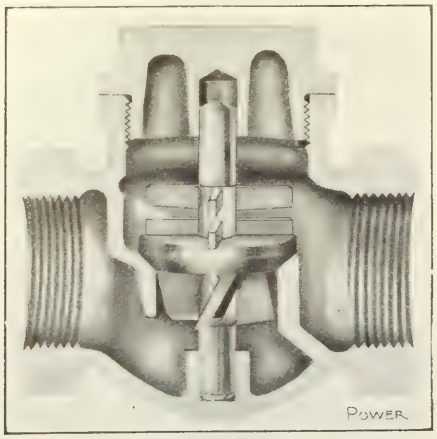


FIG. 1. CONSTRUCTION OF THE MCLEOD-MULLER TURBINE CHECK VALVE

The adjustable top, shown in Fig. 2, consists of a regulating screw around which are a gland and packing that make a water-tight joint when compressed by the stuffing nut. With this adjustable top the check valve may be used as a globe valve, and the stem can be regulated to give any size of opening desired.

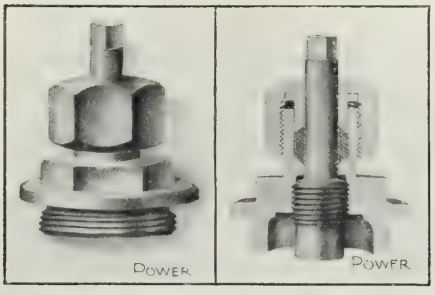


FIG. 2. DETAILS OF THE ADJUSTABLE TOP

This valve is made in vertical, horizontal and angle patterns, but the adjustable top can be used only with the two last. It can be used for water, steam, oil, or any other fluid of common density.

The H. Mueller Manufacturing Co., Decatur, Ill., holds the patents and exclusive manufacturing rights.

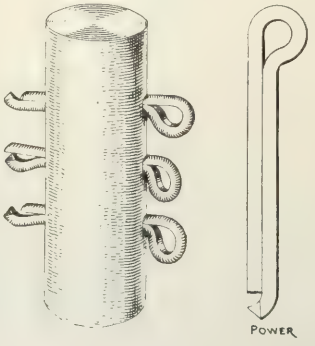
The Hess-Bright Manufacturing Co. has removed from its former location at Twenty-first St. and Fairmount Ave. to its new factory, corner of Front St. and Erie Ave., Philadelphia. The new location affords ground and floor space for much needed expansion.

New Form of Cotter Pin

To obviate the bother of spreading the tips of a cotter pin after inserting and the usual trouble incident to its removal, an improved design has been patented and placed on the market by Andrew C. Campbell, of Waterbury, Conn.

The pin, as shown, is made from a piece of half-round wire, bent so as to provide an offset eye at one end and have two limbs of unequal length with the flat sides adjacent. The tip of the longer limb is bent at an angle across the tip of the shorter limb. Hence, when the eye is given a blow, the shorter limb is driven over the longer one, and the engagement of the tips causes a separation of the limbs.

To remove the pin a screw driver or other tool having a sharp flat end, such as the tang of a file, is inserted in the eye, and by twisting it so as to expand the flattened eye, the short limb of the pin is drawn back to its original position and the pin may be withdrawn easily.



PIN INSERTED LOCKED AND READY FOR REMOVAL

The top pin in the illustration is merely inserted in the hole; the middle one has its limbs spread by a blow on the eye; and the bottom pin is about to be extracted.

The entering point being on one limb only, this pin is easier to insert in the hole than an ordinary cotter pin. Moreover, the latter must be spread on the side of the hole opposite from that in which it is inserted, while this new pin is inserted, locked and removed from the same side, making its use possible in places difficult of access.

The Rees Roturbo Pump

In 1906 the Rees Roturbo pressure chamber pump was invented and placed on the European market. The American and Canadian rights to manufacture and sell this pump have recently been obtained by the Manistee Iron Works Co., of Manistee, Mich.

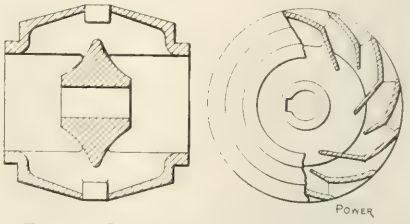


FIG. 1. SINGLE-STAGE IMPELLER IN SECTION

The characteristic feature of the pump is the self-regulation of the driving power secured by making the impeller of large capacity for storing water. From the eye of the impeller the water enters the pressure chamber, which is constructed with blades, as shown in Fig. 1, and is discharged through rearwardly directed nozzles, or guides, to the external casing. In this way the water is held together and the loss from the churning

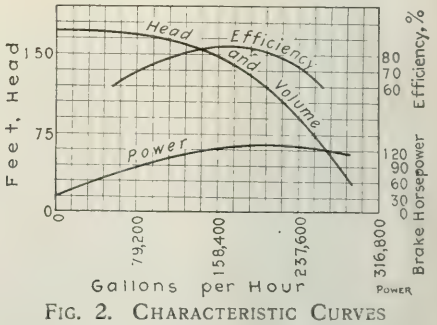


FIG. 2. CHARACTERISTIC CURVES

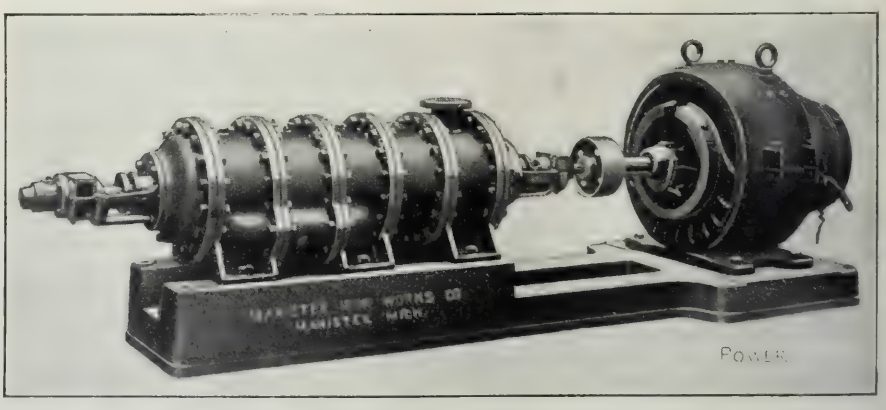


FIG. 3. LATEST TYPE OF REES BOILER FEED PUMP

action of the impeller found in pumps of early design is avoided. The reduction of the velocity of flow of the water through the pressure chamber also decreases the friction. The rotation of the impeller, together with its water, creates

a constant pressure equivalent to a certain head. A reduction in the head causes a larger volume of water to be discharged so that at a constant speed of rotation the power absorbed is perfectly constant for all heads of discharge and heights of suction possible. Thus the power required is limited and it is impossible to throw any overload, on the driving motor under these conditions. This is shown in Fig. 2. The power curve rises to a maximum at about normal head and volume. With an increase in volume the head drops off so that the power does not increase appreciably above that required for normal duty. Naturally, the efficiency is greatest at the capacity for which the pump is designed.

Fig. 3 represents the latest type of Rees Roturbo boiler-feed pump direct coupled to a continuous-current motor. The rotating parts of the pump consist of a series of pressure-chamber impellers which are mounted upon a common shaft supported upon intermediate bearings. The general appearance of these impellers is shown in Fig. 4, which gives a view of one common shaft with two impellers mounted upon it. Each of these impellers consists of two stages.

A feed pump on test is reported to have delivered 18,000 gal. per hr. against a boiler pressure of 220 lb. per sq.in. at an efficiency of 76½ per cent. Other data submitted were the characteristic curves in Fig. 2 of a comparatively low-speed pump running at 925 r.p.m. The remarkably high efficiency of 84 per cent. may be noticed. The duty of the pump is 150 ft. lift and a delivery of 158,400 gal. per hr. at the above speed, the diameter of the discharge pipe being 12 in. When this pump was tested at the works the following results were obtained:

Starting from the duty point, viz., 158,400 gal. per hr. at 150 ft. head, the power is reduced from 120 hp. to less than 40 as the discharge valve is closed. To the right of the curve when the discharge valve opened so as to reduce the head against which the pump is working, the volume with minimum pressure is increased to double that at the normal duty, showing that there is not any throttling in the pump.

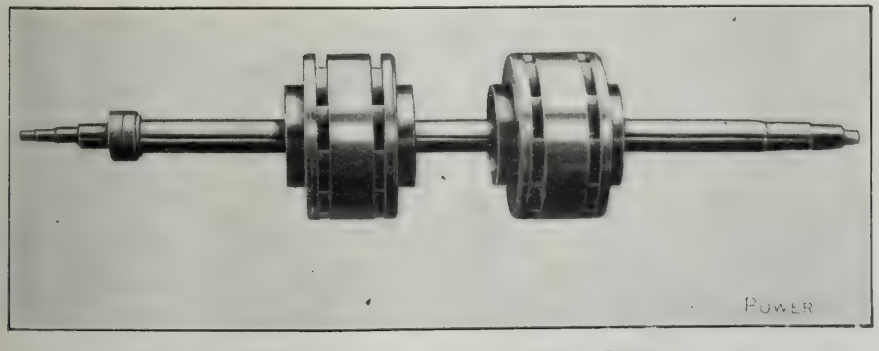


FIG. 4. TWO-STAGE IMPELLERS ON COMMON SHAFT

Test of Another Large Boiler

In view of the importance which has been attached to the tests of the 2500-hp. boilers at the Delray Station of the Detroit Edison Co., a test recently made by Prof. L. P. Breckenridge, of the Sheffield Scientific School at Yale, upon one of the 1250-hp. Bigelow-Hornsby boilers at the plant of the Hartford Electric Light Co., will be read with interest. This boiler was designed for a 2500-hp. unit some years before the Delray boilers were contemplated, but upon erection was divided, by the throwing up of a partition wall in the furnace and the separation of the steam drums, into two units of 1250 hp. each.

Five separate tests of about eight hours each were run, in one of which a capacity of 2245 hp. was maintained all day. The highest efficiency obtained was 77 per cent. and the efficiency at 180 per cent. of rating was 70 per cent. The tests were made with the boiler in service and with no preparation except an external blowing of the tubes and stopping of air leaks. The tubes had not been cleaned internally for more than a year. The fuel was taken from the ordinary supply, as purchased for use by the company, and was of only medium quality. A condensed table of the data and results is given herewith. A description of these boilers and the Taylor stokers with which they are equipped will be found in POWER, Feb. 28, 1911.

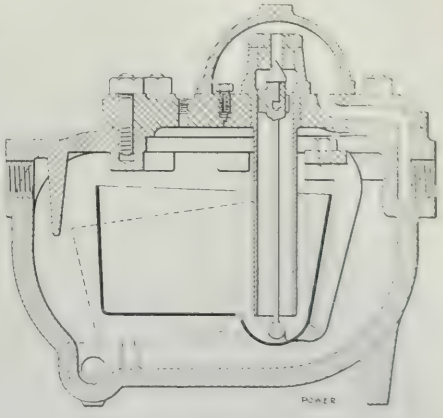
BIGELOW-HORNSBY BOILER AT HARTFORD ELECTRIC LIGHT CO.

Condensed Table of Data and Results

	Test No.	Prelim. 2	3	Final 4	5	6
Date of trial	(1912)	Feb. 6	Feb. 7	Feb. 8	Feb. 9	Feb. 10
Duration	hours	8.15	8.0	7.43	8.48	8.02
Total Quantities						
Water actually evaporated	pounds	448,048	465,527	534,349	516,369	359,285
Weight of coal fired	pounds	50,202	50,525	61,712	57,049	38,736
Weight of ash and refuse	pounds	4,485	5,228	6,039	7,265	4,373
Percentage ash and refuse in dry coal	per cent.	9.38	10.83	10.28	13.36	11.82
Moisture in coal as fired	per cent.	4.52	4.69	4.91	4.68	4.52
Pressures, Temperatures, Draft, etc.						
Steam pressure gage	lb. per sq.in.	151.8	152.3	158	153.3	152.8
Temperature of feed water entering boiler	deg. F.	211.8	212.4	213	213	213.1
Temperature of steam leaving boiler	deg. F.	450	436	431	432	433
Temperature of flue gas at uptake	deg. F.	498	526	574	535	510
Degrees of superheat in steam	deg. F.	82	69	63	64	65
Draft between damper and boiler	ins. water	0.49	0.36	0.29	0.26	0.31
Draft pressure of furnace tuyeres	ins. water	2.18	2.70	3.17	2.65	1.53
Boiler Horsepower						
Boiler horsepower developed		1,735	1,825	2,245	1,895	1,400
Percentage of rated horsepower	per cent.	139	146	180	152	112
Efficiency Results						
Water evaporated per lb. of coal as fired	pounds	8.94	9.17	8.66	9.05	9.30
Equivalent evaporation per lb. of dry coal	pounds	10.20	10.46	9.80	10.23	10.47
Equivalent evaporation per lb. of combustible	pounds	11.24	11.75	10.91	11.80	11.88
Heating value per lb. of dry coal	B.t.u.	13,630	13,787	13,625	13,280	13,280
Heating value per lb. of combustible	B.t.u.	15,250	15,180	15,100	14,870	14,870
Efficiency of boiler and furnace combined		72.7	73.7	69.8	74.8	76.5
Efficiency based on combustible		71.7	74.8	70.2	76.9	77.5

Hydromatic Steam Trap

The new steam trap gotten out by the Engineers' Appliance Co., Cleveland, Ohio, illustrated herewith, contains a bucket hinged at the lower corner. A discharge tube, in which the pole rod of the valve operates, extends from the top of the trap to a point level with the bottom of the bucket, where it is hinged by a pin. The water of condensation enters the trap through the left upper connection and, striking the baffle-plate, falls to the bottom of the trap. As soon as the water rises to the level of the bucket, it is filled and, being hinged at one end, is forced downward, which movement operates the pole rod and opens the discharge valve. The water is then forced out of the bucket, up through the discharge pipe and valve to the chamber above the valve seat and out through the discharge outlet. The bucket then assumes its original position, closing the valve. The operation is repeated with the inflow of additional water.



SECTION THROUGH TRAP

Locomotive Boiler Explosion

By M. C. St. John

A locomotive boiler explosion on the St. Louis, Brownsville & Mexico R.R., a few miles south of Bay City, Tex., killed the engineer, fireman and conductor and completely wrecked the locomotive, tender and several loaded box cars. Two views of the wreck are shown in the accompanying illustrations.

FIG. 1. EXPLODED LOCOMOTIVE BOILER

FIG. 2. LOCOMOTIVE TRUCKS BESIDE THE TRACK

The engine was running at a speed of between 25 and 30 miles an hour when the explosion occurred. The cause will probably never be known.

The boiler was thrown to one side, parallel with the track, with the front end toward the rear of the train. The trucks were derailed and after being shoved for 75 ft. by the momentum of the train were piled in a heap with the tender and six box cars. The three men in the cab were caught under this wreckage.

The cause was purely a crown-sheet failure, but whether from low water or not is not known. The crown-sheet was blown down to and below the grate line from the second row of staybolts in front on the flue sheet back to the rear end of the firebox. The flue sheet was not disturbed, or any of the tubes injured. Two rows of staybolts next to the flue sheet were intact, with the exception of a few which seemed to have been elongated. Very few staybolts were torn out of the crown-sheet, but were broken off just flush with the top of the sheet almost as smooth as if they had been sheared.

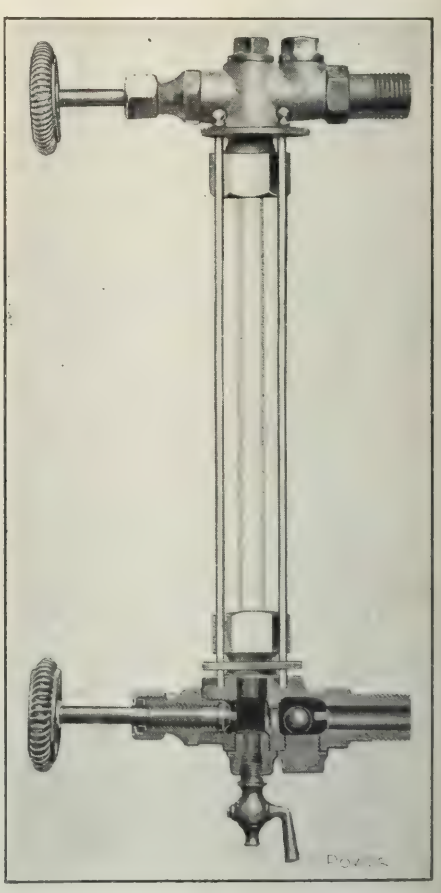
The staybolts before breaking showed indications of having stretched, in some cases, an inch or more, the diameter being reduced in section in proportion to the amount of elongation.

The boiler was an old one and had evidently seen hard service.

Watertown Automatic Safety Water Gage

This water gage possesses several features of merit aside from the ground bronze noncorrosive check balls, which, with its seats, constitute the automatic feature of the safety device.

Each ball lies in a chamber and in case the gage-glass breaks, the steam pressure acting on the upper ball and the water pressure on the lower ball forces them to their seats, automatically cutting off the escape of steam and water. When the pressure is removed the balls will fall to the bottom of the chamber. Should it be desired to shut off either top or bottom connection from the boiler by hand, the handwheels are turned, screwing the solid disks against their seats.



AUTOMATIC SAFETY WATER GLASS

The main feature of this device is its construction, which permits of accessibility to the ball chamber and to the water-glass connection, as shown in the illustration. By removing the plug from the bottom and top of the ball chamber the ball can be removed and any accumulation of sludge or scale washed out.

The pet-cock in the bottom of the water-glass connection can be removed, thus permitting of cleaning out this opening to the glass. The valves are removed by unscrewing the stuffing-box plugs which permits a straight entry to the boiler connections. The gage is manufactured by the Watertown Specialty Co., Watertown, N. Y.

SAWMILL WRECKED BY BOILER EXPLOSION

Sawmill Boiler Explosion

BY A. STALEY

On July 31, at 10:05 p.m., a boiler in the Weiner sawmill, on the outskirts of Joliet, Ill., exploded, wrecking the building and killing the night fireman, the only man on duty at the time.

This boiler was equipped with a 3-in. ball-and-lever safety valve, which, from its appearance, had not blown for a long time. The plates were $\frac{5}{16}$ in. thick and had three patches on the bottom of the shell. The boiler was 66 in. in diameter and 16 ft. long. The heads were braced with twelve ½x2-in. crowfoot braces. There were forty-four 4-in. tubes.

Employees state that the boiler was never inspected and that no insurance was carried on the boiler. Gages were never tested and the safety valve was supposed to blow at 90 lb. pressure.

It was the custom of the employees to take turns at firing the boilers at night, there being no regular fireman or engineer. The man in charge of the boiler at the time of the explosion was a new employee and had never fired a boiler before.

From all appearances it was not a case of low water, but of excessive pressure. The boiler was only supposed to be carrying between 70 and 80 lb. at the time. Pieces of the shell were scattered around the neighborhood. A second boiler was thrown about 50 ft. from its setting, resulting in a cracked girth seam on both sides of the boiler.

Trill Continuous Card Indicator

The Trill continuous card indicator is so made that a roll of paper is placed on a drum, the indicator started and a card given for every stroke of the engine without any further attention on the part of the engineer. It is attached to the engine in the ordinary way.

A roll of paper is placed on the smaller drum *D* and the free end threaded around the larger drum *E* on which the diagram is traced. To avoid the necessity of tracing a separate atmospheric line for every diagram, the second pencil *B* is provided, and set at the atmospheric line given by the first pencil, thereafter it traces a continuous atmosphere line the length of the card.

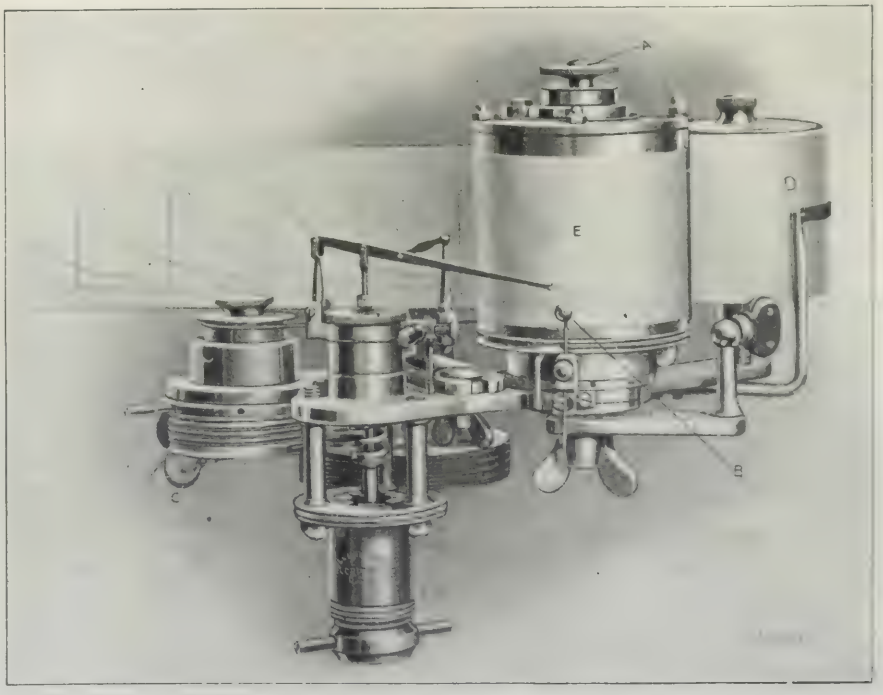
The cord is pulled out to its full length and held by the detent *C*. The cord hook is then attached to some part of the crosshead from which the driving motion is obtained. Releasing the detent *C* starts the indicator. For a continuous card, the nut *A* is screwed down, which sets the paper translating device into action, and a series of diagrams will be traced, as shown in the illustration. This strip of paper can be torn off at any place and the diagrams examined.

A spring and catch hold the tracing pencil against the drum when taking continuous diagrams assuring a uniform pressure on the pencil point.

If a continuous diagram is not desired, the nut *A* is loosened while one diagram is being traced and then tightened while the engine makes two revolutions, which brings an entire new sheet of paper on the tracing cylinder and another complete diagram can be traced by loosening nut *A*. Separate diagrams can thus be taken whenever desired.

The indicator has a spring entirely outside of the cylinder, removed from the high temperature steam, or the cutting action due to wire drawing.

The position of the atmospheric line



TRILL CONTINUOUS CARD INDICATOR

on the drum can be adjusted by turning the small nut on the pencil motion on top of the cylinder.

Each indicator has two cylinders, one having an effective area of ½ sq.in. for low-pressure work, the other an effective area of ¼ sq.in. for high-pressure gas-engine work.

This indicator is manufactured by the Trill Indicator Co., Corry, Penn.

Over the Spillway

An engineer told us the other day that he put Shims under the knock-off block. Anybody here seen Shims?

Why do some correspondents neglect to give their names when writing? We are asked: "Why do my feed pipes clog?" Is this man an engineer or a victim of the "quick-lunch" habit?

"Say, Doran, our frind Gawge Perkins, of N'York, sez that electricity have given wings to ther human moind, an' it's ther moind an' not ther body as does ther business."

"Take it from me, Horan, ther moind rs an unsafe thing to hitch yer overall straps on if ye intind to presarve ther peace an' kape yer body in ther background."

Our local police are again in hot water for raiding a pipe layout in "Lefty" Heater's place. Magistrate Feedwater says that for once the "system" is all right, but the cops are too zealous.

An *American Machinist* contributor says he pays for his vacation every year by writing for the technical papers. This is offered as a good "home-made kink" to POWER readers.

Newspaper notoriety has been achieved by an industrial-betterment worker on his return from Europe, where he was a delegate to the recent congress of accident prevention. He states that his son has to study in Germany "to learn things not to be had in America—to study, to be respectful, to be obedient." *Nicht wahr!* In one of Uncle Sam's power plants he would sure learn to be obedient and respectful, no matter how far in the rear his "learning things" straggled.

Berton Braley, story-writer and poet, was once a coal passer.

Nikola Tesla submits a proposition to the New York Board of Education whereby the dull pupil can be made bright and the bright pupil a mental "white way" by day and by night. It is the electric "bath." He would pass through the school-room air a high-frequency current of "millions of volts." The room would thus be "saturated with infinitesimal waves vibrating at great frequency." Nikola seems to have appropriated Jonah's trite remark to the whale: "You can't keep a good man down," or out of the public eye.

Emery Wheel, over to Turner's mill, lost his bearings one day last week and dropped on the floor. The way Em's been cutting up lately made his shopmates cal'late he'd go to pieces sooner or later.

The first arc-control system was installed by one Noah, a marine engineer, in the very earliest stages of its development. It operated continuously for 40 days and nights. At that time, Noah was the greatest living authority on marine propulsion and had the exclusive use of all the water-rights.

Hi Stack has stopped smoking since the Chicago smoke inspectors have got so all-fired particular. Hi asseverates that the one course to travel by is charted by Ringelmann.

An Illinois lighting company voluntarily raised its employees' pay two cents an hour, and the press report naïvely adds that the men were "surprised." Sort of a two-cent stamp of approval, eh what?

"Mirth is God's medicine," said Dr. Holmes. "Everybody ought to be bathing in it." Now turn on the water and let your neighbor hear you "splash." Everybody ought to be doing it.

Con Crete, of C. Ment, Stone & Sand, has floored the whole engine room. Says he's set against folks walking all over him.

Congress has refused permission to dam Duck River, Tennessee. In a manner of speaking, it ducked the bill. We have yet to learn that the irate petitioners have dammed Congress.

NEW PUBLICATIONS

ELEMENTARY MECHANICAL REFRIGERATION. By Fred E. Matthews. McGraw-Hill Book Co., New York City. Cloth; 172 pages, 6x9 in.; 43 illustrations; 24 tables. Price, $2 net.

In this book, addressed to "the busy man at the desk, drawing-board and throttle," the author presents in simple form the principles of refrigeration, approaching the subject from the heating point of view so that the reader by stepping up from a subject more widely known may readily grasp the meaning.

Beginning with an elementary discussion on cold and its production and a few pages on the development of mechanical refrigeration, the author takes up in logical order: commercial systems of refrigeration, including direct expansion, brine circulation, ice-freezing, etc.; the compression and absorption systems; ice-making systems, installation and operation, working pressures, cleaning, determining capacity of refrigerating machines and cold-storage duty.

With the exception of a passing reference to the dense-air process, the book has been confined exclusively to the use of ammonia. Here it is as complete as might well be expected from the limited number of pages. The busy practical man and beginner should find the book a valuable stepping-stone into the subject of refrigeration and obtain some useful hints from the chapters devoted to operation and cleaning.

REFRIGERATION, COLD STORAGE AND ICE MAKING. By A. J. Wallis-Tayler. D. Van Nostrand Co., New York City. Cloth; 655 pages, 6x9 in.; 414 illustrations; tables. Price, $4.50 net.

This book, which was closed by the author in December, 1911, is in its third edition, and, although a number of the plates have been replaced by blocks of more modern machines, a few new cuts added and the text describing the machinery revised, it is incomplete and much of its contents out of date, so far as American practice is concerned.

The introduction gives statistics of the trade in fresh provisions in England from 1891 to 1910. Chapter II deals with the theory of heat, the laws of gases and a classification of refrigerating methods into the five principal heads; liquefaction, vacuum, compression, absorption and cold air. The two systems first mentioned are treated briefly in Chapters III and IV, and the next three chapters, containing 118 pages, are devoted to the compression system.

A good description of the cycle of operation is given, and the balance of Chapter V describes at length and illustrates the ether machines of Perkins, Twining, Harrison and Tellier, Pictet's, Twining, dioxide and Windhausen's carbon-dioxide machines, all of which are out of date, although perhaps of historic interest. In the description of the ammonia compressors in Chapter VI, the old machines and some made by companies out of business for a number of years, have not been eliminated.

Condensers and water coolers are given 22 pages, and Chapter IX deals with the absorption system. Only the older types of machine are mentioned, and nothing is said of the new exhaust steam and combination methods. In the chapter on the cold-air system, references dating back to 1881 indicate that the latest material on the subject has not been incorporated. In Chapter XI, valves and fittings are given some attention, and the subject of the next four chapters, taking 111 pages, is "Refrigeration and Cold Storage." In this space ammonia and brine circulation, ventilation, insulation,

piping, and conveying machinery are covered, and here, too, most of the references date back 20 to 30 years.

"Marine Refrigeration" in Chapter XVI, and "Refrigeration in Dairies" in the chapter following, give strictly English practice, while Chapter XVIII briefly outlines industrial applications in chocolate factories, breweries, etc., and constructional application, such as tunneling and sinking shafts.

Chapter XIX takes up the various methods of ice making in Europe, and as foreign methods are not employed in this country, the descriptive matter can only be of general interest. This does not hold, however, in the chapter following on management and testing of refrigerating machinery, as many good pointers of a practical nature are given and also some space to the interpretation of indicator diagrams. A chapter on cost of working, in which old data of very little value at the present are given, and 14 pages on liquid air complete the book.

Aside from the fact that the book is out of date in many respects, it contains a great deal of valuable matter, and to those interested in foreign practice it is well worth having.

Big Dam for Seattle

A big dam on the Jordan River on Vancouver Island, 40 miles northwest of Victoria, will be built for the British Columbia Electric Ry. Co. The dam will cost about half a million dollars, will be 800 ft. long and 115 ft. deep and be built on the Ambursen type of reinforced-concrete structure.

Men are already at work on the construction and it is expected that the structure will be completed in eight months.

SOCIETY NOTE

The Sixth International Congress for Testing Materials was formally opened on Sept. 2 in the Engineering Societies' Bldg., 29 West Thirty-ninth St., by Robert W. Hunt, of Chicago, president of the American Society for Testing Materials. The proceedings and brief abstracts of discussions will appear in an early issue.

John Fritz's 90th Birthday

John Fritz, who has been ill at his home in South Bethlehem, Penn., for some weeks, passed his ninetieth birthday anniversary recently. He was born Aug. 21, 1822. The past year is marked by the publication of Mr. Fritz's autobiography.

Mr. Fritz said: "The publication of my autobiography before my death is owing to the fact that, against my wishes, these good friends would not wait for it but insisted on having it now. And so I have jotted down the record of my life, and it is given to you as I wrote it. You must not expect fine language nor eloquent periods, but only the honest record of the hardworking life of one who loves his country and his fellowmen, and who has tried to serve both."

OBITUARY

Harris Boardman

The recent death of Harris Boardman, superintendent of the Edison Electric Illuminating Co., Lancaster, Penn., occurred at Atlantic City, N. J.

Luther E. Mansfield

Luther E. Mansfield died at his home in Brooklyn on Aug. 27, aged 84 years. He was a retired manufacturer of elevators and for many years connected with the Brooklyn Electric Elevator Co. of Brooklyn. Mr. Mansfield's wife, Sarah Ann Mills, died six years ago, and he is survived by a son, Earl; a daughter, Florence; two grandchildren, and one great-grandchild.

Herbert W. Spear

Lieut. Herbert W. Spear, chief engineer of the United States revenue cutter "Seminole," died recently at Edgartown, Mass., where he had been on sick leave for some time, owing to heart trouble. He was a native of Quincy, Mass.

Lieut. Spear had several thrilling experiences while in the revenue-cutter service, one of the most notable being when he had to swim ashore from the "Gallatin" in a blinding snowstorm when that vessel was wrecked off the Massachusetts shore in the early nineties.

Warren H. Girvin

On Aug. 16, Warren H. Girvin, formerly manager of the Syracuse (N. Y.) Lighting Co., died at his home in Syracuse, N. Y. He was born in Oneida, N. Y., in 1865 and went to Syracuse about 40 years ago. Starting as a telegraph operator, he became interested in the Bell Telephone Co. and was made manager of the local office, which position he held until 1890, after which he assumed the management of the Syracuse Lighting Co. In 1900 he returned to the telephone business, but ill health compelled him to retire in 1905. A widow, son and daughter survive Mr. Girvin.

Thomas H. Brady

Thomas H. Brady, owner of the Brady Electric Manufacturing Co., New Britain, Conn., died recently at his summer home, Westbrook, Conn. Mr. Brady was one of the earliest pioneers in the manufacture of outside electrical construction material, his first contribution being the well known Brady mast-arm, patented in 1897. His acquaintance in the electric field, especially among those who may be called the old-timers, was very extensive, and his social spirit and interesting stories of the early days of the electric business made him welcome at any gathering. Mr. Brady leaves a wife and ten children.

PERSONAL

J. W. Hungate is the new chief engineer of the power station of the Spokane & Inland Empire R.R., Spokane, Wash.

E. P. Shanks is the chief engineer of the power station of the People's Traction Co., Galesburg, Ill., in place of J. H. Hibbard.

Thomas E. Bryant has succeeded J. T. Maher as chief engineer of the power station of the Richmond & Henrico Ry., Richmond, Va

E. Brown is the engineer of the power station of the Chicago & Milwaukee Electric R.R., Highwood, Ill. J. F. Shreffler formerly held this position.

N. A. Miner has succeeded J. C. Whitney as chief engineer of the power station of the Springfield & Northeastern Traction Co., Riverton, Ill., a part of the Illinois Traction System.

R. L. Cardiff, formerly chief electrician of the Coast Counties Light & Power Co., Santa Cruz, Calif., is now in charge of the plant, having succeeded C. W. Coleman as general manager.

R. G. Black. formerly superintendent and chief electrician of the Toronto Electric Light Co., Toronto, Ont., who sailed for Europe on Nov. 15, to consult specialists, has returned to Toronto again somewhat improved in health.

L. W. Hench, for the past three years power engineer of the Youngstown Consolidated Gas & Electric Co., has resigned and will become the Youngstown representative of the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Penn.

Robert A. Foresman has been appointed superintendent of motive power of the Des Moines (Iowa) City Ry. For the past six years he was chief engineer of the power plant of the St. Joseph Ry., Light, Heat & Power Co., St. Joseph, Mo. He will be succeeded in St. Joseph by John Elftman, who was assistant engineer of the power station.

Jules de Fischer, chief electrical engineer of the Budapest Local Ry. Co., and Marcell Jellinek, electrical engineer, Budapest, are making a six weeks' visit to the United States, inspecting the suburban and interurban electric-railway systems in the Eastern and Central States. The Budapest Local Ry. is a large suburban and interurban railway system, serving the city of Budapest and neighboring cities and is equipped with the 1200-volt system.

Moments with the Ad. Editor

Is there anything advertised in POWER with which you are not familiar?

Did you ever think of that before?

If so, what did you do?

Were you content to let it be, or were you ready to investigate?

Some of you engineers don't lick enough stamps.

You are backward about sending for samples, booklets and other literature which would make you the wiser, in many cases, and which these advertisers are only too glad to send you.

You've gained most of your knowledge through the hard school of experience, and you've earned it.

But there are a whole lot of little things you can get from hints and data offered you by the advertisers in this paper.

Many large factories are being planned and built, or enlarged, every year for the purpose of manufacturing some *new* specialty or some *new* equipment for *your* very plant.

Then, don't you realize that these manufacturers have to offer you something better and something different or new to get you interested?

The question is—Are *you* keeping in touch with these *new* things?

Are there any new "kinks" offered you that you are not "hep" to?

Our advertisers want you to have all the information available about their particular machines or products, whether you buy now, or a year from now; it matters not when.

It's helpful to them—and to you.

The writer was in a power plant the other day talking to an engineer who has spent 21 years at his profession.

He said he had been a regular subscriber to "Power" for 16 years. Asked if he read the advertising section, his answer was decidedly in the affirmative.

On his desk, (one of his own make, by the way) lay a catalog of a concern that makes a well known lubricator.

On noting that he was not using this lubricator in his plant, he was asked: "How came you to write for this catalog?"

His answer was characteristic of every "result getting" engineer. He wrote for this catalog because the advertiser offered to show him a way to better lubrication, and a way to save considerable money on his cylinder oil. He wanted to know the *how* and the *why*, and he wrote for the particulars.

We know of several instances where engineers have answered advertisements out of as much curiosity as of interest, and they got information and data that was helpful and unexpected. And the advertiser got the order.

Why not take an inventory of yourself today? Run through the advertising pages of "Power" and see if any of these advertisers "has anything" on you.

Are they offering something or offering to *do* something to help your plant conditions that you are inclined to doubt, off-hand, without investigating?

There are local conditions in your plant that do not exist in any other plant, perhaps; *so*, when any of these advertisers spends his good money to tell you how a certain condition can be overcome, why not find out *what* he has got?

Remember this—"Power" is your paper. It's edited for you. Every advertisement you see in "Power" is written direct *to you* and *for you*, by advertisers who recognize the buying influence you have, and that your say can make or break 75 to 90% of the sales that are made in the power plant field.

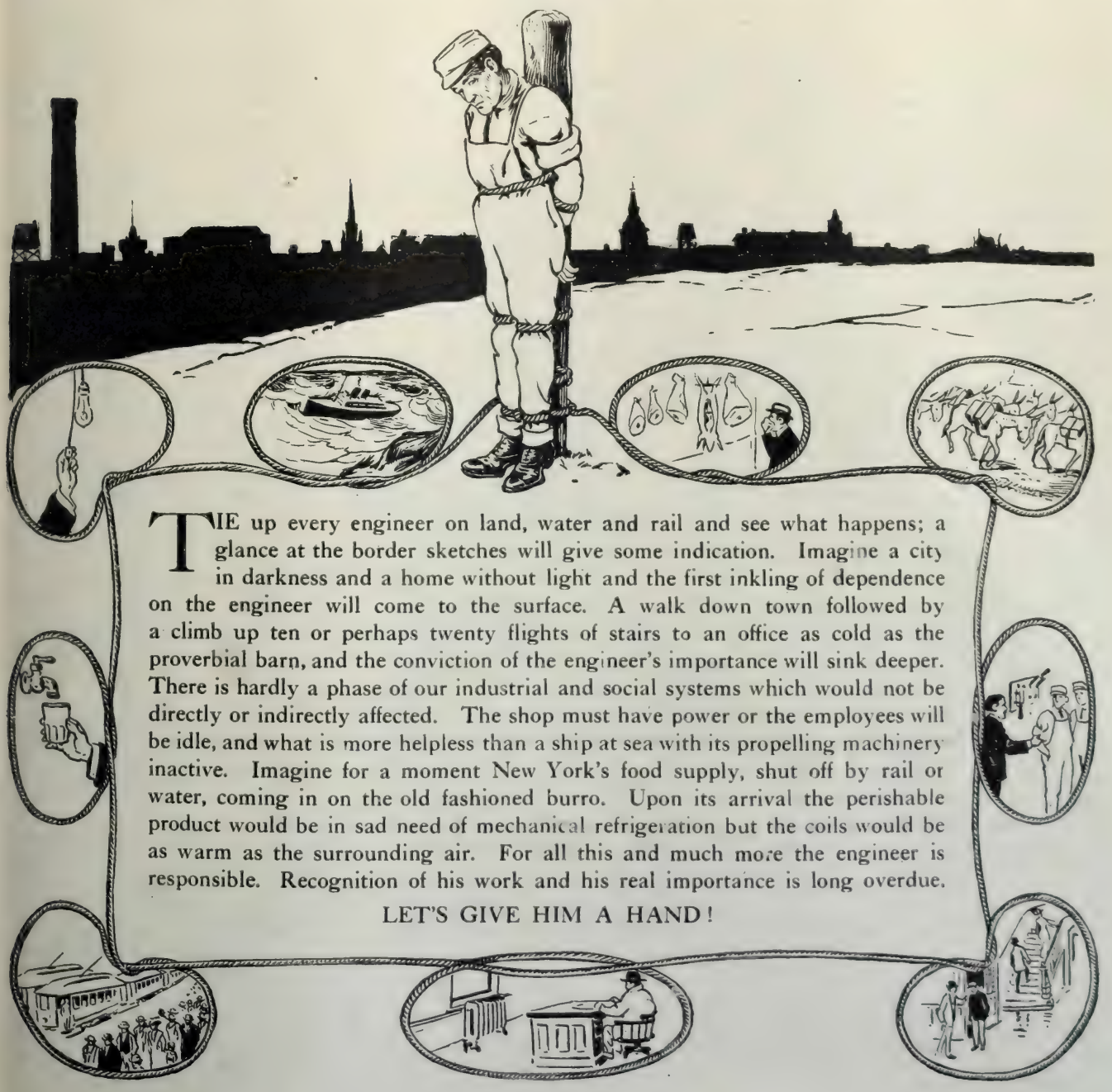
Don't admit that there are helpful things advertised in "Power" today with which you are not familiar.

Get *all* you can out of "Power" advertising *now*.

POWER

Vol. 36 NEW YORK, SEPTEMBER 17, 1912 No. 12

What Would We Do Without Him?



TIE up every engineer on land, water and rail and see what happens; a glance at the border sketches will give some indication. Imagine a city in darkness and a home without light and the first inkling of dependence on the engineer will come to the surface. A walk down town followed by a climb up ten or perhaps twenty flights of stairs to an office as cold as the proverbial barn, and the conviction of the engineer's importance will sink deeper. There is hardly a phase of our industrial and social systems which would not be directly or indirectly affected. The shop must have power or the employees will be idle, and what is more helpless than a ship at sea with its propelling machinery inactive. Imagine for a moment New York's food supply, shut off by rail or water, coming in on the old fashioned burro. Upon its arrival the perishable product would be in sad need of mechanical refrigeration but the coils would be as warm as the surrounding air. For all this and much more the engineer is responsible. Recognition of his work and his real importance is long overdue.

LET'S GIVE HIM A HAND!

The Eyermann Steam Turbine

By Wilhelm H. Eyermann*

A description by the inventor of a radial-flow steam turbine which has been built in capacities up to 2400 kw.

The constructional difficulties incident to the type are pointed out and the methods of overcoming them described.

*Translated in abstract from "Kraftmaschinenbau," by E. P. Buffet.

The fact that, in spite of some marked advantages in principle for the radial type of turbine over the axial type, the axial has so far outstripped its rival may be ascribed to constructive difficulties in the radial design and to a belief that it is impossible to get the necessary number of rows of blades on one or two wheels.

Naturally, the rotor and guide blades of a radial turbine must be dismounted axially. This is readily possible with one or two wheels, but difficult with more. It is, therefore, a vital condition with the radial turbine that the number of blade rows be comparatively small. This is possible when the pressure drop in the first stage is made quite large, say from 12 to 2 atmospheres absolute pressure, the arising velocity-energy being transformed in one or two rows of rotor blades and the remainder of the head utilized in a small number of pairs of blade rings working upon the impulse system.

In the year 1904, when the author worked out his turbine system, there were customary for impulse turbines 15 to 30 and for reaction turbines of high speed 50 to 120 pressure stages. The consideration that the lost work of the steam in leakage through the clearance between the blade rims must constitute a large portion of the total losses induced the author to make an experiment with a few stages and comparatively high steam velocities. Such a turbine of 250 hp. gave a Rankine efficiency of 54.4 per cent., a good result considering the date and the experimental construction.

On the ground of what was then learned, there was built a 1200-hp. turbine which in 1909 was tested in the mechanical laboratory of the Charlottenburg Technical High School. A section of this turbine is given in Fig. 1. The operation is as follows: After the steam has successively passed the stop valve, quick-closing valve and governor valve, it enters through a side connecting branch into the cast-steel ring *a*, expands in the nozzle *b* to about 1.5 atmospheres absolute, imparts its velocity to two rotor-blade rings and then expands in five further pairs of guide and rotor-blade rims to the condenser pressure. The entire transformation of energy takes place almost in one plane in a very short and simple way. The rotor is carried upon the end of the shaft without an outer bearing rendering a stuffing-box unnecessary.

The design of the rotor presents two main difficulties: first, through the bending forces due to one-sided admission, which tends to cause unsymmetrical steam pressure, and, second, an eccentric pull by the centrifugal force of the blades. It would not be impossible to build the rotor strong enough to keep its flexure within limits; but the advantage of lightness would thus be lost. In the accompanying design the axial pressure of the live steam is balanced by opposing on the back side of the wheel a steam pressure of suitable magnitude. For this purpose, the back side of the wheel opposite to the blade rims is provided with labyrinth grooves which hold the steam in the chamber *c* from passing into the chamber *d*. The steam pressure in the chamber *c* is maintained at the proper degree by the passage of the requisite quantity of steam from the chamber *e* into the chamber *c* through the passage *f* between the wheel and the body of the stuffing-box and through holes in the wheel disk. If the pressure on the labyrinth side is too low, the disk moves to the left by a small fraction of a millimeter and opens the crack *f* wider, letting more steam pass through it into *c*, whereby the pressure rises and the shifting movement comes to rest at the proper distance to maintain a balanced pressure. This displacement necessary for pressure regulation amounts, within the widest practical limits of pressure and temperature, to hardly 0.004 in. That this exceedingly small axial movement suffices is explained by the fact that the expansion of the steam in the blading, in the throttling passage and in the labyrinth, proceeds according to almost the same law and, therefore, the pressure and volume of the steam in all parts of the turbine vary in nearly proportional amounts. This arrangement insures that, even when the turbine is under a light load, the correct axial adjustment takes place with great positiveness, but if very little or no steam be supplied to the turbine, and therefore no considerable axial pressure be exerted, it naturally does not work. The disk must therefore be insured against striking in this case. This is effected by a collar *g* which is placed in the bearing with an axial play of about 0.04 in. In normal service, however, these guide surfaces do not touch.

The eccentric centrifugal force of the blades is balanced by sector-shaped counterweights placed on the back side of the disk. The outer surface of the

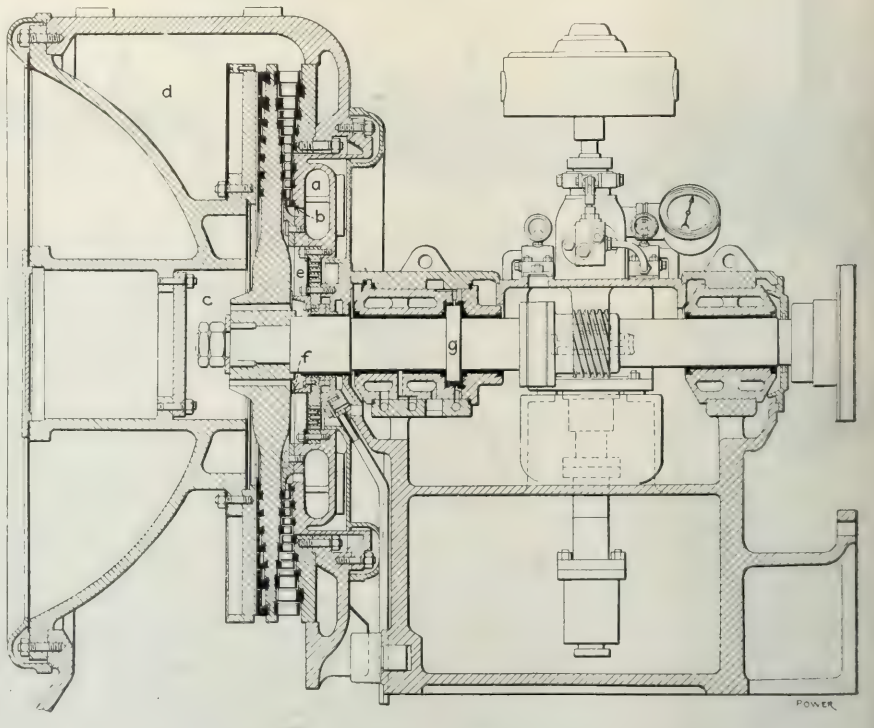


FIG. 1. EYERMANN 1200-HP. TURBINE TESTED IN 1909

counterweight carries the aforementioned labyrinth grooves whose form is shown in Fig. 2. This labyrinth form has been worked out experimentally so as to have a strong throttling effect, which may be explained by the fact that in the space between two clearances is produced a regular vortex which causes a great con-

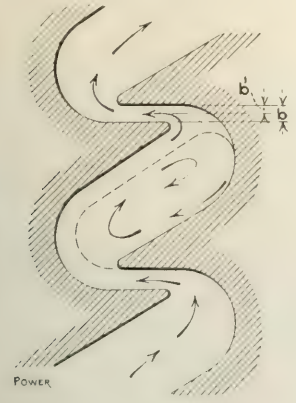


FIG. 2. LABYRINTH PACKING

traction of the area available for steam flow, as the drawing shows. The cross-section b of the passage is made as narrow as permissible with respect to precision of manufacture and unavoidable changes of shape in order to keep the steam quantity down as small as possible. This smallest dimension amounts, according to the conditions, to from 0.012 to 0.08 in. The rounded shape of the vortex chamber causes the whirling steam

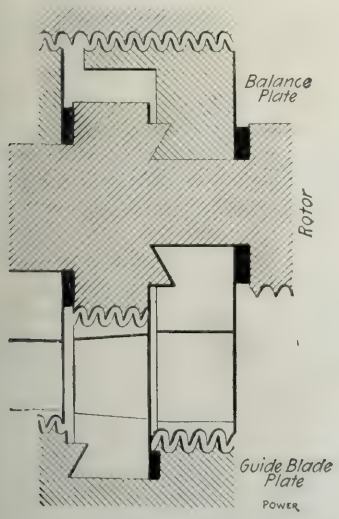


FIG. 3. BLADING

current to sweep by the point of outlet b with considerable velocity; the outflowing steam splits off from this vortex and, by reason of its momentum, presses against one wall of the outflow cross-section, so that only the width b is available for the flow while the remainder of b is filled out by an eddy. Only the width b' is to be considered as the effective cross-section of the orifice, while in actual construction the space b may be made about 50 per cent. wider than b'.

The blades are milled from the solid, nickel steel or special bronze being used according to the service. They are fastened by setting the half dovetail-shaped blade foot into a corresponding groove of the disk and securing by calking as shown in Fig. 3. The shrouding is riveted on, either in one piece with the blade, or in sectors. For the low-pressure stages it is provided with labyrinth grooves like those of the counterweight. These grooves are turned in the already bladed wheel-disk or guide-blade plate, and the fastening of the blade has thus to stand a sharp test of its security. The shaft rests in two bearings between which are placed the worm for driving the governor and oil-pump shaft and the safety governor.

The regulation is controlled indirectly by means of an oil servomotor. Fig. 1 shows, above the worm, the control valve operated by the governor for the adjustment of the regulating valve. This valve is, on normal turbines, a double-seated one and works by throttling. At the lower end of the governor shaft is a screw pump for furnishing oil under pressure to the bearing and the regulating mechanism. The governor shaft, together with its bearing and the pump, can, after loosening the oil connections, be easily taken out.

The casing is in one piece and is so placed upon the frame that it can freely expand both radially and axially, while remaining true with respect to the plane of the wheel center and to the axis. The dismounting is performed axially. The cover and wheel can be taken off in ten minutes by means of a special tool, which leaves all the inner parts of the turbine accessible for inspection.

In order to adapt the foregoing model and elements of construction to low-pressure, back-pressure and exhaust-steam turbines, the high- and low-pressure stages are divided between two separate wheels. This gives a somewhat better efficiency of the high-pressure wheel, since it may be made larger in diameter. The first turbine built in this way, running at 3000 r.p.m. with a load of 1200 kw., 12.35 atmospheres absolute initial pressure and 331 deg. C. (628 deg. F.) before the stop valve and 0.025 atmos-

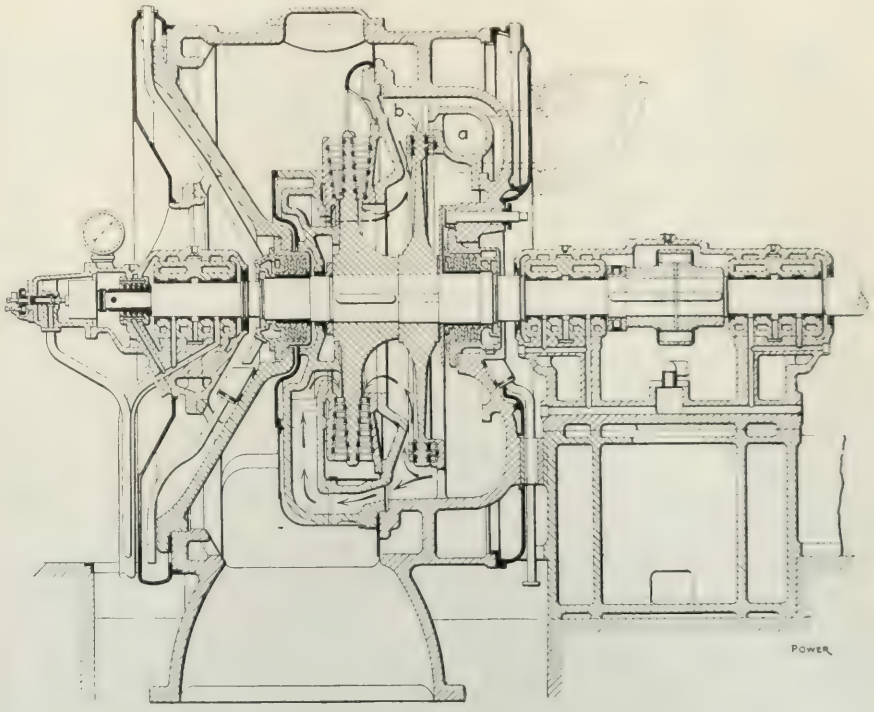


FIG. 4. COMBINED VELOCITY- AND PRESSURE-STAGE TURBINE 2400 H.P.

phere condenser pressure, ran upon 6.3 kg. (13.861 lb.) of steam per kilowatt-hour.

The turbines hitherto built have been constructed, according to existing ideas, with the smallest possible number of stages, that is, approximating as closely as possible to the critical steam velocity. Experiments in the mechanical laboratory of the Technical High School at Charlottenburg have indicated, however, that this steam velocity may advantageously be exceeded, so that it may be possible still further to decrease the number of stages. Consequently, for a speed of 3000 or of 1500 r.p.m., one may get as the full limit of power under normal conditions, with steam admitted on one side, 1700 or 4500 hp., and with admission on both sides, 3500 or 9000 hp. The dimensions of machines so constructed are shown in Figs. 5 and 6.

For these turbines, as well as for that in Fig. 4, a new method of regulation is employed. The governor is seated on the free end of the shaft, and offers to the passage of the oil a different section of orifice for each speed. This oil operates a servomotor which controls the throttle regulating valve. The oil pump is here driven, not from the turbine shaft but by a special electric or steam drive, which method leads to a considerable simplification and cheapening of the construction.

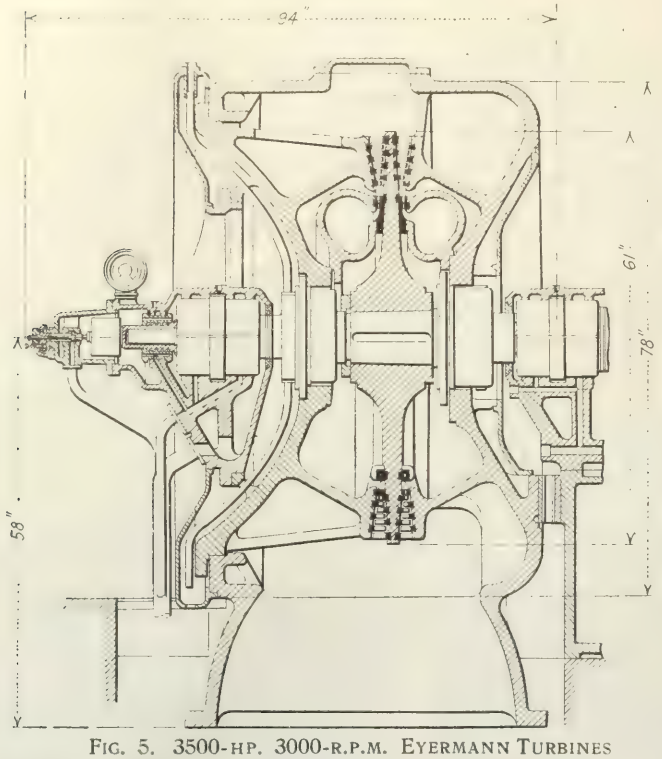


FIG. 5. 3500-HP. 3000-R.P.M. EYERMANN TURBINES

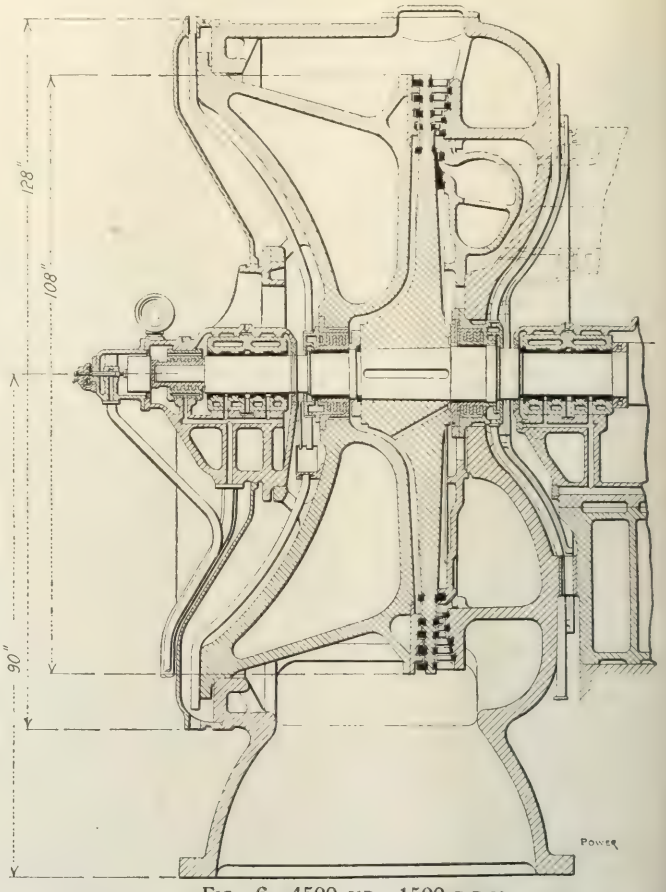


FIG. 6. 4500-HP. 1500-R.P.M.

The New Allis-Chalmers Engine

Description of the construction of a new Corliss engine built by the Allis-Chalmers Co., and table of tests. Among the interesting details are the valves, especially adapted for high temperatures, the piston construction and the governor.

The Allis-Chalmers Co., Milwaukee, Wis., has recently brought out a new Corliss engine, which has many attractive features. It is not an old design fixed up, but an entirely new machine. The frame is in one piece of massive design, the outside being a smooth casting, Fig. 1, which is easy to keep clean. A lip cast around the base of the frame catches any oil that may get outside, but very little can do so on account of the arrangement of oil guards, to which particular attention has been given.

The crank oil guard entirely incloses the crank and has a rectangular opening in front with two large doors, as shown in Fig. 2, the panels of which are wire screens, permitting air circulation to keep the internal parts cool and prevent oil being thrown out. The doors are large enough so that the crank brasses can be keyed up without taking off anything, and by removing four small bolts on the bar to which the doors are hinged, the doors can be removed, and the opening is large enough to permit the brasses to be taken off from the pins without taking off the oil guard.

The frame has unusually heavy jaws to carry the shells, which inclose the main bearing. The bottom shell is spherical, resting in a recess of the same form in the bedplate, which design allows the shells to align themselves with the shaft under all conditions. To remove the bottom shell requires only lifting the shaft about $\frac{1}{32}$ in. when the bottom shell can be rolled out. The quarter boxes are both adjustable with a wedge running the entire length, and each wedge has two draw bolts and two setscrews, so that the wedge can be positively held in any position desired.

The cylinder, shown in section in Fig.

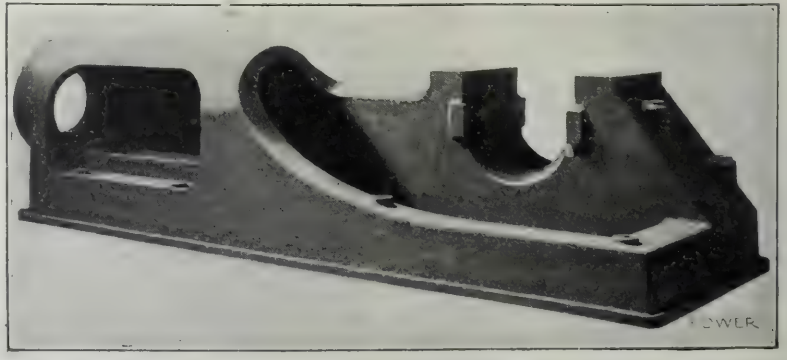


FIG. 1. FRAME OF THE NEW ALLIS-CHALMERS CORLISS ENGINE

3, has been designed with the smallest clearance consistent with getting the steam into and out of the cylinder, with the least possible losses. The steam valves are of the three-bar type, which design allows the valve to retain its form and remain tight, with very high temperatures due to high steam pressure and superheat. The center of the exhaust-

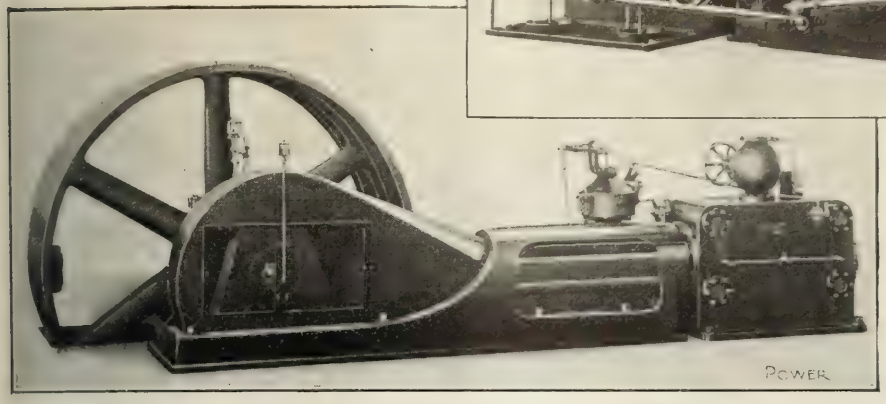
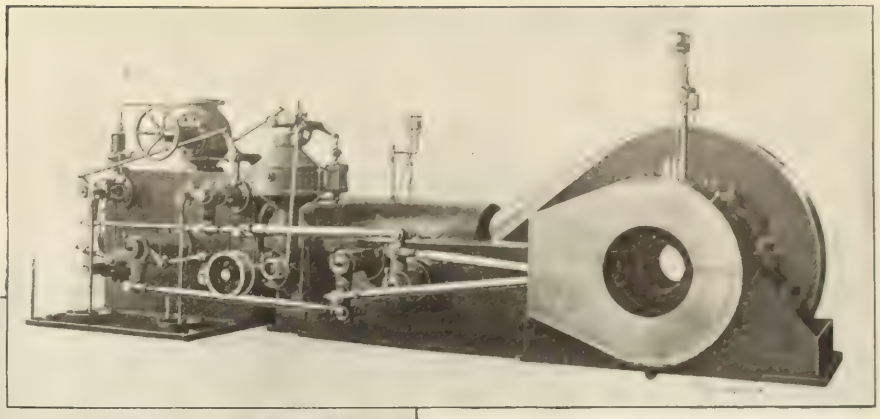


FIG. 2. VIEWS OF OPPOSITE SIDES OF THE NEW ENGINE

valve chamber is on a line with the bottom of the cylinder, and is so placed that the piston does not travel into the valve chamber. This design reduces the clearance as much as it can be with certainty that the piston will never under any condition strike the valve. All valves are multi-ported with a short travel.

The piston is made wide, giving a broad wearing surface and has two rings, which travel over the counterbore, so that no shoulders will be worn at the end of the cylinder. Fig. 5 is a section of the piston showing particularly the method adopted to prevent the follower nut from falling into the cylinder if a stud bolt breaks. The safety collar is larger than the opening in the follower for the head of the bolt, so that if the stud breaks off in the threads the broken-off portion with the nut cannot fall out.

The piston rod is forged with a large collar, and the end of the rod has a nut for securing it to the piston. The other end of the rod screws into the crosshead and is held with a locknut. The cross-



head shoes are both adjustable and interchangeable.

The valve-gears, Fig. 4, all have separate eccentrics for actuating the steam and exhaust valves, and the steam valves are under the control of the governor from 0 to 75 per cent. cutoff. The steam eccentrics are connected through rocker-arms and reach rods direct to the steam arms which hang from the bonnets. A disconnecting device is provided in the

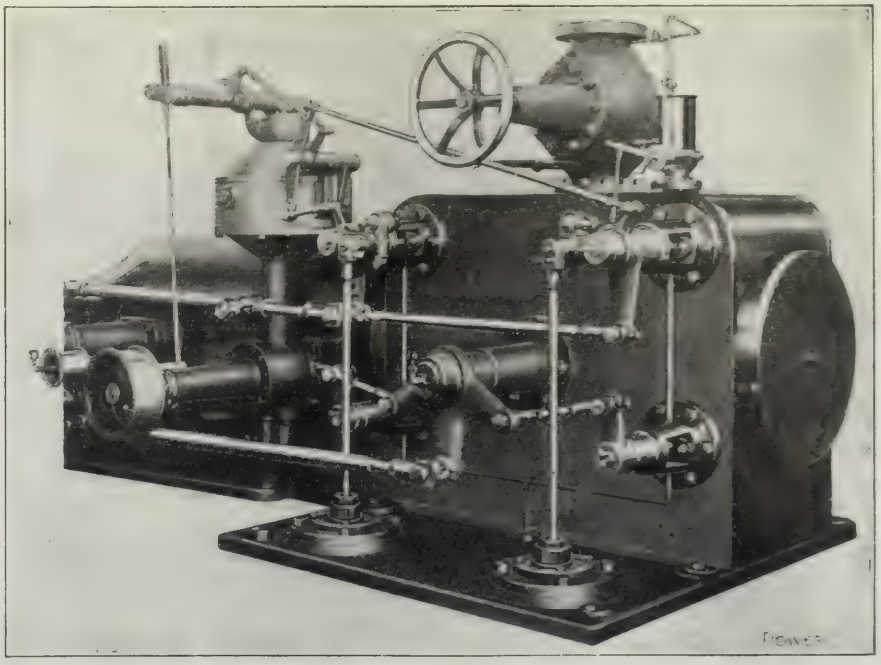


FIG. 4. DETAIL OF THE VALVE-GEAR

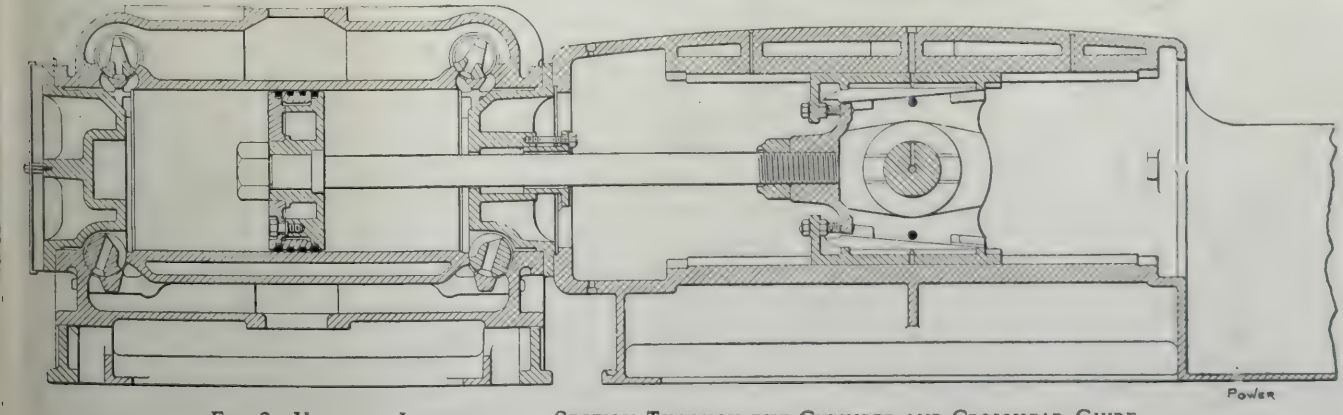


FIG. 3. VERTICAL LONGITUDINAL SECTION THROUGH THE CYLINDER AND CROSSHEAD GUIDE

reach rod from the rocker to the crank-end steam arm, and is arranged so that one part of the rod telescopes into the other. A center pin with a spring (shown just below the governor in Fig. 4) on the outer part, when released engages the hole in the near rod and the clamping device is then screwed up, which locks

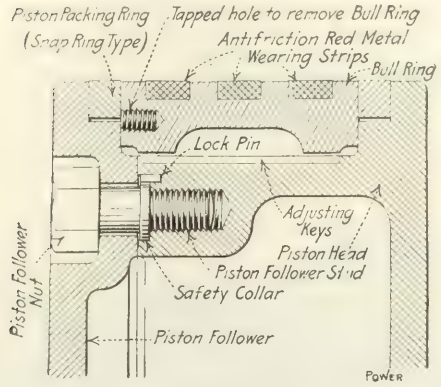


FIG. 5. DETAIL SECTION OF THE PISTON

the two parts of the rod solidly together, precluding lost motion in the disconnecting device. The exhaust valves are actuated through a wristplate, which gives the quick opening and closing movement necessary to properly operate the exhaust valves on a Corliss engine.

One of the most striking features on this new engine is the governor, which is a radical departure, and has proved so satisfactory in its operation, that the

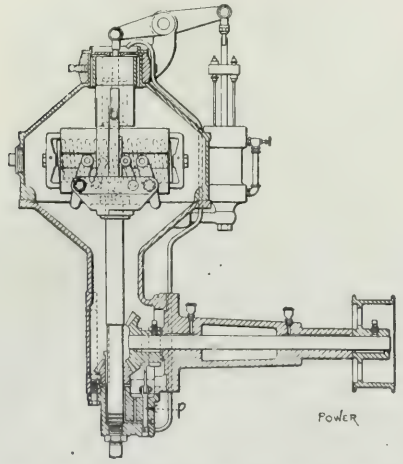


FIG. 6. SECTIONAL ELEVATION OF THE GOVERNOR

Allis-Chalmers Co. has adopted it, and is using it on all steam and gas engines, also steam and hydraulic turbines. Its external appearance is shown in Fig. 5 while Fig. 6 shows it in section. All of the moving parts are inclosed in a cast-iron case which prevents the throwing of oil around the engine room, and, what is more serious, into the generator of direct-connected machines with the serious result of oil-rotting the insula-

TESTS OF ALLIS-CHALMERS TYPE H 20X36 IN. SIMPLE CORLISS ENGINE

Date of test	Steam Pressure 100 lb. Gage Condensing and 120 lb. Gage Non-Condensing	Steam Pressure 80 lb. Gage						Steam Pressure 100 lb. Gage							Steam Pressure 120 lb. Gage							Steam Pressure 100 lb. Gage Condensing and 120 lb. Gage Non-Condensing							Steam Pressure 100 lb. Gage
	11–11–10	11–14–10	11–15–10	11–16–10	11–16–10	11–17–10	11–17–10	11–17–10	11–21–10	11 22 10	11–23–10	11–23–10	11–25–10	11–25–10	11–25–10	11–28–10	11–29–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10	12–21–10
Approximate load	Full	Full	1¼	¾	¾	¼	¼	¼	Full	Full	¼	1¼	¼	¾	¾	Full	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼	¼
Duration, hr	2	2	2	1	1	1	1	1	3	3	1	2	1	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2	2
Steam pressure at throttle, lb. gage	100.33	82.56	80	81	80.6	81 71	100.71	100	119.6	119	119 2	119.3	120	121.1	119.34	99.78	100												
Steam temperature in calorimeter, deg.F	275	272	275	262.14	269.5	258.6	264.1	274.6	277 3	277 3	277.2	269.4	273.89	275.2	271.56	273.9	268.85												
Exhaust pressure at cylinder, lb. abs	4.2	13.62	13.72	13.32	13.38	13.38	13.38	13.57	13.35	13.3	13.61	12.8	13.03	22.2	22.2	13.75	13 14												
Revolutions per minute	120.67	120.96	121.89	120 08	120.58	121 08	120.92	120.8	120.58	120.87	120.77	121.32	120.64	121.02	120.43	120.18	120.06												
Total steam per hr. (corrected for condenser leakage) lb	7395	6299	8138	3763	4958	2884	3108	7176	7778	4010	9996	3001	5712	8256	6163	9860	5269												
Quality of steam, per cent. dry	99	98.4	99.6	98.8	99.25	98.6	98.5	99	98.8	98.8	98.8	98.3	98.5	98.6	98.5	99.3	98 7												
Total dry steam per hr., lb	7320	6260	8100	3720	4920	2844	3061	7104	7885	3962	9875	2950	5626	8141	6071	9785	5200												
M.E.P. head end lb	54 22	39.85	48.39	20.77	30.76	12.2	14.91	48.43	53.96	25.19	67.85	15.8	40.43	52.29	38.03	62.81	35.61												
M.E.P. crank end lb	58.44	40.53	48.9	23.98	32.38	15.22	20.15	49.02	55.93	31.36	68.97	21.27	43.35	53.77	39.72	63.46	37.21												
Total I.H.P., lb	381.3	272.6	332.2	150.4	213.2	92.85	118.4	329.7	371.3	191	462.8	125.6	282.6	359	261.9	425.1	244 7												
Dry steam per I.H.P. (actual as per test), lb	19 2	22.95	24.35	24.73	23.1	30.6	25.86	21.56	20.7	20.74	21.32	23.48	19.9	22.68	23.17	23	21 21												
Dry steam per I.H.P. (back pressure reduced to 16 lb. abs.) lb		24.85	26.23	27.2	25.14	33.6	27.65	23.18	22.3	22.2	22.77	25.55	21.63			24.6	23												
Kw. output (by integrating wattmeter)	264.2	174.6	218.8	88.3	130.8	43.83	62	213	244	116.6	310	71.7	180	237.5	167.1	283	152.2												
Dry steam per Kw. (actual as per test) lb	27.73	35.83	37	42.13	37.6	64.9	49.4	33.4	31.5	33.95	31.85	41.15	31.25	34.28	36.3	34.57	34.13												
Dry steam per Kw. (back pressure reduced to 16 lb. abs.) lb		38.80	39.84	46.35	41	71.3	52.8	35.86	33.95	36.35	34	44.77	33.96			36.96	36.95												
Mechanical efficiency per cent		89.9	92.2	84.6	86.9	74.4	77	90.4	92	86.8	93.75	83.6	89.3	92.5	89.5	93.1	87.4												
Friction HP		27.8	20.2	23.2	28.2	23.75	27.2	31.7	30.3	25.1	29.8	20.6	30.3	26.8	27.5	29.1	30.9												
Thermodynamic efficiency (based on I.H.P.) per cent	58.5	77 4	74.7	72	77 5	57 8	63 6	75.8	73 3	73.6	71 8	63.8	75 3	82	81 5	71.5	76 2												

ion. This governor is of the single-tension spring type, extremely powerful and sensitive in its operation, and so arranged that a stream of oil is automatically delivered by the little pump *p*, Fig. 6, into the top of the case, keeping all moving parts of the governor flooded.

Within a few months, this type of governor has been installed on seventeen 10,000-hp. engines in the Seventy-fourth St. and Fifty-ninth St. power stations of the Interborough Rapid Transit Co., replacing the old weighted fly-ball governors. The regulation obtained with the new governor is so much better that the expense of making the change was justified, although the old type of governor was considered good enough to have remained in use for the past eight or ten years.

The accompanying table gives some test results from a 20x36 simple engine under different conditions of load and pressure, condensing and noncondensing.

Flue Gas Analysis for Beginners—III

By J. E. Terman

Collecting an average sample from the boiler, the apparatus for doing it and full directions for carrying on the work.

Orsat totals as a check on the analysis.

In any test, carefulness with each part contributes to accuracy in the final result, but generally one or more main features are more important than the others. Thus, in a boiler test, pressures are to be noted, temperatures read, ashes weighed and numerous other records carefully kept, but above all, the amount of water evaporated and the number of pounds of coal burned to produce this evaporation must be accurately ascertained.

The most important feature in flue-gas analysis is obtaining a correct average sample of the gas. The appendix to the code of "Steam Boiler Trials," by J. C. Hoadley, page 287 of Volume 6 of the *Transactions* of the American Society of Mechanical Engineers, advises having an orifice through which the sample is drawn for each 25 sq.in. of the cross-section of the flue, principally because it collects samples of equivalent composition when located at two different points in a flue.

The samples collected are not necessarily average samples of the gas passing through the flue; possibly are not as good average samples as may be collected through a single sample tube properly located in the flue.

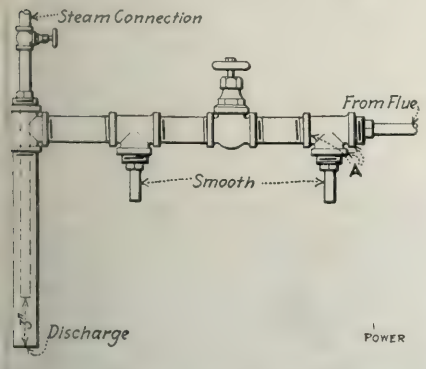


FIG. 1. APPARATUS FOR DRAWING SAMPLE FROM FLUE

If all of the cross-sectional area of the flue was occupied by the products of combustion moving at a uniform velocity, an arrangement such as was recommended by Mr. Hoadley would be all right. As an actual fact, however, the velocities vary widely at different points in a flue, and at some points the flue is carrying only air leakage from the outside. Evidently, under such conditions, a sampler which draws an equal volume from all parts of the cross-section of a flue in a given time, does not obtain an average sample of the gases passing through it.

Suppose a flue carries O and CO_2, both gases entirely separated, each occupying exactly one-half of the area of the flue and the CO_2 moving twice as fast as the O. A true sample of the gas passing through the flue would contain two volumes of CO_2 to one of O, but a sampler that would draw an equal volume from each square inch of the cross-sectional area of the flue would show equal volumes of O and CO_2. The actual variations in velocity greatly exceed those just assumed, and the gases are not of uniform composition. Other disadvantages of the sampler just described are its high cost and its being difficult to install.

Usually sampling is done by drawing the sample through a single tube with the end open and usually perforated part of its length. The tube should have its open end at that part of the flue where the gases move the swiftest, for a sample taken here is most nearly a true average one. To eliminate the effects of air leakage through the setting, the tube is often placed immediately adjacent to the furnace, but sampling here being difficult, due to high temperatures, the writer would advise the beginner to be satisfied with a sample taken at the nearest available point after the gases leave the boiler.

Where the gases are moving fastest can usually be found by looking for the path of probable least resistance, usually the shortest path, and where gas may be deflected, or made to eddy, by noticing the shape of the flue connection. A few tests of samples drawn while a uniform fire is maintained, with the sample pipe moved to different points in about the location judged to be right, will usually determine the correct point by the higher CO_2 content in the gas. Another way to locate the proper point for sampling is to measure the temperature of the gases and place the end of the sample pipe at the hottest point.

ASPIRATOR

Several different means may be used to collect the gas sample, but the writer considers the one about to be described the best that has come to his notice. The sample to be analyzed should be taken from a large quantity drawn in a steady stream from the flue. To get it a draft must be produced considerably in excess of that caused by the stack. An aspirator to produce this draft, which may be operated by steam, water or compressed air, is shown in Fig. 1. This can be made of ¾-in. and ¼-in. pipe.

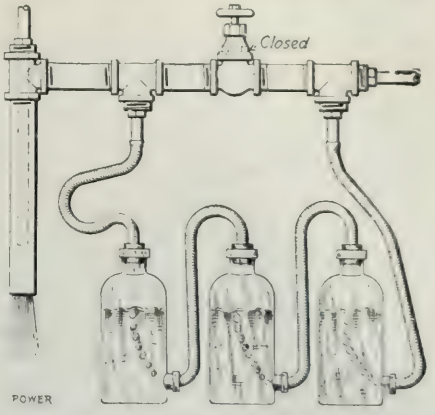


FIG. 2. SATURATING WATER WITH FLUE GAS

The discharge end should be about 8 or 10 in. long and the internal steam pipe should be about 3 in. shorter than the outer pipe. The tees shown should be connected together by short nipples. The two ¼-in. nipples projecting downward from the horizontal tees should be about 2 in. long and ground, or filed smooth on the outside so that pieces of rubber tubing may be slipped over them easily. The valve between the tees may be either globe or gate. A valve regulates the flow of steam and the amount of draft produced.

The joints *A* and all the joints on the pipe leading from the flue, must be perfectly tight to prevent an inflow of air from affecting the sample. Fig. 2 indicates how a draft is produced by the outrush of steam, causing a partial vacuum in the vertical pipe above the discharge nozzle. The vacuum causes a flow of gas in the pipe from the flue when the valve is open and the two outlets below the horizontal tees are closed.

Bottles for Collecting Samples

Some kind of portable container to collect the gas to be analyzed is required. For this and for saturating water with

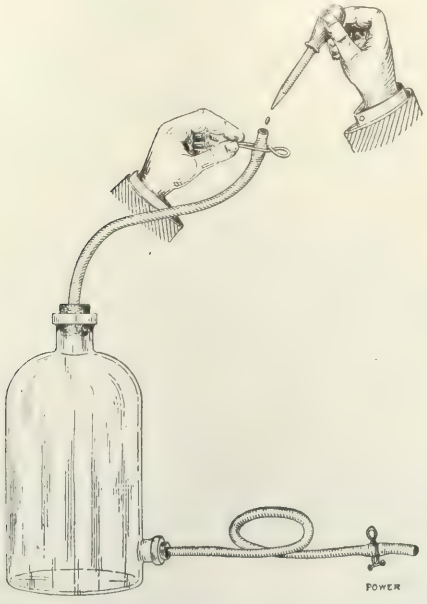


Fig. 3. Completing the Filling with a Dropper

the flue gas, the writer has found a double-necked bottle, such as is shown in Fig. 2, very convenient. Short lengths of glass tubing inserted through holes cut in rubber stoppers are convenient for attaching the rubber tubing. Stoppers can be easily made from round gasket stock of suitable diameter. The rubber tubes attached to the bottom connections should be the same length on all bottles, as this length determines the head of water tending to cause a flow from the bottle while collecting the sample.

The pinch-cocks on the rubber tubing serve to stop the flow of gas or water when desired. They should be as near the ends of the tubes as possible, to avoid air pockets when making connections with other apparatus. Many recommend using pinch-cocks to regulate the flow of water from the sampling bottle, and consequently the amount of the sample collected in a given time, counting the drops of water to judge the rapidity of flow. After numerous attempts to follow this advice without very good results, the writer adopted the following method:

While taking a sample for a given time the flow of gas into the sample receptacle should be uniform. The composition of gas in a flue is constantly changing, and if the rate of flow of water from the sample bottle varies, the resulting sample will not be a true average. To obtain a predetermined rate of flow, the writer took pieces of glass tubing about 2½ in. long, and heated one end in a bunsen burner until it was nearly closed. A number of these tubes with openings of various diameters were tested for flow by connecting them to the bottom of a sampling bottle filled with water. Each was marked with the number of minutes it took to empty the bottle of two-thirds of its contents, this representing a good quantity for a gas sample when using the size bottles adopted.

To take a continuous sample of flue gas to be analyzed at 30-min. periods, a 30-min. tube would be selected. The manipulator could then open the pinch-cock on the rubber tube and without further attention to it, be certain that in 30 min. he would have the correct quantity of gas for a sample. Fig. 4 shows one of these tubes at *B*.

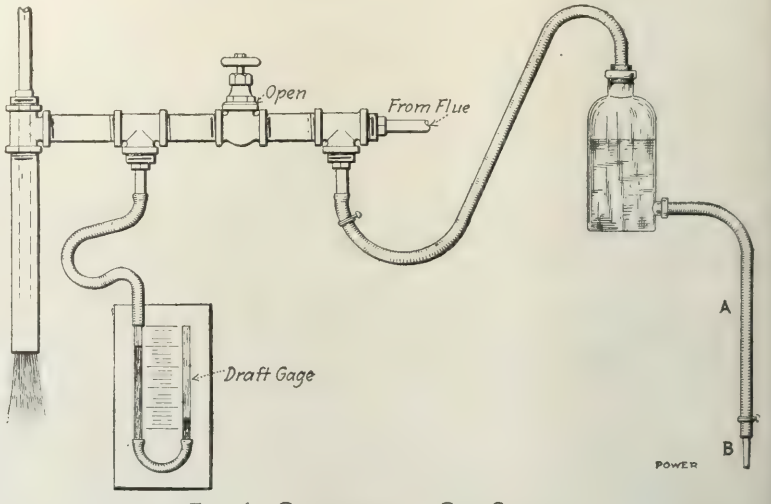


Fig. 4. Collecting a Gas Sample

Liquid Over Which Sample is Collected

To collect a gas sample, the bottle is first filled with some liquid to be displaced as the gas collects. Mercury, oil, water and other liquids have been used, but for ease of handling and cheapness, water is to be preferred by the beginner and generally by the experienced operator, when certain precautions are taken. If a sample remains long in contact with water, some of the CO_2 content will be absorbed. Therefore, to get accurate results, the water should first be saturated with the gas to be analyzed.

This can be done by using the valve on the horizontal pipe shown in Fig. 1. Several sample bottles are connected together and to the two outlets, as shown in Fig. 2, and the ¾-in. valve closed, steam is then turned on the aspirator until gas begins to bubble up through the water. Quite a draft will be required if three or four bottles are used because the head to be overcome is the combined depth of water in all of the bottles. Complete saturation of the water will take about 30 min.

Collecting the Sample

The bottle in which a sample is to be collected and all the rubber tubing and its connections should be completely filled with the saturated water. A medicine dropper is convenient to complete the filling of the tubing, used as illustrated in Fig. 3. This done, the bottle is attached to the sampling apparatus, as in Fig. 4. It should be on the side of the valve nearest the flue, so that any leakage through the packing gland, or other joints, will not affect the sample.

The draft gage, although not essential, is convenient to use in sampling because it enables obtaining uniform draft conditions. Varying the amount of draft while collecting a sample will have the same effect as varying the length of the tube *A*. The draft gage gives a positive indication of a substantial flow of gas through the pipe from which the sample is being drawn.

After the bottle is connected, the aspirator is started, the ¾-in. valve in the horizontal pipe being opened and the steam flow adjusted until there are 4 or 5 in. of draft on the sample pipe. After making these adjustments, a short time should elapse for all air in the pipes to be removed, then both pinch-cocks on the connecting tubes are opened. Care should be taken to have the tube *B*, for regulating the flow of water, inserted in the bottom connection. To remove the sample from connection with the aspirator, always close the pinch-cocks first. While collecting a sample, the saturated waste water from *B* should be caught in

another bottle, or receptacle, so that it may be used over again.

Orsat Totals

A beginner knowing little about chemistry is likely to make the mistake of assuming that as long as he follows the directions the results are bound to be correct. He should be always on the lookout for mistakes when he obtains strange or unexpected results in analyses, and not conclude too hastily that something is radically wrong with the furnace or the firing methods until he is certain that the sample represents the average of the products of combustion and was correctly analyzed.

A fairly reliable check on an analysis is to add the total of the CO_2, O and

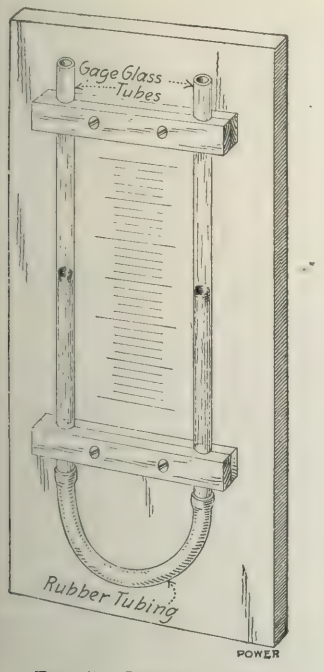


Fig. 5. Draft Gage

CO, as found for each analysis, or the Orsat totals, as they are called. But for the presence in the flue gas of small quantities of other gases besides CO_2, O, CO and N, the Orsat totals for each analysis would be about 21 per cent. or more; instead the totals rarely reach this amount, especially right after firing fresh fuel. However, they should very seldom be below 18.5 per cent., unless the sample is taken during a very short interval of time immediately after firing.

A great many analyses of flue-gas samples were made for the boiler division of the United States Geological Survey at the fuel-testing plant at St. Louis during 1904, 1905 and 1906. Out of 270 such tests, as reported in "A Study of Four Hundred Steaming Tests," the average Orsat totals were 19.17 per cent. with a minimum of 18.45 and a maximum of 20.07. These tests were made by expert chemists, and the figures obtained may be considered representative for such work. If the Orsat totals are less than 18.5 per cent., another portion of the gas sample should be analyzed with the greatest care, so as to be certain of results. The chief cause of incorrect analyses is haste to complete the operation. At least 20 to 30 min. with skillful handling is required with the apparatus described here, and the beginner should take longer until he becomes accustomed to the work.

Testing Furnace Conditions

With draft conditions uniform and the fuel evenly distributed over the grates, a certain thickness of fire for each kind of coal will give the best results. After this has been determined the boiler attendants can tell fairly well by the condition of the fire whether the fuel is being economically burned. To determine the proper thickness of the fuel bed, a boiler should be run for two or three hours with the fire of the thickness to be tested, and a continuous sample of the flue gas taken and analyzed at half-hour periods. Records should be made of the results obtained, and different thicknesses tried in the same manner. The thickness of the fire should be changed about 2 in. at a time until it reaches a thickness where the lack of draft, or other conditions, render its further increase impracticable.

After all tests have been made, it is well to repeat them before trying to deduce conclusions. Then, if the analyses show approximately the same results, it is fairly safe to assume that the samples are representative ones for the furnace and fuel tested and a decision can be made as to the thickness of fire producing the best combustion. The CO_2 may not run as high as it should, due to air leaks, but the best results shown by the analyses will indicate the proper thickness of fire.

If, in a horizontal tubular boiler, the highest CO_2 is less than 10 per cent. and with not more than 0.1 or 0.2 per cent. of CO present, air leaks should be looked for. If it is desired to get the CO_2 up to about 12 per cent., the setting will probably have to be painted with one or two coats of thick paint to prevent air leakage. The latter has a pronounced effect on the efficiency with any type, but is particularly detrimental with the horizontal tubular boiler, as in it all leakage mingles with the heated gases before they pass through the tubes which comprise about 90 per cent. of the heating surface in this boiler.

To discern if the fireman handles the fires properly at all times, a large receptacle, say of 5 gal. capacity, can be used for collecting the gas sample, and a continuous one taken over a 10-hour period. A portion of this sample may be transferred to a smaller sample bottle and analyzed the following day, if it is not convenient to make the analysis at closing-down time.

Draft Gage

A simple draft gage for use as described before, can be made of two glass water gages connected at the lower ends by rubber tubing, as in Fig. 5. The outfit is fastened to a small board and a paper scale is pasted or tacked to the board between the tubes.

Brown Double Flue Return Tubular Boiler

Patents have been granted James T. Brown, 71 Griswold St., Detroit, Mich., who has invented a new design of steam boiler. It is essentially a return-tubular boiler of ordinary make, to which two double flues are attached by water legs. These flues run diagonally across what would be the combustion chamber with an ordinary setting. The smokebox of the boiler is separated by a brick partition wall, as is also the furnace. These flues are built into the bridgewall at one end and into a separating wall at the rear end of the combustion chamber. As one flue is higher than the other by a little more than its diameter, the furnace grates are placed at a corresponding level and in line with the inside diameter of each flue.

The smoke chamber of the boiler is provided with a damper on each side of the partition wall. In operation one side of the furnace is fired and the products of combustion pass through the tube connecting to that side of the furnace into a rear chamber communicating to one-half of the tubes of the boiler. The gases then pass toward the front of the boiler through the small tubes and down through the open damper to the opposite furnace for the purpose of igniting the unconsumed gases by the flame in this furnace. The gases of combustion from the second furnace then pass through the second flue into the opposite partition in the back of the setting to the front of the boiler through the tubes on the other side of the boiler, escaping to the atmosphere.

When the second furnace is fired, the opposite damper in the smoke chamber of the boiler is opened and the same operation is repeated. In this way the unconsumed products of combustion after each firing are passed through a second furnace where they are ignited. This necessitates opening and closing a damper at each firing and also demands that alternate firing shall be used.

The Union Electric Light & Power Co., St. Louis, Mo., will lay 113 miles of electric cable to carry current to be derived from the Keokuk hydro-electric plant. The total cost will be about $1,000,000.

Tom Hunter, Hoisting Engineer

By Warren O. Rogers

Hunter explains mine haulage systems, what produces deterioration of the ropes and the method of inspecting them.

The stresses of ropes, due to load and the bending around sheave wheels, are important matters to consider in rope haulage systems.

While going to my room one evening my attention was directed to haulage cables by the slapping of the elevator ropes. This was a subject upon which Hunter had not touched, and meeting him later in the evening I produced a good cigar and informed him that I wanted some information regarding mine haulage, wear on ropes and other general data. We went into the grill room and while partaking of a light lunch, Hunter enlightened me regarding rope haulage systems.

Haulage Systems

"There are several systems of mine haulage," said Hunter, as he inspected a "bluepoint" oyster, "not including electric and air locomotives. There are four systems which may be grouped under rope haulage: the gravity plane, engine plane, tail rope and endless rope.

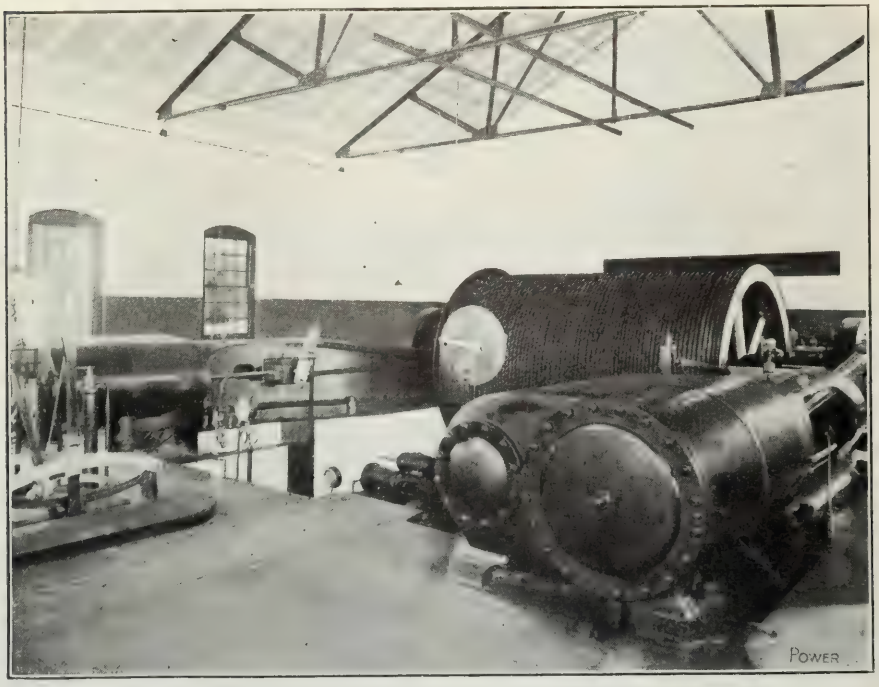


Fig. 1. Haulage Engine with Straight-faced Drum

"A gravity or inclined plane is one on which the gravity pull of the loaded cars will draw the empty cars to the top of the incline. No power is used to operate the system other than gravity. The tracks usually have a pitch of from 3 to 10 deg., according to the length of the incline. The rope is wound on a long drum; this is a better arrangement than when winding on a head wheel."

"Why is that?" I asked, nibbling a bit of salad.

"Because the head wheel is grooved and the rope will wind on the high side and wind off at the lowest point, so that there is a continual sliding of the coils of rope from the outside to the inside of the wheel groove. With a flat drum this surging of the rope does not occur. If two sheave wheels were used, one behind the other, the rope would wind around each, making a half turn at a time about them; but the wheels should be grooved similarly to the drums of winding engines. The speed of the cars is controlled by a brake arrangement at the drum.

Engine Plane

"An engine plane is an incline up which the loaded coal cars are drawn by a haulage engine (Fig. 1) located either in the mine at the head of the slope or on the ground surface. On some of the small engines the drum is loose on the shaft so it is not necessary to reverse the engine when 'empties' are run down the slope. On large outside engines the drum is generally secured to the crankshaft and the engine is run reversed when handling 'empties' down the plane. The other two systems of rope haulage are not so much used on slope work."

"It must be hard on wire haulage ropes, especially in wet mines."

Rope Deterioration

"It is," replied Hunter, "for they are subjected to excessive wear and tear. The deterioration of wire rope is both external and internal. The first is due to the rope wearing against foreign objects, and the latter from the chafing of the wires against each other. Another cause is permanent distortion, caused by overloading the rope or from bending around sheaves of too small diameter. Corrosion of the wire from acid water is another cause for deterioration."

Fig. 2 shows an arrangement of guide sheave wheels.

"I suppose a larger rope is used than is really necessary to handle the load to make up for the wear." I made this remark at a venture.

"That is the practice at some mines the outer surface of the rope being made of as large wires as is possible for the diameter used.

"Some managers make a mistake by increasing the speed of the rope in order to handle a greater tonnage of coal, but high rope speed increases the wear and tear. A better way is to increase the load on the rope, within limits, of course thus producing a greater output with very slight increase in the wear on the rope."

"The strain on a cable must be tremendous if the load is not handled intelligently," was my next remark.

"Well, most engineers are careful in starting, but, of course, if care were not exercised in starting with a load the life of the rope would be shortened. Every hoisting rope about a mine should be inspected every morning before the men are allowed to get on the cage."

Examining the Rope

"How can a rope be examined, covered as it is with dirt and grease?" I asked.

"The best way is for the engineer to run the engine slowly and have a man at the rope to do the inspecting for any signs of weakness. This man passes a piece of hemp rope in a single turn around the hoisting rope, holding each end in his hands. If there are loose wire ends sticking out of the cable (Fig. 3) they will catch on the rope in passing. A bunch of oily waste is sometimes held in the hand and the cable allowed to run through it. If there are any broken ends projecting from the cable they will catch in the waste.

"Haulage cables are not so likely to cause serious accident by failure as a

FIG. 2. ARRANGEMENT OF GUIDE SHEAVE WHEELS ON A CURVE

hoisting cables, and are therefore not discarded as quickly. Ropes with broken wire should be discarded and when the stretch has gone the rope should be replaced by a new one. Some superintendents do not wait until the rope shows signs of failure, but change it when a certain tonnage has been hoisted."

"Is there any particular place where a rope begins to give out?" was the next question.

"Surest thing in the world!" answered Hunter, as he dissected a small block of ice cream. "The first few feet of rope next to the cage, and where the rope enters the socket, are subjected to more severe shocks and sharper bending than other portions of the rope. The rope between the head sheave and drum when the cage is at the bottom of the shaft is also subjected to severe strains. Some ropes will give out much quicker than others."

ATTACHING THE ROPE

"How is the rope attached to a drum? It must be firmly secured as so much depends upon it."

"A common method is to pass the rope through a hole in the shell of the drum and twice around the shaft, the loose end being clamped to the protruding rope between the shell and the shaft. The precaution to observe is that of making the radius of the hole as large as possible so as to avoid a sharp bend at that point."

"Isn't there something that can be used to protect the ropes? I know that some kind of dope is used on gears to lessen the wear."

"A common preventive against the action of acid mine water, and to eliminate wear to some extent, is the frequent application of tar and black oil. A compound for hoisting ropes is made from freshly slacked lime which is mixed with coal tar. It is applied by saturating a piece of waste and the cable run through it. For haulage ropes raw linseed oil, thickened with lampblack and boiled with an equal amount of pine tar, is used. Black oil is also used where friction grips are not employed."

One matter which was not clear to me was how the ropes were attached to the cable hook, cage and drums for a hoisting engine. When the question was asked, Hunter replied that a rope socket was used to fasten the rope to the cage.

"It has a tapered hole through the body," said he. "The rope is put through the small end of the hole, the end projecting a short distance. Then the end of each strand is untwisted, some of the wire cut out and the remaining wires are bent back on themselves. The rope is then drawn back into the socket and a round wedge is driven into the center of the rope to spread the wires against the socket.

"Rope thimbles are sometimes used instead of a socket. The rope is bent around the thimble and the end secured by clamps. Where hooks are used on haulage ropes they are linked to the thimble." Fig. 4 shows one method of attaching a chain and hook to a haulage rope.

ROPE SIZES AND STRESSES

"How do engineers figure the size of ropes to withstand the stresses?" I next asked.

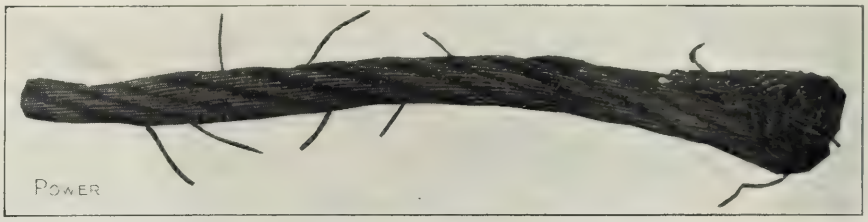


FIG. 3. CONDITION OF HOISTING ROPE AFTER SIX MONTHS' USE

FIG. 4. SURFACE SLOPE OF THE SUSQUEHANNA COAL CO., NANTICOKE, PENN.

Hunter leaned back in his chair and said: "Say, you are the most inquisitive cuss I've ever met, but you are learning a whole lot. The working load of a hoisting rope is one-fifth of the breaking strength. In any event, the working load should never exceed the elastic limit.

"Now, there are two stresses on hoisting ropes, one from the load being handled and the other from the rope bending about the engine drum or around a sheave wheel. The latter stress is often given no attention when calculating the working load of a rope. This is one reason why some ropes fail unexpectedly; the bending stresses sometimes are greater than those exerted by the load.

"I have a table somewhere," continued Hunter, fishing through the contents of a long, bulging pocketbook. "Ah! here it is," giving me a blueprint bearing the heading, "Wire Ropes for Inclined Planes."* This table gave the tension on ropes from 1- to 90-deg. planes for a 2000-lb. load.

FIG. 5. HAULAGE ROPES DRAGGING ON GROUND BETWEEN SPOOLS

Wire Rope for Inclined Planes		Grade of Slope in Feet and Degrees	
Degrees of Incline	Tension on Rope in Lb. per 2000 Lb. load	Grade in Ft. in 100 Ft. of Incline	Incline in Deg.
90	2000	1	¾
80	1970	2	1½
70	1880	2½	1½
60	1732	3	1¾
50	1552	3½	2
45	1414	4	2¼
42½	1350	5	3
40	1286	6	3½
37½	1216	7	4
35	1148	8	4½
32½	1074	10	5¾
30	1000	12½	7
27½	922	14	8
25	846	16	8
22½	764	18	10 1–10
20	684	20	11¼
17½	600	23	13
15	518	25	14
12½	432	26½	15
10	348	30½	17
8	278	36	20
7	242	46½	25
6	208	57½	30
5	174	75	35
4	138	84	40
3	104	100	45
2	70	....	..
1	34	....	..

*Courtesy of the Hazard Mfg. Co.

"The total strain on a rope for shaft work is equal to the load lifted, the weight of the rope, the amount of frictional resistance to be overcome by the load and rope, and the amount of resistance due to acceleration of speed in hoisting.

"A thing which most hoisting engineers don't know is that this last factor enters into the strain upon the rope when conical drums are used. On inclines the resistance due to acceleration of speed enters into the strain upon the rope and varies with the angle of the slope."

"Well, tell me how to work this table?" said I, indicating the table shown herewith.

"In those tables no allowance has been made for frictional resistance or for the strain due to acceleration of speed in hoisting. To determine the strain upon a rope for a given slope with a given load, just multiply the strain per ton given in the table for the particular angle of slope by the number of tons in the load and make the necessary allowance for additional strain, indicated in the table, due to a sharp incline in the slope. A maximum of 150 lb. per ton is to be added for the rolling friction of the cars. Taking a factor of safety of from five to seven or a working load of from one-fifth to one-seventh of its maximum strength, the rope is in no danger of being overloaded."

Hunter arose, lit a cigar and as we passed out of the grill room concluded his explanation of ropes by saying: "The main thing about making a rope last is to run it over large-sized drums, sheave wheels and guide pulleys, which should be increased with the working load when it is greater than one-fifth of the ultimate strength of the rope."

As we passed the cleark's desk, he handed Hunter a note from a mine superintendent consenting to our viewing the mine-pumping machinery the next day. Not knowing what experiences might be encountered, I spent the rest of the evening getting together clothing suitable for the visit.

Absorbing Vibrations of Steam Mains

BY A. L. TEMPLE

Every operating engineer and superintendent of a steam power plant has experienced much trouble with vibrating steam mains and leaky joints. The writer was connected with a plant in which a 12-in. steam main had a tendency to vibrate and at times would set up a considerable end-to-end motion at a point where the pipe made a right-angle turn into the engine room from the boiler room. On visiting a neighboring plant a device was seen which suggested the one described.

The springs were made of ½-in. material, 2½ in. in diameter and 5 in. long. It was difficult to estimate the force causing the vibration and the writer knows no definite method of calculating it. However, the range of the springs to absorb the shock to which they must be subjected was calculated as follows:*

$$P = \frac{187{,}500\, F\, d^4}{R^3}$$

Where

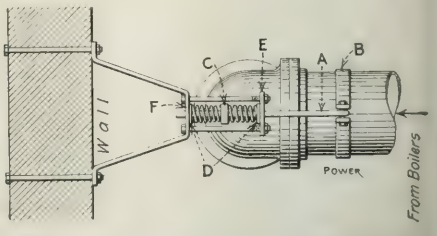
P = Load applied to the springs;
F = Compression of one coil;
d = Diameter of the wire;
R = Mean radius of coil, which is equal to ½ $(D - d)$.

Therefore:

$$P = \frac{187{,}500 \times 0.5 \times 0.5^4}{1^3} = 5850 \text{ lb.}$$

Therefore it is seen that a pressure of 5850 lb. will be required before the coils are tightly closed. From this calculation these springs were considered strong enough to use for a trial, and the device designed with the support f made of ⅜x2½-in. iron and the end piece e of the same material. The bolts on each side of the springs were threaded about 4 in. from the inner edge of the end piece e, so that sufficient tension could be put on the springs when the device was placed upon the pipe.

The sleeve washers d were turned to fit the holes in the pieces f and e and



ABSORBER ATTACHED TO STEAM MAIN

were grooved on the spring side so that each spring would get its pressure always in a true horizontal position. The collar piece c was threaded and screwed on the plunger rod a and secured by locknuts on each side. It was also grooved on each side to center the spring in the same way as at pieces e and f. The band b was made of ⅜x2-in. iron. The plunger rod was ¾ in.

After several adjustments of the springs, sufficient tension was found to absorb the vibration before the coils were compressed tight. However, at times there was a very slight vibration, lasting only a short time, showing that the adjustment was probably correct; the force of the vibrating main was something less than 5850 lb. in this case.

If there are other methods or data on the vibration of steam mains, they should be made known.

*Formula taken from Kent's Mechanical Engineers' Pocket Book.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Inspecting and Testing Electrical Apparatus—V

By A. L. Cook

Power Station Load Tests

After new apparatus has been put in service, it is often desirable to make a load run to determine the performance of the machine under specified conditions. Where it is not possible to use the regular station load for the purpose, some form of rheostat must be provided.

A water rheostat is generally used for loading a generator, its design depending upon the amount of load to be carried and upon the voltage. An ordinary wooden barrel may be used in making a small water rheostat, with one terminal placed at the bottom and the other arranged so that it can be raised and lowered to adjust the load. The terminal at the bottom should be covered by a piece of board so that it cannot come in contact with the upper terminal if lowered too far. The terminals may be of any convenient material, but generally an iron plate or casting is most easily obtained. The form and size of the bottom terminal is not very important, but the movable terminal should preferably consist of a triangular-shaped iron plate so as to vary within wide limits the amount of surface in contact with the water. A square foot of surface would ordinarily be sufficient for this terminal.

The water usually requires treating in some manner to reduce its resistance. Common salt can be used, but this is objectionable when the rheostat is indoors because of the chlorine fumes formed. Washing soda is very satisfactory for this purpose as it does not give off offensive fumes. Sulphuric or muriatic acid may also be used; in fact, almost any substance of an inorganic nature which is soluble in water, will serve the purpose. When adding any substance to the water to reduce its resistance, it should be remembered that as the temperature of the water rises, the resistance decreases greatly; hence the material should be added carefully. The objections to this type of water rheostat are mainly the difficulty of regulation for a steady load and its limited capacity. If water is run into the barrel steadily, a somewhat greater capacity is obtained, but the water escaping from the barrel carries with it some of the salts in solution, so that it is difficult to keep the strength of the solution and consequently the resistance constant.

When building a water rheostat to carry a definite load, it is desirable to have some data concerning the size required. The writer has found that 20 kw. carried on one barrel will cause the water to boil violently. If a barrel is used to carry as great a load as this, difficulty will be experienced in keeping the load sufficiently steady. If the evaporation caused by the boiling is replaced, by allowing a small amount of water to flow into the barrel continuously, about 15 kw. can be handled successfully. Starting with cold water the rheostat will carry heavier loads for a short period. For example, if the temperature of the water is 60 deg. F. at the start, it will carry 55 kw. for 15 min. before, the temperature exceeds 200 deg.

The water-cooled rheostat is much superior to the type just described. This consists of resistance wire, or iron grids immersed in water which is constantly circulated to keep the resistance cool. The load capacity of this type is limited only by the amount of cooling water which can be brought into effective contact with the resistance. No salt or other substance should be used, as the water is employed only for cooling and not as a conductor. The load capacity of a resistance contained in an ordinary barrel and furnished with a plentiful supply of water is surprisingly large. With ordinary cast-iron resistance grids or iron wire immersed in an ordinary barrel of water, 150 kw. can be carried continuously without difficulty. A ½-in. pipe can be inserted into the bottom of the barrel, with a suitable valve for controlling the supply of water. It is desirable to insert in this pipe, an ordinary insulating joint such as is used for combination electric and gas fixtures, to prevent grounding the circuit; the barrel should also be set on insulators. The water can be allowed to run out over the top of the barrel.

The carrying capacity of iron wire when immersed in water is much greater than in air. Table 1 gives the carrying capacity for ordinary sizes of iron wire. This data is calculated from a formula published in the *Electrical World,* May 6, 1909, and also gives approximate lengths of wire required for 125 volts; for other voltages the length can be calculated by direct proportion. The lengths of wire given are only approximate, as iron wire varies considerably in resistance.

TABLE I

Size, Birmingham wire gage	Carrying capacity amperes	Length feet per 125 volts
6	557	91
8	408	82
10	298	74
12	220	67
14	146	58
16	100	52
18	66	44

If the correct load is not obtained at first, it is an easy matter to change the length of wire and determine, by trial, the amount required; generally a single test will suffice. If the normal voltage is applied and the current is determined, the correct length of wire can be calculated from the equation

$$l = L \times \frac{C}{c}$$

where

L = Length of wire tested;
C = Current flowing during test;
c = Correct current;
l = Correct length.

For example, if it is desired to determine the length of wire required for 250 amp. at 125 volts, the required length of No. 10 wire can be calculated from the table and would be

$$\frac{298}{250} \times 74 = 88 \text{ ft.}$$

Assume that it is found by test that only 230 amp. flow with 125 volts applied. Substituting in the equation

$$l = 88 \times \frac{230}{250} = 81 \text{ ft.}$$

This equation assumes that the temperature of the wire remains constant. The tendency would be, however, for the resistance to increase and the current would be slightly less than the amount given by the equation. This will serve, however, to give the approximate length of wire required and save time in making trials. The iron wire can be made up in long spirals and mounted loosely on wooden rods, or can be wound spirally upon a wooden frame built to fit the barrel. It should be kept entirely immersed in the water when carrying current as it would soon melt if exposed to the air. Connection can be made to the coils by means of copper wires.

It is often necessary to make runs at several different loads. This can be arranged conveniently by providing several separate circuits of resistance wire, with switches for throwing on the required coils. The water-cooled rheostat is not readily adjustable but this is no drawback, since generally a constant load is required for testing. If it is necessary

to adjust the load slightly, an ordinary water rheostat with a movable terminal as previously described, can be connected in parallel, most of the load being carried by the water-cooled rheostat.

The water-cooled rheostat is most suitable for voltages up to 2200 volts. The other type, using electrodes immersed in a conducting solution, can be used for much higher voltages, but its use under these conditions requires special precautions. The same kilowatt load can be carried by a given size of water rheostat irrespective of the voltage of the circuit, the size of plate and the resistance of the solution being varied to suit the conditions. Sometimes when rheostats are used on high voltages, for example, 11,000 volts, the water available already contains sufficient impurities to give a suitable resistance. Care should be taken, however, that the voltage drop between terminals of a rheostat is not too great, as burning of the wooden tank would result. The writer has found that six barrels in series on 12,000 volts operate very successfully, and the number could be reduced to four with good results. One electrode in each barrel should be at the bottom and the other near the top, and the distance between them should not be less than 12 in. Variation of the resistance can be made only by adjusting the amount of salt or other substance in solution, as very little change can be obtained by raising and lowering the electrode. The barrels should each be set upon separate insulators and should not touch each other. The wire leading to the bottom terminal must be well insulated to withstand the voltage. It is not practicable to supply water to the barrels for cooling during operation; consequently the capacity is limited to about 15 kw. per barrel, as previously stated.

Generally tests made at high voltage require the absorption of large amounts of power, and in such cases barrels would be unsatisfactory. It is sometimes possible to place the terminals in a pond or river, but this method is somewhat limited, as it is impossible to control the resistance of the electrolyte. While a considerable variation in the resistance between the terminals can be obtained by varying the distance between them and the amount of surface immersed, it is not always possible to obtain the required load. If the surface immersed is reduced too much, arcing occurs at the surface of the terminal and the load becomes unsteady. It should be borne in mind that the largest part of the total resistance is near the terminals; consequently a considerable change in the distance between terminals will not make much change in the resistance; that is, if two plates immersed in a pond are 5 ft. apart, the resistence would not be doubled if the distance was increased to 10 ft. The size of terminal depends upon the current; pieces of iron pipe make good terminals, and need not be very large; a 4-in. pipe, 5 ft. long, being sufficient for 250 or 300 amp. An insulator should separate the terminal and the hoisting apparatus so that no current can pass along the rope and endanger the operator. Means for measuring the current in each leg of the circuit should be provided as a guide to the adjustment of the terminals. When trying a rheostat for the first time, it is best to lower the generator voltage as much as possible before connecting the rheostat so as to remove any possibility of excessive load.

Where the station load is greater than that required for the machine under test, the latter may be operated in parallel with the others carrying the regular load. The load on the machine being tested may be held steady by blocking the governor at the required position, and allowing the other machines to take the fluctuations in load. If the machine is run in this manner, care should be taken that the load is not thrown off with the governor blocked, as excessive speed would result.

When load runs are made, temperature readings should be taken of the principal parts of the machine, such as field and armature windings, field and armature iron, etc. The temperature of the rotating parts must necessarily be taken immediately after the test. The temperature of the air near the machine should also be taken, the thermometer being placed so as to avoid any current of hot air from the machine, but so as to give a reading fairly representative of the conditions of the air surrounding the machine. The bulbs of the thermometers used on the machine should be in contact with the part tested and covered with cotton waste to prevent radiation.

Machines are generally rated for a certain temperature rise above a room temperature of 25 deg. C., which is the temperature specified in the Standardization Rules of the American Institute of Electrical Engineers. If the room temperature at the time of test is not standard, the rules provide that the observed temperature rise shall be changed one-half of 1 per cent. for every degree Centigrade which the room temperature varies from 25 deg.

Storage Batteries

Tests of storage batteries would ordinarily take the form of discharge tests to determine that the battery has the specified ampere-hour capacity. Sometimes charge and discharge runs are made to determine the ratio of ampere-hours output to ampere-hours input. To test the ampere-hour capacity, the battery is discharged at a constant rate until the voltage drops to the specified value. The product of amperes discharged and the time of discharge in hours gives the ampere-hour output. The rating of batteries is usually based on the current that they will deliver for 8 hr., but it is not always convenient to make such a long run when testing. A sufficiently accurate test can be made by discharging the battery at a higher rate for a shorter time. For the ordinary type of lead storage battery, the performance at different discharge rates is given in Table 2, using the 8-hr. rating as 100 per cent.

TABLE II

Time of discharge, hours	Rate of discharge, per cent.	Ampere-hour capacity, per cent.	Voltage at end of discharge, volts
8	100	100	1.8
5	140	87.5	1.75
3	200	75	1.7
1	400	50	1.6

The battery can be discharged by means of a water rheostat or other form of load. Since the potential will be about 2 volts per cell at the start and will drop to 1.6 or 1.7 at the end, suitable provision must be made to keep the current constant during the test. After the battery is fully charged and has been allowed to remain on open circuit for about an hour, the specific gravity of each cell is observed; the discharge may then be started. A single cell should be selected as a pilot cell and frequent readings of its specific gravity, voltage and current made. The temperature of the electrolyte in this cell should also be observed. The voltage of each cell must be noted as soon after the start of the test as possible, and again near the end to detect any low cells. The discharge should be continued until the average voltage per cell, obtained by dividing the total voltage by the number of cells, reaches the specified value; which will be approximately that given in the last column of Table 2. After the test is finished, the specific gravity of each cell should be read to detect any low cells. If the temperature of the battery differs from 70 deg. F., upon which the standard guarantees are based, a correction should be made by allowing one-half of 1 per cent. change in ampere-hour capacity per degree Fahrenheit, the capacity being decreased for temperatures below 70 deg. and increased for temperatures above 70 deg. A change in temperature also affects the final voltage slightly, but for ordinary temperatures the correction is so small that it may be neglected.

Bulletin 4958, just issued by the General Electric Co., is an attractively bound book of 132 pages, devoted to the use of the "higher voltages" and their advantages in the operation of direct-current electric railways. It contains numerous tables comparing the costs of 1200-volt systems with those of 600 volts; illustrates and describes station and car equipments, locomotives for railroads of higher direct-current voltages, and describes the various systems employing

direct current at 1200, 1500 and 2400 volts. The bulletin contains a large amount of interesting information for electric-railway operators.

Tungsten Lamps for the Overloaded Plant*

By A. Gordon

The operating engineer often finds that the plant, which was quite ample when first installed, has become dangerously overloaded.

Not only must the floor areas be abundantly illuminated, but show cases, signs, and all kinds of decorative effects are demanded, and the installing of new lamps goes on so incessantly, that the last few additions may prove to be the proverbial last straw to break the back of the generating camel.

The 16-cp. lamps cost about 25c. each in barrel lots, and the current consumption is 3½ watts per candle, or about 13 lamps per horsepower. With 25-watt tungsten lamps, about 20 per cent. greater candlepower is obtained, and a horsepower will light about 30 lamps. The tungsten lamp offers a splendid opportunity to the operating engineer to economize in the plant with the least possible investment. In the overloaded plant, this lamp has proved a blessing; to the engineer suddenly called upon to provide light or power for additional areas, it offers the simplest solution.

In some buildings the space given to the steam and electrical plant is so limited, that to install an additional boiler or generating unit would upset the whole surroundings, or disturb established departments so much, that the outlay would hardly be warranted by the benefits received. If the plant is large, and the additional current required is for lighting only, and arc or carbon incandescent lights are in use, supplying additional light at small expense, is quite easy.

Assume that the plant has a total capacity of 2000 sixteen-candlepower lamps of 3.1 watts per candle or 50 watts per lamp, and 200 arc lamps taking 5 amp. at 110 volts or 550 watts per lamp. Then the generating units would have to supply 210 kw.

For the 50-watt carbon lamps, substitute 25-watt tungsten lamps, and the power required will be reduced 50 kw. and 19.1 cp. will be obtained per lamp or practically 20 per cent. more light than the carbon lamps give on the same current consumption. Or, if the wiring were rearranged, a very substantial saving could be made in the cost of renewals, as the larger tungsten lamps have a much longer life than the smaller sizes.

For the arc lamps might be substituted 400-watt bowl-frosted lamps, with reflectors, properly selected for the service required, and suspended at the proper height to permit using open prismatic reflectors, of either the intensive or extensive type, depending upon local conditions.

As the tungsten lamp is a depreciating light, and the tendency is always for more light, it is best to use at least 400-watt lamps in most cases. By replacing the 200 arc lamps with the same number of 400-watt lamps, 30 kw. will be saved, and with the 50 kw. saved by replacing the carbon lamps, 80 kw. additional is available for light or power. For power about 100 hp. could be used in motors, or an area equal to more than one-third of that supplied by the old lamps could be lighted.

The writer was obliged to furnish additional light and power, from a unit already overloaded. In 1906, 12,150 sq.ft. additional floor space was added, in 1907, 14,000 sq.ft., and in 1910 a fur room, which requires 22 kw. to drive the refrigerating plant. In 1911 a new building was erected at the rear of the building which shut off the daylight to such an extent that many arc lamps, previously run after dark only, had to be used continuously. This spring, 220 arc lamps were replaced with 300- and 400-watt tungstens, and, although additional lamps were installed, the load was reduced about 250 amp., and now another extension of lights or power, equal to 27.5 kw., is possible.

In addition to its load-reducing possibilities the tungsten lamp has many other advantages. Every engineer who has been responsible for the trimming, repairs, and maintenance of 300 or 400 arc lamps, knows the amount of trouble and labor involved in keeping them up to high efficiency.

When the internal mechanism becomes worn, and the insulation becomes charred, the resistance becomes defective and the current consumption has increased from 4.5 to perhaps 6 amp. per lamp. Where many arc lamps are used, there are sure to be some dead ones from some cause or other, and it usually is a nuisance to trim them. Burn outs of tungsten lamps are very infrequent and it only requires a few moments to replace a bulb.

In old plants where the wiring was not figured as liberally as it would be today, many circuits have become overloaded, the insulation deteriorated, and the copper crystallized so much, that the drop in potential has gone from 1 per cent. as at first calculated to even 10 per cent. in some cases. In other cases the wiring was so overloaded at the point of distribution that it was positively dangerous, when due care was not exercised in the proper use of fuses.

With the change to tungsten lamps, the cost for fuses should be reduced at least 50 per cent. The cost of glassware, especially inner globes, which is quite an item with arc lamps, would probably be sufficient to pay for renewals, when the 300- or 400-watt sizes are used.

For a new building, the cost of installing a plant for tungsten lamps, need not exceed 60 per cent. of the investment necessary, when arc or carbon lamps are used. At present it is possible to install a duplicate generating plant, with boilers of ample capacity, at less than a single plant would have cost a few years ago for the same service. With the use of tungsten lamps, the boiler plant can be reduced very materially, or, if boilers of liberal proportions are installed, a cheaper grade of fuel can be used to do the work. Anthracite screenings or buckwheat can be mixed with bituminous coal, in such quantities as will permit the plant to be operated without violating the most stringent smoke laws.

Only one objection has been advanced to using tungsten lamps in stores, that was in connection with the matching of colors. The complaint was more fancied than real as it was found that the ordinary inclosed arc lamp, when compared with the Day Light Arc or the Moore tube, was just as defective in the matching of certain shades of blues and drabs as the tungsten. Even daylight itself, when reflected from a red brick wall was found to be much inferior to the Moore tube for matching very delicate tints.

*Read before the second annual meeting, Institute Operating Engineers, New York City, Sept. 6 and 7.

CORRESPONDENCE

Burnt Spots on Commutator

In the July 16 issue, Mr. Hill answers Mr. Kramer's inquiry regarding burnt spots on the commutator. He mentions loose contacts, etc., which will probably cause burnt spots, but he did not mention irregular speed of the prime mover.

We have a two-cylinder four-stroke-cycle tandem gas engine, direct-connected to a 220-volt six-pole generator. The trouble started by the commutator burning in three equidistant spots; it kept getting worse and finally the burnt spots met. We turned down the commutator and tested for electrical defects, but found none. The machine ran all right for about four months, when the trouble appeared again.

The trouble was finally traced to the engine. There was poor compression in one cylinder and lost motion in the wrist and crankpins, which caused the engine to run unevenly and to pound. After correcting this the trouble disappeared.

The opinion seems general that high mica causes sparking, but I have always found on a well kept commutator that sparking comes from other causes, and the high mica is the result of the sparking, which burns the commutator and leaves the mica high and black.

J. C. Halvey.

Cooperstown, N. Y.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Some Ignition Troubles

By James H. Beattie

Designs of internal-combustion engines have been much improved during the past few years, particularly in reference to the ignition system. A few years ago the hot tube was the prevailing method of igniting the charges in the engine cylinder. Electric ignition was later developed, but for a long time it was not equal in reliability to the hot tube. It, however, possessed features superior to the older method, and designers devoted much attention to its improvement. Even with the improvements made, the ignition apparatus for a long time remained the most unreliable part of the engine. the custom being to use batteries with a simple induction coil. The demand for automobile engines with absolutely reliable ignition has perhaps done more to hasten the development of the modern high-tension magneto than any other factor.

Of late. manufacturers of stationary engines have followed the lead of the automobile-engine manufacturers, and are putting out engines equipped with "built-in" magnetos, driven by gears, in many cases so designed that the engine may be started without batteries. The liability of trouble from the ignition apparatus has been much reduced. It will be interesting to see how such apparatus will succeed in the hands of the average operator. Of course, the high-tension magneto is harder for the average operator to understand than a set of dry cells and a spark coil, but the liability of trouble is so much less that the average operator can afford to call the expert when trouble does occur.

The owner of a small vertical engine, fitted with make-and-break ignition, supplied by a set of batteries, complained that it missed and backfired. The batteries were in good condition, but it was found that the points of the igniter did not remain in contact long enough to allow the current to build up sufficiently to give a spark hot enough to ignite all the charges, some of which passed into the exhaust unburned and resulted in muffler explosions. Adjusting the igniter trip rod so that the points remained in contact longer effectually cured the trouble.

In another instance a similar engine fitted with the same type of igniter missfired, due to the points being corroded and pitted so as to make poor contact; one point was deeply pitted while the other was considerably longer than it had been originally. This condition is common with make-and-break igniters where the direction of the current is the same all the time; particles of the platinum are carried from the positive electrode and deposited on the negative, causing it to build up. Reversing the wires of the igniter from time to time will prevent this.

It would seem that short-circuited spark plugs or run-down batteries would be apparent to almost anyone, but for such troubles the repair man is often called. Many operators think their batteries are all right, and go so far as to prove it by testing the size of the spark. They fail to remember that a battery having energy enough to give a good spark in testing may be so nearly exhausted that it will not ignite the charges successfully. The real condition of a battery can be ascertained only by an ammeter. The amount of current necessary to operate one engine may be insufficient for another, due to one having a more efficient igniter, and also due to the location of the igniter. If the igniter is located in a part of the cylinder where it is exposed to fresh gas it will operate more surely than if located in a pocket where exposed to burned gases.

A puzzling case of miss-firing was traced to one of the make-and-break igniter points being loose. The point was inserted in a hole drilled in the movable electrode and secured with a small setscrew. The setscrew had worked loose so that the point had slipped back and did not make good contract with the other point. This method of securing the points is very poor, both because the setscrew may work loose and because the heads are likely to become hot and cause preignition.

An interesting case showing how the location of the igniter affected the operation of the engine, was that of a small horizontal engine operating a deep-well pump. The ignition had always given trouble, as ten fresh dry cells were required, and this caused the igniter points to burn because of the strong current. The igniter was located in a pocket in the cylinder head, and a hole about 5/8 in. in diameter communicated from the igniter pocket to the cylinder proper; this hole was nearly 3 in. in length. The repair man could not rectify this, however, as the mistake had been made by the designer. The igniter being located in the pocket was exposed to burned gases practically all the time, there being no opportunity for the pocket to be scavenged, which accounts for a very strong spark being required.

An engineer complained that his engine had a knock which he could not locate. The bearings were in good condition and the trouble proved to be due to too early ignition. The igniter was located over the inlet valve and was exposed to fresh mixture; hence the engine did not require as early ignition as other engines with which the engineer was familiar. The rate of propagation of the flame through the compression space is determined largely by the location of the igniter; hence some engines require earlier ignition than others.

Heat Value of Liquid Fuels

By A. A. Potter

The heat of combustion of a fuel can be determined from its chemical composition, but more easily by means of a calorimeter, in which the products of combustion give up their heat to a known weight of water.

The fuel in a calorimeter may be burned at constant volume or at constant pressure. The various bomb calorimeters, such as the Mahler, the Parr and the Emerson belong to the constant-volume class. In this case a weighed quantity of the combustible is burned with oxygen in a stout receptacle, or bomb, placed in a vessel containing a known weight of water. Ignition is affected by a hot wire, or by an electric fuse, the combustion producing a temperature rise of the water surrounding the bomb. Multiplying the temperature rise by the known weight of water, and dividing the product by the quantity of combustible in the bomb, gives the heat value of a unit weight of the combustible.

A well known calorimeter of the constant-pressure type is the Junker's, a brief description of which will be given for those not already familiar with its operation. Fig. 1 shows this calorimeter arranged for testing gaseous fuels. The gas to be analyzed, after passing through the meter *M* and sensitive regulator *R*, is consumed at constant pressure in a bunsen burner, and gives up its heat to water flowing through the calorimeter *C*. This calorimeter is somewhat similar to a fire-tube boiler, the products of combustion passing up the combustion chamber

and down through copper tubes, which are surrounded by water flowing in the opposite direction. For determining the heat value of liquid fuels, a burner and balance, as shown in Fig. 2, are substituted for the gas meter, regulator and bunsen burner.

In using a calorimeter of the Junker's type the heat of combustion is determined by measuring the amount of fuel burned, the quantity of water passed through the apparatus and the temperature rise of the water. Multiplying the weight of water passed through by the temperature rise and dividing by the amount of fuel used, gives the heat of combustion.

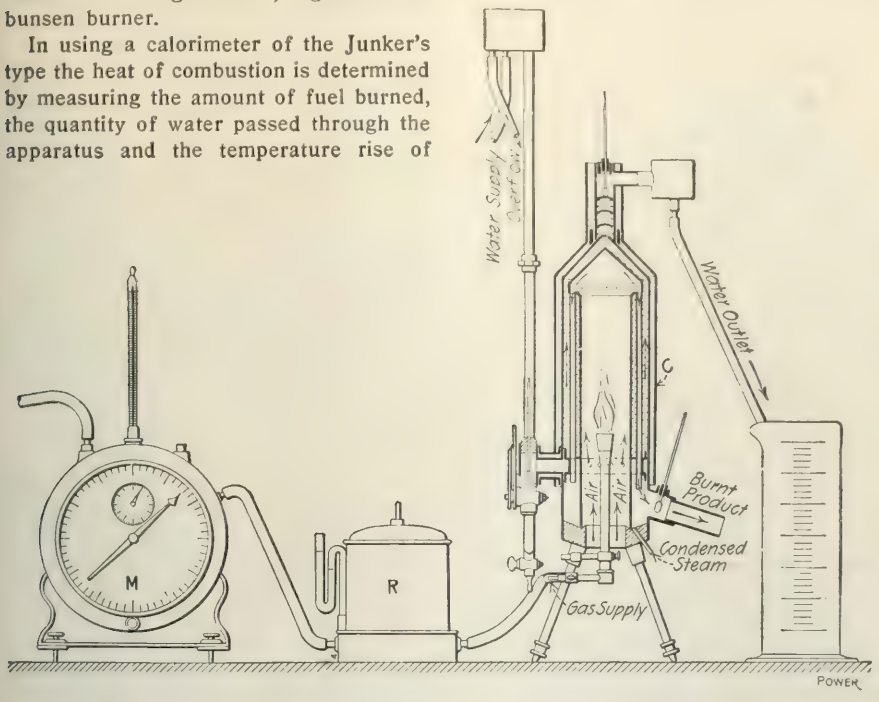


FIG. 1. JUNKER'S CALORIMETER AVERAGED FOR BURNING GAS

sene, solar oil and denatured alcohol by means of the Junker's calorimeter, the results of which are given in Table 1. These values represent averages of 28 runs in the case of gasoline, 29 for kerosene, 17 for solar oil and 21 for denatured alcohol.

The heat values are given in B.t.u. per pound, calories per kilogram and B.t.u. per gallon. From the values of B.t.u. per gallon it is evident that the cheaper petroleum fuels, such as solar oil, contain more heat units per gallon than do the lighter and more expensive fuels like gasoline.

The results of the tests also show that the heat of combustion does not differ much for the various petroleum distillates.

Table 2 was prepared as an aid in the conversion of degrees Baumé into specific gravity and in the calculation of pounds per gallon for liquid fuels.

TABLE I. HEAT OF COMBUSTION OF LIQUID FUELS

Kind of Fuel	Specific Gravity (At 60° F.)	Degrees Baumé	Heat of Combustion: Calories per Kilogram High	Calories per Kilogram Low	B.t.u. per pound High	B.t.u. per pound Low	B.t.u. per gallon High	B.t.u. per gallon Low
Gasoline	0.7380	60	10,790	10,220	19,411	18,415	119,400	113,300
Kerosene	0.7971	46	10,630	10,114	19,130	18,205	127,100	121,000
Solar oil	0.8250	40	10,429	9,965	18,770	17,940	129,100	123,330
Denatured Alcohol	0.7970	46	5,763	5,194	10,373	9,349	68,920	62,120

When a fuel contains hydrogen the foregoing method will give the high heat value, which includes the latent heat of the steam formed by the combustion of the hydrogen in the fuel. In the case of internal-combustion engines, the steam formed by the burning of the hydrogen in the fuel escapes through the exhaust uncondensed; hence the available heat is termed the low heat value which can be determined by collecting the steam condensed in the calorimeter, calculating the latent heat required to produce this weight of water at atmospheric pressure per unit weight of fuel, and subtracting the result from the high value.

TESTS WITH LIQUID FUELS

There being very little recent data published on the heat of combustion of the common liquid fuels, the writer made several series of tests on gasoline, kero-

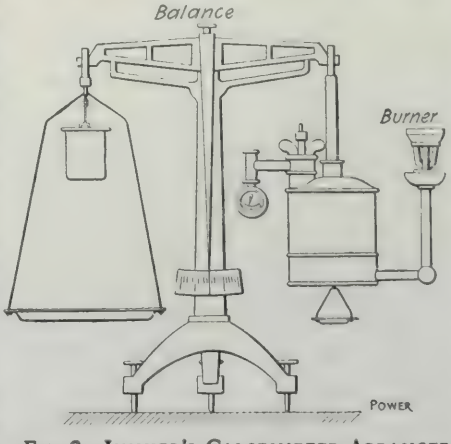


FIG. 2. JUNKER'S CALORIMETER ARRANGED FOR BURNING OIL

TABLE 2. RELATION BETWEEN SPECIFIC GRAVITY AND WEIGHT PER GALLON

Specific Gravity	Degrees Baume	Pounds per Gallon	Specific gravity	Degrees Baume	Pounds per gallon
1.000	10	8.336	0.775	51	6.462
0.993	11	8.277	0.771	52	6.428
0.986	12	8.220	0.767	53	6.394
0.979	13	8.161	0.763	54	6.358
0.972	14	8.104	0.759	55	6.324
0.966	15	8.051	0.755	56	6.290
0.959	16	7.997	0.751	57	6.258
0.953	17	7.944	0.747	58	6.212
0.947	18	7.891	0.743	59	6.195
0.940	19	7.837	0.739	60	6.153
0.934	20	7.785	0.736	61	6.133
0.928	21	7.736	0.732	62	6.101
0.922	22	7.687	0.728	63	6.070
0.916	23	7.638	0.724	64	6.038
0.911	24	7.590	0.721	65	6.006
0.905	25	7.541	0.717	66	5.975
0.899	26	7.493	0.713	67	5.946
0.893	27	7.444	0.710	68	5.916
0.887	28	7.395	0.706	69	5.886
0.881	29	7.347	0.703	70	5.856
0.876	30	7.298	0.699	71	5.827
0.870	31	7.254	0.696	72	5.797
0.865	32	7.210	0.692	73	5.771
0.860	33	7.166	0.689	74	5.743
0.854	34	7.122	0.686	75	5.715
0.849	35	7.079	0.682	76	5.688
0.844	36	7.038	0.679	77	5.659
0.840	37	6.998	0.676	78	5.632
0.835	38	6.696	0.672	79	5.603
0.830	39	6.918	0.669	80	5.576
0.825	40	6.878	0.666	81	5.548
0.820	41	6.839	0.662	82	5.517
0.816	42	6.804	0.658	83	5.487
0.811	43	6.760	0.655	84	5.457
0.806	44	6.721	0.651	85	5.427
0.802	45	6.683	0.648	86	5.402
0.797	46	6.644	0.645	87	5.374
0.793	47	6.608	0.642	88	5.353
0.788	48	6.571	0.639	89	5.316
0.784	49	6.534	0.636	90	5.304
0.779	50	6.498			

A resolution passed by the National Gas Engine Association reads: "It is the sense of this convention that the oil-inspection laws in Wisconsin and similar laws in other states are a burden to the consumer and unnecessary."

A further investigation of the oil-inspection laws of several of the states convinces us that they are inadequate for present conditions. In North Dakota and some other states, owners of traction engines cannot make use of low-grade fuels since the law under which oils are inspected makes no provision for admitting oils of this character. When there was no market for gasoline the tendency of the oil companies was to use as much of the volatile constituents as possible in the kerosene. Inspection was then necessary to protect the public. The old inspection laws are, therefore, practically obsolete. Furthermore, when these laws were passed nobody thought of using distillate for power purposes. What is needed now is either a modification of the oil laws which will enable the free sale of distillate or their abolishment altogether.—*Gas Review.*

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating in Connection with Steam Turbines*

By August H. Kruesi

Using turbines in connection with heating is advocated primarily because any work first done by the steam, as generating electric current, is almost wholly a gain as compared with using live steam separately for heating. Steam, after expanding, and giving up all of the energy available in a prime mover of perfect efficiency, can still give up the great bulk of its original heat when condensed in a heating system.†

As compared with the exhaust from noncondensing engines, steam from a turbine at the same pressure would contain substantially the same heat and amount of moisture, if the efficiency is the same. If the turbine is less efficient, the steam extracted from it must be dryer and contain more heat. The measure of the relative performance is the amount of electrical energy obtained as a byproduct from the steam extracted for heating, assuming the steam delivered to the heating system is of the same quality in both cases. In other words, the comparative performance is gaged by the water rates of the two machines. As between slow-speed Corliss engines of highest grade and turbines designed primarily for condensing operation the advantage, in the past, has often been with the engine, under usual district steam-heating conditions. In hot-water heating systems, the heat transfers are usually made at lower temperatures and correspondingly lower pressure, so that more work may be abstracted by the turbine with very little reduction in the heat value of the steam. Experience with such systems indicates that the turbine is on a parity, at least as to steam consumption, and points to the desirability of laying out district steam systems to employ the same low pressures.

*Abstract of a paper read before the National District Heating Association at Detroit.

†At a steam pressure of 150 lb. gage and a feed temperature of 132 deg. F., expansion to atmospheric pressure in a perfect prime mover would deliver 0.05 kw.-hr. per lb. of steam and leave 84 per cent. of the original heat in the exhaust steam, which would contain 13½ per cent. moisture. In a prime mover of 50 per cent. efficiency 92 per cent. of the original heat would remain in the exhaust and 4½ per cent. moisture. The latent heat of the original steam is 78 per cent. of its total, and most of it remains unimpaired after expansion, and available in any heating system working at a low enough temperature to condense the steam.

Turbines for Heating Work

The author illustrated one of a series of Curtis turbo-generators of new design now in manufacture, at present comprising machines of 500, 750, 1000, 1500 and 2000 kw. capacity for 60-cycle generators at 3600 r.p.m. While primarily intended to give good results in condensing operation, these machines have been especially designed with reference to extracting steam for district and industrial heating purposes. They have governing mechanism, valves, nozzles, etc., proportioned for a sufficient flow of steam to carry the full rated capacity noncondensing against back pressures up to 1 lb. gage with 150 lb. initial pressure.

The performance actually obtained on a 750-kw. unit running entirely noncondensing against atmospheric pressure, is shown in the accompanying diagram. Comparing curves *A* and *B* shows that, in the full-load region, the noncondensing performance of the machine designed to run condensing is practically as good as that of a straight noncondensing machine. The water-rate curve at 10 lb. gage pressure also shows good results.

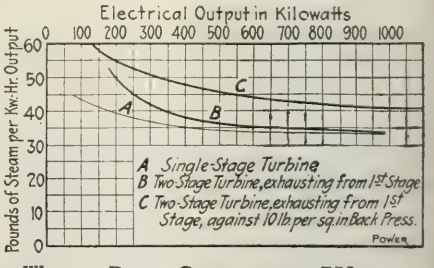
STEAM EXTRACTION FROM 750-KW. CURTIS TURBINE.

Gage pressure extracted steam	Steam extracted, lb. per hr.	Total steam to turbine, lb. per hr.	Combined water rate lb. per kw.-hr.
Electrical output, 250 kw.			
—	0	5,600	22.4
0	9,200	10,000	40.0
5	4,000	8,200	32.8
5	10,500	11,600	46.3
10	4,000	8,600	34.4
10	12,000	13,300	53.0
Electrical output, 500 kw.			
—	0	9,600	19.2
0	17,000	17,800	35.6
5	4,000	12,200	24.5
5	19,300	20,400	40.8
10	4,000	12,800	25.6
10	21,600	23,000	46.0
Electrical output, 750 kw.			
—	0	13,700	18.25
0	25,000	25,700	34.2
5	4,000	16,200	21.6
5	28,200	29,200	38.9
10	4,000	16,900	22.5
10	31,600	32,900	43.9

The accompanying table shows how this turbine can be used to carry different loads, and at the same time deliver various amounts of steam to the heating system, together with the total steam consumption under the assumed conditions. The steam pressure was 150 lb. gage with no superheat and the vacuum 28 in. For back pressures of 5 and 10 lb. the table shows in each case the effect of extracting a small amount of steam and also the maximum amount obtainable under the conditions from a machine of regular design. The condenser is in operation for the balance of steam required to carry the electrical load. At the maximum, the steam extracted is substantially the same as the live steam to the machine, the stage valve in these cases being entirely closed. Curves *A* and *B* show that when extracting the maximum of steam, the condenser might be shut down without materially affecting the economy. These machines are not limited to the figures stated.

Variable Load and Automatic Valves

Operating turbines in connection with a heating system in some cases have been handicapped by a variable electrical load, and in at least one instance, operating a heating system from a turbine has been claimed actually to take more coal than


Electrical Output in Kilowatts
0 100 200 300 400 500 600 700 800 900 1000
Pounds of Steam per Kw.-Hr. Output
60 50 40 30 20 10 0
A
B
C
A Single-Stage Turbine
B Two-Stage Turbine, exhausting from 1st Stage
C Two-Stage Turbine, exhausting from 1st Stage, against 10 lb. per sq. in. Back Press.
Power

Water Rate Curves for 750-kw. Turbo-generator

operating independently, furnishing steam through a reducing valve to the heating system. An increase in electrical load will naturally increase the pressure in the turbine casing and, lacking any regulation to prevent it, will result in a corresponding increase of pressure in the heating system. The whole heating system must be raised in temperature to that of the increased steam pressure. Consequently, if the pressure is suddenly increased the steam-absorbing capacity of the heating system is very great while the pressure is rising. This will increase the draft of live steam to the turbine correspondingly, and may lower the pressure in the boiler room, which would lead to the conclusion that heating by extraction is wasteful of coal, because it is more difficult to maintain a steady steam pressure in the boiler room. The waste is probably more apparent than real in many cases. Consider an increase in pressure from 3 lb. gage to 5 lb., occurring at intervals, so that the higher pressure exists for half the time, the increased temperature and heat-radiating ability of the system, taking a room temperature of 70 deg., is only about 2 per

cent. Station operators will occasionally prefer to use live steam, irrespective of economy, from a disinclination to take the trouble to adjust a hand-operated controlling device to suit the variations in load, when it is so simple and easy to set a live-steam reducing valve to hold any desired pressure in the heating system.

To meet the requirements, however, of a constant heating-system pressure (which may be altered at will) under variable load conditions, an automatic valve-gear has been developed in connection with the new turbines referred to. The valve ports are arranged for successive opening or closing of the nozzles, all being either wide open or tightly closed and only one group subjected to throttling at a time. This valve may be hand actuated to supply variable amounts of heating steam.

An equivalent result can be reached with turbo-generators at present in use, by an automatic reducing valve between the turbine and the heating system. This will prevent an excessive flow of steam to the heating system with a sudden increase in electrical load, and thereby prevent waste of steam which must necessarily result from an attempt to maintain a given average pressure, with frequent variations in pressure, and consequently in temperature and flow. This arrangement cannot be quite as efficient as the automatic stage valve, because it involves some throttling of the heating steam, but this loss may be made small by occasionally adjusting the hand-operated stage valve.

Taking steam from turbines for heating systems is no more complicated than operating engines. A safeguard for the turbine must be provided, however, where there is more than one source of steam to the heating system, in the shape of a nonreturn valve. This prevents the turbine taking steam from the heating leader and so operating as a low-pressure machine, which might result in excessive speed if the circuit-breaker of its generator should trip or its load should fall to a low value for any other reason. Some turbines have been installed and used as high-pressure machines part of the time, and either wholly or partly as low-pressure machines at other times. This often can be done advantageously, simply by providing a connection between the turbine emergency governor and the nonreturn valve, so that the latter will trip in case of excessive speed. The latter would be automatically avoided under all ordinary conditions where the turbo-generator operates in parallel with the prime movers furnishing the steam. This makes a very flexible arrangement in existing plants containing steam engines and helps to maintain the best overall station performance under variations of electrical load and weather, which would otherwise fail to give a favorable relationship between the two loads.

Steam and Hot-water Systems

Some think that hot-water systems have an advantage in connection with power generation by reason of a lower back pressure, or even a partial vacuum on the prime movers. The writer has carefully studied this question in the heating of large factories and is satisfied that, if the prime movers are properly proportioned with respect to the two loads and of suitable type, there is no marked difference, if the steam-distribution system is designed for future growth and care to reduce the back pressure to economical limits. Using a vacuum return-line system has been found practicable in the largest industrial establishments. Some of these systems have more radiation surface than most district steam systems, and it would seem entirely practicable, and economical as well, to make use of this system, at least in part.

The balance of the paper was devoted to the measurement of exhaust steam under pulsating flow.

Ventilation of a Dispensary Building*

By A. M. Feldman

The building is a new dispensary attached to the Lebanon Hospital, New York City. Only one story and basement have been built, but provision has been made for five more stories. The first story, Fig. 1, contains a main corridor running the length of the building, with clinical rooms on both sides. The corridor serves as a waiting room for patients, and as considerable numbers congregate there at one time, efforts were concentrated on the ventilation of the corridor.

The scheme is to exhaust the air from the corridor. To admit fresh air all the radiators in the clinical rooms are direct-indirect radiators, Fig. 2, having fresh-air inlets through openings in the walls under the windows, with special galvanized-iron castings over the radiators that can be lifted to get at the radiators for cleaning. Only a few sections of each radiator are inclosed. To prevent cold air bypassing the radiator, the two sections next to the ends of the casing have solid legs.

The doors have louvers in the bottom

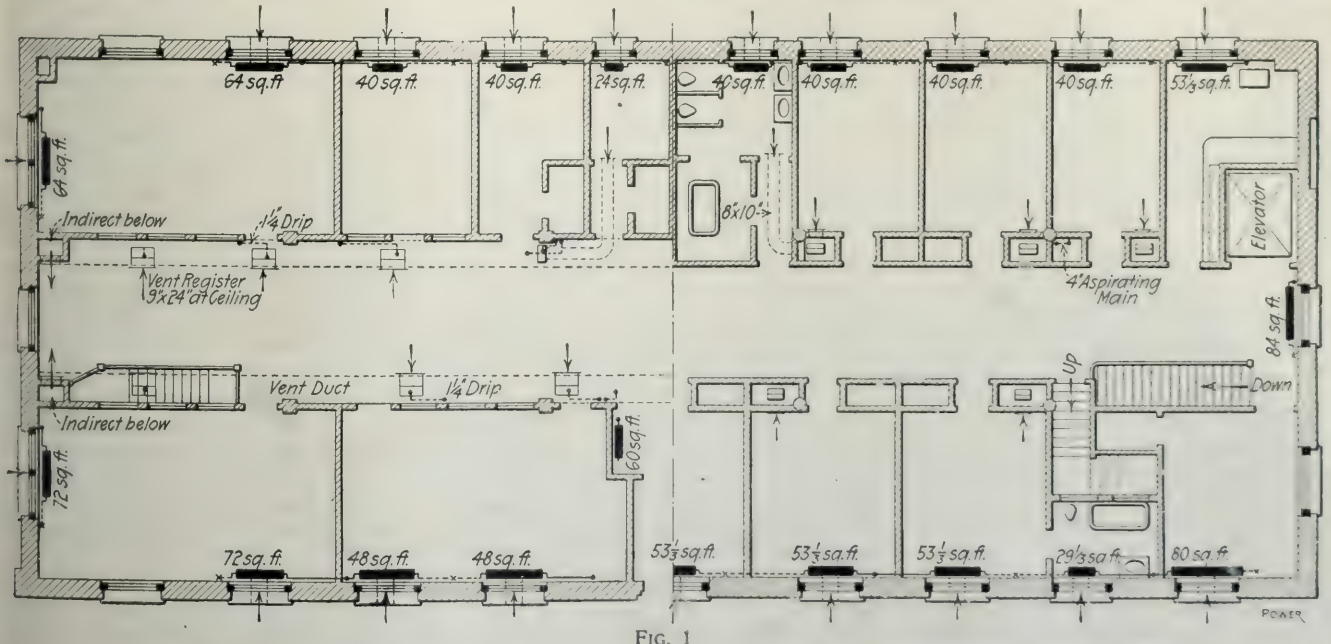


Fig. 1

Half Plan of Ground Floor

Half Plan of Future Upper Story

*Abstract of a paper read before the American Society of Heating and Ventilating Engineers at Detroit.

panel. Thus the exhausting effect of the system in the corridor induces air flow through the lower panels of the doors from the rooms which are supplied with tempered air by the direct-indirect radiators.

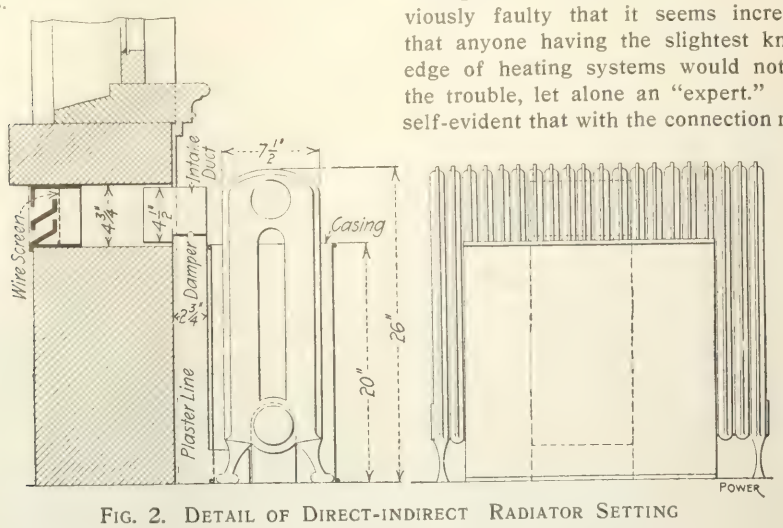


FIG. 2. DETAIL OF DIRECT-INDIRECT RADIATOR SETTING

To compensate for heat lost from the end windows of the corridor and to provide an additional supply of fresh air, indirect stacks are in the basement. These discharge air through large registers at both ends of the corridor. The ducts and registers are proportioned so that the air is heated not much above 70 deg. F.

Exhaust register boxes under the corridor ceiling pierce the concrete ceiling and discharge into a tightly constructed roof space where two 2½-in. steam mains

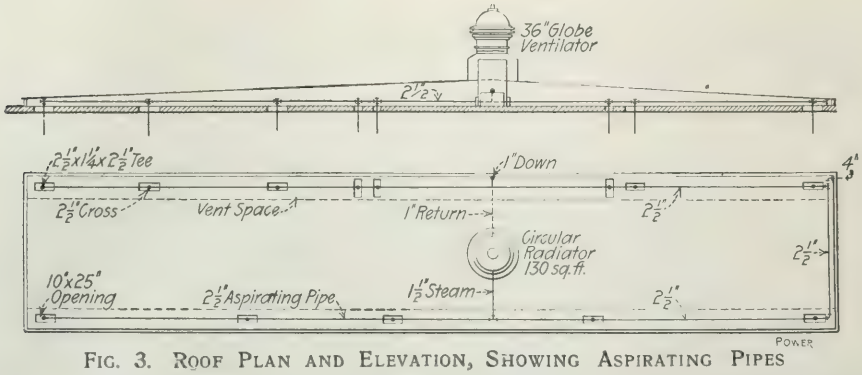


FIG. 3. ROOF PLAN AND ELEVATION, SHOWING ASPIRATING PIPES

and a 130-sq.ft. circular radiator furnish heat for aspirating purposes. Tees on the pipes allow extending aspirating risers through the future vertical ventilating flues required when the upper stories are built. A typical upper story floor is shown in Fig. 3.

No dampers were installed in the registers of the exhaust or on the indirects. But to prevent any unnecessary escape of heat after the dispensary closes in the afternoon, one main damper was installed in the ventilator on the roof, operated by a chain from the corridor. The damper is unbalanced so that it will always be open unless the chain is pulled down. The system has been in operation the past winter, and has proved a success.

CORRESPONDENCE

Expert Piping

The piping layout described by Mr. Essinger in the Aug. 20 issue is so obviously faulty that it seems incredible that anyone having the slightest knowledge of heating systems would not see the trouble, let alone an "expert." It is self-evident that with the connection made as shown the returns from the heating coils would fill the exhaust pipe above the back-pressure valve and then back up through the coils. This would cut off the escape of steam from the engine exhaust line and the back pressure would immediately rise until it was sufficient to overcome the back-pressure valve plus the head of water, when it would open, allowing the water collected above to fall into the feed-water heater and the pressure to drop until the valve closed. Then the cycle would be repeated. Obviously such a condition would lead to more or less flooding of the heating coils, to say nothing of the water-hammer due to the mixing of the water and steam when the back-pressure valve opens.

The obvious solution is to carry the return pipe from the coils to the feed-water heater. I would also advise Mr. Essinger to bleed the exhaust line just above the back-pressure valve.

W. L. DURAND.

Brooklyn, N. Y.

Mr. Essinger's system will not give satisfaction and is liable to cause serious trouble. Assume that the exhaust head is 25 ft. above the back-pressure valve and the return from the coil. If there is a check valve in the radiator return, what is to prevent the exhaust pipe above the back-pressure valve filling with water and producing

$$25 \times 0.437 = 10.925\ lb.$$

back pressure, which would mean at least 11 lb. at the engine. With an engine constant of 3 this would mean a loss of

$$11 \times 3 = 33\ hp.$$

There would, of course, be a slight increase in heat to the feed water, meaning perhaps 1 per cent. less fuel.

A more serious trouble would be the water-hammer. For a convincing argument it would only be necessary for Mr. Essinger to request the presence of the boss. Immediately after shutting down the engine, lift up the back-pressure valve with its load of water above. The solid thud of the water striking the heater plates will be instructive and may perhaps crack the heater. With the engine running and some pressure in the heater, if there is no drain above the back-pressure valve the pressure will force the water up the pipe with many a bang. I have seen an 8-in. cast-iron exhaust pipe cracked for 4 ft. of its length from this cause. Mr. Essinger's piping will undoubtedly be changed after a few trials. A better way would be to drain the coils through a trap to the heater, and a drain should also be placed in the exhaust pipe just above the back-pressure valve.

A. C. WALDRON.

Revere, Mass.

The only thing wrong with the connections shown by Mr. Essinger in the Aug. 20 issue is that steam as soon as turned on will condense in the radiators and run back through the return line into the exhaust pipe above the back-pressure valve until it fills the exhaust pipe and backs up in the radiators. The water on top of the back-pressure valve will prevent its opening until the back pressure in the exhaust pipe over-balances the water pressure. Then the engine may stop, or anyone that happens to be outside of the building will get a shower bath, and the plant will have to be shut down while the engineer makes repairs to the piping that may be broken by water-hammer.

The design is not only bad but dangerous. Before any heating is done it will be necessary to disconnect the return pipe from the exhaust pipe and connect it to a trap or vacuum pump and discharge the condensation either to the sewer or through an oil separator to the heater. If I was engineer in this plant I would rather run the chance of being discharged in showing the employer that this system is dangerous than try to run it and then be discharged as incompetent for causing serious damage to the equipment.

J. C. HAWKINS.

Hyattsville, Md.

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POWER

The Collegiate in the Power Plant

Although engineering colleges have for many years been graduating young men well equipped to enter the practice of engineering, who have proved their good judgment and ability by their works, the rank and file of power-plant operators are still more or less prejudiced against the college-trained man. This is due, in part, to at least four reasons, the treatment usually accorded to the beginner by the experienced man, misjudgment, envy, and a misunderstanding of what the college man is trying to accomplish.

Too much is often expected of the young man just out of college. During his four years of college work he has accomplished something more than simply accumulating facts. His reasoning power has been developed. He has learned to adjust himself to his environment or to change his environment if he finds it unsuitable. His ideals have been formulated, both professional and humanistic. He has been associated with his teachers who are, for the most part, also engineers of standing in their profession. He has gained from this association a rational viewpoint and he neither expects too much nor too little from his fellows. But in contrast to all this he is not a craftsman nor yet a professional man. He still lacks one prime requisite that the college cannot provide him and that is engineering judgment. This may only be acquired by meeting the tests and crises of everyday experience in his chosen work. He is likely to make the mistakes in judgment of any tyro and is worthy of the kindly criticism and patient consideration that should be given every beginner. Sadly enough, this is too seldom given either to the college man or to his fellow from the school of life. The experienced man reasons that he had a hard time getting what he now knows; therefore let the beginner dig, and especially let him blister if he is a college man. This is palpably unfair.

Human life is and always will be a great leveling agent. We do not stop to inquire, unless it be to satisfy our curiosity, whether the man who has made a success is college trained or not. To be successful he must possess some of the attributes of true greatness. Given two men of equal ability and enterprise, one college trained and the other trained only in the school of experience, the one may reach as high a professional standing as the other if they each give equal service.

One thing that apparently makes the practical man the natural enemy of any collegiate is the fact that the latter is willing to work for low wages to get a start. The man fresh from college is but a beginner and must expect beginner's wages. He is worth no more. While going through this period he is not the competitor of the experienced man. The fact that he is being paid an amount which is truly commensurate with the service he gives does not mean that wages will be lowered generally because of his action. He is merely a beginner and is receiving the consideration which is given to any beginner. The college man is not "undercutting"; he is simply paying for his experience while learning in the same manner as an apprentice. The practical man forgets that the college-trained man is not a finished product.

If any trade, craft, organization or profession is to continue to exist and to progress, it must give aid and consideration to the beginners. There is plenty for the young college man to do in the power-plant field. He cannot supplant the experienced operator unless the latter is willing to give up. Excellence demands full payment. There is no short cut to success. Whether we are college men or post graduates in the school of hard knocks, beginner or adept, let us not forget that we are all working to the same end and that we are all students in the great university of human experience.

Why a "Graduate"?

The United States Civil Service Commission announces an examination to secure eligibles from which to make certification to fill a vacancy in the position of engineer of tests at the Watertown Arsenal. An applicant must be a *graduate* in civil or mechanical engineering, of at least three years' experience, etc.

Again we protest against this graduate stipulation. The essential is, that the applicant shall know what he is required to know in order successfully to administer the position. It is nobody's business how he acquired this knowledge if he has it. There is many a civil and mechanical engineer who has got his knowledge by study outside of institutions authorized to issue degrees or certificates of graduation. They are just as adaptable to and just as useful in a position such as that under consideration, as a graduate engineer, and just as much entitled to its opportunities. It is the business of the Commission to ascertain if they have the required qualifications, and not how they got them.

Flue Gas Analysis

We have frequently called attention to the importance of flue-gas analysis, its advantages directly in increased economy of the plant, and indirectly to the engineer himself who gets the credit for bringing about improved conditions. We have predicted that gas-analyzing apparatus will become as essential a part of the boiler-room equipment as the indicator now is of the engine-room equipment.

When flue-gas analysis first began to attract notice, many regarded it as a means for the chemist to show off his knowledge, but nothing that would ever be of any practical worth, nor half so dependable as the experience of the trained fireman. Gradually the scoffers discovered that the analysis had real value and as the apparatus became somewhat simplified they found it was not so impractical as it at first appeared.

To one not familiar with it, it looks very complicated, and we have been glad to help disabuse those who regarded it so, with the series on "Flue Gas Analysis for Beginners," concluded in this issue.

Those who may have filed the article should correct two typographical errors that appeared in the first installment. On page 331 in the first column, next to last paragraph, pipette No. 2 should have been No. 3. In the next column, last paragraph, the reagent for the first pipette was for determining CO_2 and not CO.

Graphite in Boilers

There have been too many cases of the successful use of graphite for the prevention and removal of scale, and the avoidance of pitting in boilers, to leave its usefulness for this purpose longer open to debate. There may be differences of opinion as to why it does it, and how it does it, but in numerous cases the feeding of graphite into a boiler has not only prevented the formation of new scale, but has removed large quantities of scale already accumulated.

It is doubtless true also that in some cases graphite has been tried without success. It may be that graphite, like most other things, has its limitations, that it is very good in some cases and not in others. It may be due also to the kind of graphite used, and how it was applied.

The action is probably a mechanical one. The presence of the graphite in the scale destroys the adhesion of the particles which would otherwise weld themselves solidly together, so that the scale either does not form, or is easily broken up by the disturbances incidental to the ordinary operation of the boiler. But it is essential that the right kind of graphite should be used. One is apt to think that, while nothing but the most refined product is fit for a bearing, anything will do for so crude a process as mixing with boiler scale to make it brittle. The crudest product of the graphite companies, such as is sold to foundries for facing molds, etc., is therefore used, with the consequence that a lot of the worst kind of scale-producing material is deliberately pumped into the boiler, and not enough real graphite to counteract its effect, combined with that of the scale-forming material already present.

It is no proof of the adaptability of a graphite to the purpose in hand that it is high in carbon. Coal is carbon, so is sugar, but they would not do. The graphite must be not only pure, free from sand and other foreign substances, but it must have the lubricating and preservative properties upon which its action in the boiler evidently depends.

We shall be glad to have accounts of *bona fide* instances of the use or attempted use of graphite in boilers, and the results obtained, with enough of the conditions and attendant circumstances to indicate to others who desire to try the experiment the best way to proceed.

Four Pass Boilers

The advantage of using four instead of three vertical passes with horizontal water-tube boilers is being seriously discussed. With such a construction it is advisable to make the tubes twenty feet long in order not to restrict the passages for the gases. It can be easily seen that under certain conditions this type has several advantages over the three pass. In the first place, it has been demonstrated that the temperature of the escaping gases is from twenty-five to fifty degrees Fahrenheit lower, which, of course, means that less heat goes up the stack.

Probably the greatest objection to the four-pass boiler is its increased draft requirements, which must be taken care of by increasing the height of the stack or by some other means. This increase, however, would at the most not be greater than the draft necessary for a three-pass boiler with an economizer; in fact, it is doubtful if it would be as great, whereas the increased efficiency due to the lower stack temperature would more than offset it. Other objections are one more baffle to keep tight and a longer boiler setting and longer tubes to contend with. On the other hand, no breeching is required, as the gases can be brought down through the last pass into a flue underneath the floor and thus out to the stack which arrangement gives a clear space at the side of the stack for the breeching connection. As for the extra baffle, it is very little more trouble to look after it when tending to the other two, and blowing off the longer tubes is a small item. To prevent an increased tendency for air infiltration because of the longer setting, a covering can be applied which in itself is an economical investment no matter how long the setting.

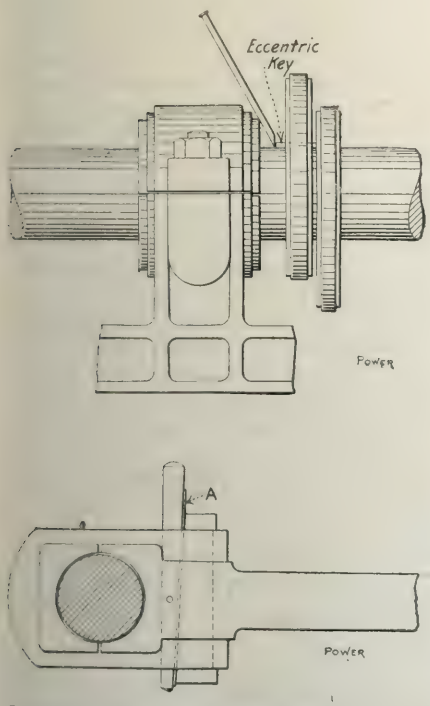
These objections, therefore, are not very important except, perhaps, that of greater draft, but when they are all put into the balance with the fuel saving resulting from the extraction of from twenty-five to fifty degrees more heat from the gases it will no doubt be found that they are greatly outweighed.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Key and Gib Kinks

While employed aboard a high-speed passenger vessel, I saw an engineer use a very ingenious way of overcoming a loose eccentric key. After driving the key back to its proper position, a portion of the shaft in front of the key was projected by means of a chisel bar and sledge, thus holding the key from working out and also making it fit the keyway closer. While it was necessary to stop the engine about five minutes, it saved much time, for, as shown by the illustration, to remove the key the bearing cap would have to be removed. After reaching port the shaft was hammered down and smoothed and a new key fitted.



SHOWING ECCENTRIC KEY AND POSITION OF LINER FOR ADJUSTMENT OF BRASSES

Another kink of probable interest to engineers who have gib and key connections about the plant, is the following manner of keying them up: Find the amount of clearance to give the bearing, then cut a slip of paper brass the thickness of the clearance and the same shape as the key. For example, it is desired to give the bearing 0.008 in. clearance. Cut the slip from 0.008-in. paper brass, place it on top of the gib between it and the key, as at *A*, Fig. 2, then drive the key up solid. Mark the key and drive it out enough to remove the slip and then drive it up to the mark and the bearing will have 0.008 clearance.

HENRY C. BRIGGS.

Jersey City, N. J.

Emergency Repair to Air Compressor

In a New England manufactory using considerable compressed air in operating its machinery, an accident occurred recently which, for the moment, indicated a protracted shutdown of some parts of the plant.

A split tube in the intercooler of a 12 and 20 by 8-in. cross-compound belt-driven air compressor flooded the high-pressure side of the machine and resulted in splitting the cylinder from end to end entirely through both the cylinder and jacket walls. The only other compressor in the plant, a 10x10 duplex belt-driven machine, set up adjacent to the cross-compound, was incapable of supplying the demand even when operated at its highest speed.

When a shutdown of some departments seemed inevitable, the trouble was reported to the manager. Evidently he had had a similar experience, for he instantly issued orders to take out the broken high-pressure cylinder on the cross-compound machine, disconnect the rods and connect up the outlet of the intercooler on this machine to the suction opening on the small duplex machine. From the position of the machines in the engine room, this connection could be made very readily.

The alterations were completed in a few hours. The unloading device on the suction of the cross-compound, low-pressure cylinder throttled down and adjusted to maintain 25 lb. air pressure in the intercooler. and the combination unit started. The results were entirely satisfactory. With some slight adjustment of the pressure regulator on the duplex compressor, it was found that when supplied with air at an initial pressure at 25 lb., the machine delivered all the air required for the plant's operation.

This makeshift arrangement kept the plant in operation until a new high-pressure cylinder was procured for the cross-compound compressor. For several days after the cylinder had been received the air supply was sufficiently satisfactory to warrant the continued use of the combination compressor until a good opportunity presented itself to install the new cylinder.

A comparison of the indicator diagrams taken from the small compressor under the ordinary operating conditions with those taken when supplied with the air at 25 lb. from the low-pressure cylinder of the cross-compound show that the volume of air delivered by this compressor increased very rapidly with the increased pressure on the suction.

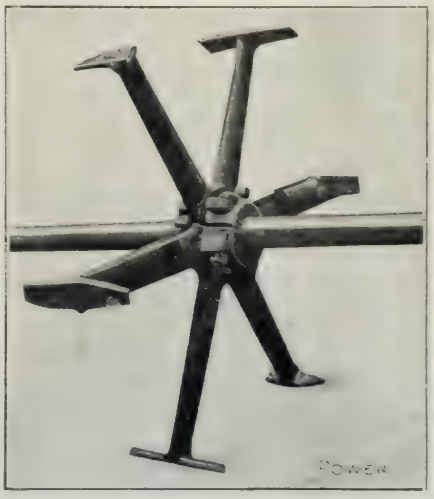
W. M. FLEMING.

Holyoke, Mass.

A Fractured Pulley

The illustration shows the result of a curious and interesting mishap to a pulley, due to a belt slipping off the driven pulley and remaining on the driver.

The fractured pulley was on a countershaft about 6 ft. long and 1⅛ in. in diameter. There was nothing to restrain the free motion of the countershaft as the machine it operated was located on the floor above and driven by a binder pulley. The machine was not running and the operating belt hung loosely, so that the countershaft was free to revolve without hindrance. The countershaft, driven by a 4-in. double belt, was 20 ft. from the main or driving shaft.



SHOWING EFFECT OF CENTRIFUGAL FORCE ON PULLEY

The belt was long, soft and flexible, and of a special tannage, and when it fell on the shaft, clung reasonably well. The driver was 24 in. in diameter and made 250 r.p.m., and as the receiving pulley was likewise 24 in. in diameter, the normal speed of the countershaft was 245 r.p.m., allowing 2 per cent. for slip.

When the belt was down on the shaft, however, which was slightly less than 2

in. in diameter, the acceleration was 12 to 1, or might possibly have reached 3000 r.p.m., the slippage of course greatly reducing this result, as the belt was loose. The speed increased sufficiently, nevertheless, to fracture the pulley through the centrifugal force developed, a peripherial speed of probably 8000 to 10,000 ft. per min. having been reached before rupture.

The fragments scattered in many directions, one piece cutting into a 2-in. plank to a depth of 1½ in. as cleanly as if it had been cut with an axe. Another travelled 75 ft. across the ceiling, penetrated a window, cleared a roadway 22 ft. wide and struck a locker in an adjoining building. A third passed through a lunchbox hanging on a post 50 ft. away.

The entire incident might have been avoided by the use of an ordinary belt iron or hanger, and while it is our practice to take every precaution, this was one of the numerous locations where no provision had been made.

H. E. Osgood.

Wilmington, Del.

Emptying an Oil Barrel

To empty an oil barrel is more or less a troublesome job. We draw our daily supply from three storage tanks for use on gas engines and a supply of oil has also to be placed in the engine crank cases from time to time, to replace that used or drawn off when the machines are cleaned out.

The arrangement shown in Fig. 2 has more than paid for itself; it spills no oil on the floor and eliminates all the strong-arm work of lifting the barrel. It consists of a gooseneck *A* made of ¾-in. brass pipe; the short end acts as a spout and can be inserted into the filling hole in the tank or run into the crank case.

The long end will reach the bottom of the bulge in a barrel when inserted through the bunghole and takes out nearly every drop of oil. A 1x¾x1-in. tee *B* can slide up and down on this leg, making the device adjustable for use in drawing oil out of receptacles having a hole big enough to admit the pipe. A ¾-in. short nipple *C* at the lower end of the tee can be fitted into the barrel bunghole, the thread on the nipple making a joint usually tight enough for the purpose. If the bunghole is large, some waste will make this joint tight. The upper 1-in. outlet is fitted with a reducer *D* and babbitted to fit the brass tube. This serves as a gland, as a little oil makes it slip easily on the tube and at the same time acts as packing. The 1-in. side outlet is fitted with a reducer *E* into which a brass tube *F* about 3 in. long is screwed, the outer end being screwed into a brass bushing whose outside is threaded to fit a ½-in. hose coupling *G*.

A small electrically driven air compressor *H* was available which supplied compressed air for starting the engines. A globe valve and a tee were inserted in the line leading to the air receivers, Fig. 1. From the outlet of the tee a line was run over to the oil storage tanks where it connected with a ½-in. hose about 8 ft. long. A globe valve controls the admission of air to this line in which a small pop safety valve *J* with a light spring limits the pressure to about 7 lb.

When a barrel is to be emptied into one of the tanks it is rolled in on the floor and blocked. The gooseneck is inserted through the bunghole and the ¾-in. nipple screwed in to hold it; the brass pipe reaches to the bottom of the barrel and over the tanks *K*, *L* and *M*. The

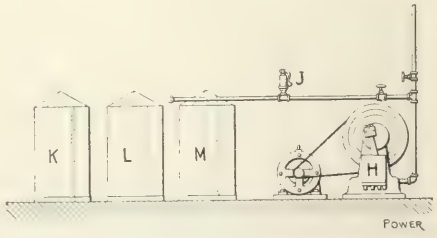


Fig. 1. Arrangement of Piping

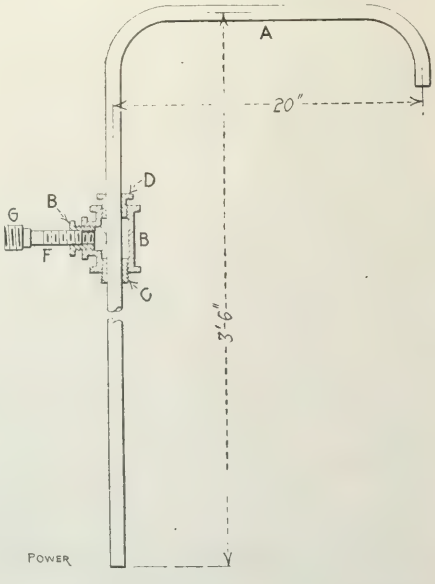


Fig. 2. Details of Air-oil Lift Nozzle

globe valve controlling this line is opened and the compressor is started pumping air into the barrel while the oil flows into the tank. The device requires no further attention until the barrel has been emptied, which is known by the air coming out of the spout.

The safety valve with the 7-lb. spring is an important accessory. Some oils are not as fluid as others, and the carrying capacity of the ¾-in. brass gooseneck is limited. With a heavy oil the air pressure may rise rapidly; the safety valve limits the rise to a pressure the barrel can safely withstand and one sufficient to overcome the hydrostatic head of the oil. The omission of the safety valve may give the installer of such a device a very "messy" lesson in elementary hydrostatics, the partial loss of the oil and the total loss of the barrel.

E. Ellsworth.

Cleveland, Ohio.

Specifications for Dutch Ovens

Will some fellow-engineer state in Power a desirable method of building an ignition arch or dutch oven in connection with return-tubular boilers 60 in. by 18 ft., using oil as a fuel?

Possibly some reader with experience has the patterns for a suitable framework for supporting the tiling, and will give the desired information as to the size and shape for the best results.

F. B. Daggett.

St. Joseph, Mo.

Loose Valve Stops Pump

Recently I stopped our feed pump to test an injector I had set up. After trying out the injector to my own satisfaction, I told the fireman to start the pump. A few minutes later he reported that it would not start. This surprised me as it had run well all the morning. I tried it myself and it made a few strokes and stopped. Thinking the valves had been displaced in some way, I removed the steam-chest cover, but found them all right. I then opened the water end, thinking one of the follower plates might have become loose and jammed, but found the piston and valves in order.

I then tried to move the rods with a bar, and they moved easily. I put everything together again, removed the bonnet of the exhaust valve and, as I suspected, the disk had come loose and fallen off the stem square on its seat. The pressure on the top had closed the valve tightly. I screwed the disk-holder back on the stem, put the valve together, turned on the steam and the pump then ran without further trouble.

E. C. Garlick.

Milwaukee, Wis.

Knocks in Air Compressors

We have a duplex two-stage air compressor in operation nine hours a day and seven days a week. Some Sundays the main part of the power plant is shut down and the engineer goes to church, leaving the compressor and pumps in care of the oiler.

One Monday morning before starting up, the oiler reported that the compressor began knocking about Sunday noon and got steadily worse toward the close of the day. After starting the main engine, the engineer went to the compressor and opened the throttle, thinking to turn it over slowly and locate the trouble. It made two or three turns, when the crank

disk broke through the pin-bore. An examination showed that the piston rod, working loose, had unscrewed about ⅛ in. out of the crosshead, taking up all the clearance on the head end and the air piston, striking the head, caused the fracture through the crank disk.

Everything was stripped on that side of the compressor, but aside from the broken crank and a bent connecting-rod no further damage was done. A 1¾x3½-in. steel band was forged and bored $\frac{1}{16}$ in. less in diameter than the 26x3½-in. disk and shrunk on. The compressor has been in operation two months since the accident and the crank is as rigid and runs as well as ever.

Another accident happened to this same compressor that may be of interest. A new engineer, not familiar with compressors, had charge of the machine. One morning in starting up he forgot to open the air-discharge valve on the high-pressure side. The machine turned over two or three times, but had to be shut down. Finding the outlet closed, he opened it and started once more. Everything seemed to go all right until the pressure got up to about 50 lb., when there came a clear metallic knock on the high-pressure side as the crank was on the low quarter or the low-pressure crank was at the crank end. The sound seemed to be in the pillow block or crosshead. The quarter boxes were taken up and the crosshead shoes adjusted, the crank brasses examined and even the wristpin taken out and refitted, but still the knock remained.

As everything else had been examined, the cylinders seemed the only remaining points of attack. The steam cylinder showed everything to be all right, and it was not until the air piston was taken out that we found the trouble. The piston jam nut had become loose so that the piston had about $\frac{1}{16}$ in. play on the rod. When the low-pressure piston reversed its direction from the crank end, the fresh charge of air into the high-pressure cylinder would cause the piston to jump back against the nut, causing the knock. On the other stroke the tapered end of the rod would take up the jar, thus showing the knock at one point only. After the nut was taken up and everything put together again the machine ran smoothly.

A 48x24-in. slide-valve engine had a 15x48-in. air compressor built tandem to the end of the cylinder. The compressor piston rod was screwed into the elongated steam piston nut and locked with a setscrew. For some days there had been a slight knock which was getting gradually worse, especially when a heavy load was thrown on. The air and steam cylinder heads were taken off and the air piston looked as if it may have touched the head, but, although the rod was slightly loose in the nut, check marks indicated it had not turned. This was also shown by the clearance of ⅛ in., which was correct.

The heads were put on and slight adjustments made to the crank and crosshead and the machine put to work again. The knock was there as before. After a couple of days the steam-cylinder head was again taken off to make sure that the main piston nut was set up tight, which it proved to be. It was found, though, that the air piston rod was slightly loose in the threads again, but had not turned appreciably from the mark. This gave a clue to the cause of the trouble.

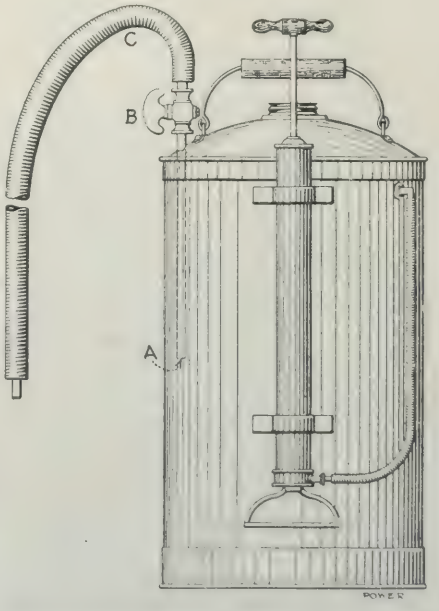
The engine was belted to a direct-current generator situated on the side opposite the crank. The heavier strain on the main bearing was on this side. There had been a slight wobble to the crankshaft for some time, but the engine ran so well that no attention was paid to it until it was proved that this side-play had increased sufficiently to take up the clearance in the air cylinder when heavily loaded. A half turn on the inside quarter-box adjusting screws stopped the noise. Closer adjustment of this box made the main bearing run warm.

T. H. HEATH.

Seattle, Wash.

Oil Can for Lifting Waste Oil

The illustration shows a can for taking up oil from places difficult of access in the usual way. The can is the ordinary galvanized, 5-gal. size and the pump is



VACUUM CAN FOR LIFTING OIL FROM INACCESSIBLE PLACES

of the bicycle variety with a leather plunger. A metal valve stem is soldered on the bottom end in place of the hose connection found on the pump. To the valve stem or check valve, is attached a small rubber hose, the other end of which is connected to the top of the can. A small pipe *A* is soldered on the top of the can, to which is attached a small petcock *B* and a piece of rubber hose *C*.

Gaskets should be placed under the caps on top of the can to make them airtight. A few strokes of the pump removes the air from the can, whereupon, by placing the end of hose in the oil to be canned, and opening the cock, the oil flows into the can.

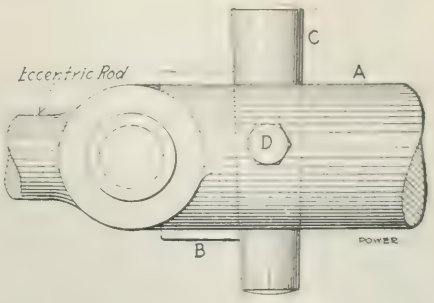
O. O. LEMON.

Bristol, Va.

Pump Detail Improvement

A portable engine turned out in large quantities has a ram-type single-acting boiler-feed pump, driven by an eccentric on the crankshaft. The ram is connected with the eccentric rod by a parallel pin secured by a washer and split pin. The ram end is forked and the eccentric rod end fits between the faces of the fork.

The pin has proved troublesome in use; every six months or thereabouts a knock developed, and to remove it the holes had



KEY AND CROSS FOR ADJUSTING WORN PIN

to be reamed and a new pin fitted. It was undesirable to enlarge the ram and shorten its stroke so as to admit of the use of a large pin. Even if this were done the trouble would still occur, though not as soon.

Means of adjustment for wear became necessary and the illustration shows how it was accomplished. The standard ram *A* was long enough outside the stuffing-box so that the existing slot had simply to be deepened and a brass block *B*, its adjusting key *C* and capscrew *D* fitted. The adjusting may be easily understood from the illustration.

Since fitting the pumps as described the manufacturers have received several letters of thanks from customers, and many of their earlier engines have been altered locally in this particular. Of course, the pump is only a minor detail, but it is careful attention to detail which frequently lands a repeat order and always satisfies customers.

A. L. HAAS.

London, England.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

High Speed Indicating

In the discussion of Mr. Clayton's paper before the A. S. M. E., reported in POWER, Aug. 13, Arthur L. Rice spoke of the difficulty of taking diagrams at high speeds. What does he call "high speed?" I indicated some of the fastest locomotives without difficulty, at over 300 r.p.m. almost daily for a long time; I have also got good diagrams at 720 r.p.m. in testing lubricants for the Standard Oil Co., and at least 25 years ago at a meeting of the Franklin Institute, got clean, practically useful diagrams at 1120 r.p.m., using a Crosby indicator which I had altered for railway work. I also "hooked on" at that speed. It was then the record, but I suppose has long ago been beaten.

ROBERT GRIMSHAW.

Dresden, Germany.

Shorter Cutoff—Greater Load

The article in POWER of Aug. 6, under the above heading, while being an explanation of the matter, will not, in my opinion, fill the bill for the engineers who are immediately concerned in this argument. In the first place, those who cannot see why shortening the cutoff on the low-pressure cylinder of a compound engine will increase its load, are not very apt to get a clear understanding of the subject from a combined indicator diagram, which in itself is apt to be a greater puzzle to them than the question of the cutoff itself.

No doubt many engineers are familiar with a combined indicator diagram and all that it stands for, but a greater number are not, and on them Mr. Low's explanation is entirely lost. The little editorial under the same heading is more likely to bring a ray of light on the subject to the engineer who does not understand it. It is rather hard to explain, and I would suggest to the subscriber who asked for the explanation (supposing him to be operating a compound engine) that he take some of his friends who are against him in the argument into his engine room and try to demonstrate to them that he is right (which he certainly is), and then analyze the reasons why it is so. Let him take a set of diagrams and get the area of them with a certain receiver pressure and figure them out before all hands, so that everybody is satisfied they are correct; then increase his receiver pressure three pounds by shortening the cutoff on the low, and then ask each one whether the load on the low-pressure will be more or less on the next diagram and then proceed to take another set of cards and show them just what the difference will be.

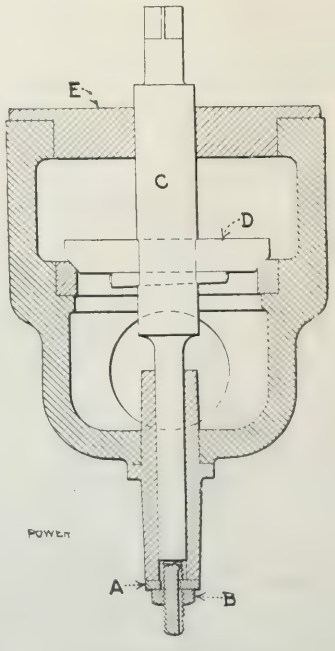
I think if he does this he will be able to show them so that they will understand the matter. It is much easier to understand a thing when you can see it done.

WILLIAM N. WING.

Brooklyn, N. Y.

Valve Seat Reamer

The valve-seat reamer shown by Mr. Benefiel, in the July 16 issue, is a good tool. We have several similar reamers for use on the valve seats of our high-pressure pumps, where the valve stems do not project through the bottom. These reamers have no bottom guides. For a gas-engine valve-seat reamer we have Mr. Benefiel's reamer "beat to a frazzle."



VALVE-SEAT REAMER

The illustration shows the advantage of our reamer over the other for use on a valve seat where the valve stem has a bottom guide. The sliding collar *A* and nut *B* on the end of the cutter spindle *C* provide means for feeding the cutter down on the seat, and prevent the cutter *D* from chattering. The spindle guide is shown at *E*.

We ream the seats lightly every time the valves are ground, and find this makes it easier to remove the hard scale that forms, and saves a lot of time in grinding.

H. H. DELBERT.

Titusville, Penn.

Mr. Greene's Noisy Heater

On page 883 of the June 18 issue, Mr. Greene states that my suggestion as to the cure for his noisy heater did not work when put to the test. I am still of the opinion, however, that, if he will make the seal deep enough, the noise will stop. The supply pipe should be run down on the outside of the heater and entered at a point on a level with the exhaust-steam inlet. This way of supplying the cold water would work best, as if steam is formed in the pipe inside the heater, it will rise to the top without interference and be replaced by water from below, no condensation taking place, no vacuum and consequently no noise.

W. T. MEINZER.

Brooklyn, N. Y.

Load on Stay-bolts

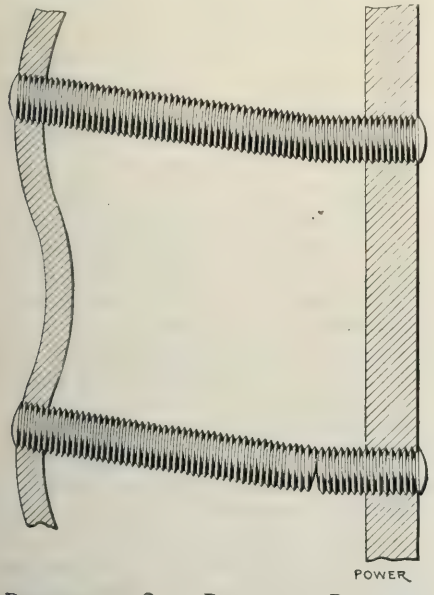
In a recent article the methods of calculating the load on a stay-bolt was given, and the result compared with the tensile strength of stay-bolt iron gave a factor of safety of eight. This being higher than the factor of safety used in some other parts of the boiler, especially the longitudinal seams, why is it used?

The factor of safety might be reduced in several ways, each open to objections. It may be reduced to that of the seams, by reducing the diameter of the bolts, but this will reduce the size of the tap, which is likely to increase the breakage of taps when motor driven, as both sheets are tapped at once with a tap long enough to reach through so that the continuous threads on the bolt will mate in both sheets without strain. Perhaps the most important point is the added strain thrown on neighboring bolts when one breaks. The pitch may be increased, but here the bulging resistance of an unstayed flat sheet is involved and the added strain on bolts near a broken one is objectionable.

Even with a factor of safety of eight stay-bolts break for several reasons. In practice, with the fire sheet about half

as thick as the wrapper sheet, the bolts almost invariably break close to the wrapper sheet and seldom against the fire sheet, and far more broken bolts are found high up toward the crown sheet than near the mud ring.

The wrapper sheet is subjected to a temperature of, say, 370 deg. F. The outer temperature of the lagging being less, the mean temperature of the plate is below 370 deg. The fire sheet is exposed to the great heat of the fire on one side and its temperature may be greatly increased if the water side is covered by scale. The expansion of the fire sheet is, therefore, considerably greater than that of the wrapper, and this relative movement of the sheets bends the bolts approximately ⅛ in. in fair sized boilers. The thicker the scale the hotter the fire sheet gets, greater expansion occurs, and



Bending of Stay Bolts and Buckling of the Sheet

the bolts are more severely bent. Both sheets are firmly held by the mud ring. The lower bolts are subjected to but little bending strain, the upper ones suffering the maximum distortion, which is why the maximum breakage is at the top, notwithstanding the greater length of bolts as the water leg widens toward the crown sheet.

As the back head is stiffer than the flue sheet, any horizontal expansion travels forward, causing greater breakage of bolts toward the front end of any horizontal row than at the back, since the movement of the sheet at that point is both upward and horizontal, a bolt being bent along the diagonal of these two movements. The amount of this bending depends upon the stiffness of various parts, such as the flue sheet, and its ability to bend between the sheets on the one hand and the expanding flues on the other. Breakage is a maximum at the upper and forward corners of the fire box.

The stiffness of a row of bolts tends to cause a slight bend or wave in the sheets to which they are attached, as shown in the illustration. If each sheet remained vertical, then the two ends of the bolts would be horizontal, but with the fire end above the outside end. This requires a bend at each sheet. The outer sheet being comparatively thick, retains practically its flat form, but the thin fire sheet takes part of the bend from the bolts. The total distortion of the bolts at the wrapper being greater than at the inside sheet causes more frequent breaking at that point than at the fire sheet, which is evident when the alternate heating and cooling of boilers in irregular service is considered.

This repeated bending, when combined with the moderate tensile stress, being the cause of practically all of the fractures found, has given rise to the vibration test for stay-bolt iron, in which a threaded bolt, while under tension, is subjected to thousands or hundreds of thousands of slight bends, the comparative number of bends or distortions representing the relative values of different irons. It has also given rise to the flexible stay-bolt, with a ball joint at the outer sheet, so that bending at this point is avoided.

E. G. Medart.

Quincy, Ill.

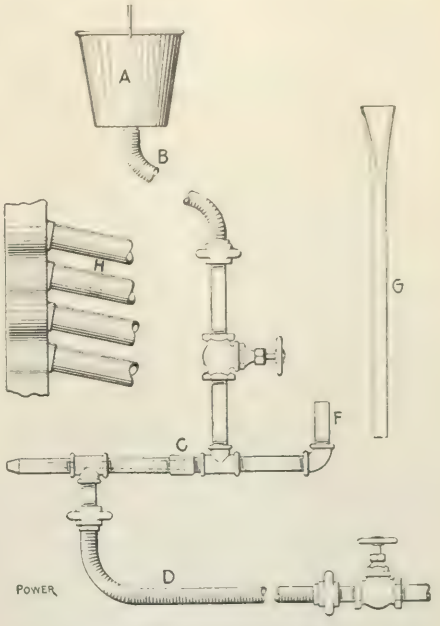
Removing a Tube Header

In the June 18 issue, A. M. Quinn gives instructions for removing a broken header from a water-tube boiler. The illustration shows a burner made of ½-in. pipe for removing a header or tube without doing any drilling or cutting.

To operate, fill the pail *A* with fuel oil and hang by a hook or other means to the crow-foot brace on the manhole. The hose *B* should be long enough to bring the burner to the opening in the header where the tube is to be heated. Place a piece of broken brick inside the tube and about 5 or 6 in. from the point where it is expanded into the header, as at *H*. Next place a piece of oily waste next to the brick in the tube. Turn on and ignite the oil from the bucket as soon as it issues from the burner nozzle *C*. Now ignite the waste and turn on the air or steam, which enters at *D*. An auxiliary air inlet is shown at *F*. If the burner is properly made and adjusted, the tube will be heated hot enough in 10 min. to be pried loose from the header with a pinch bar *G* made for the purpose. The edge of the outer hole in the header serves as a fulcrum for the bar. If the tubes removed are to be put back in the boiler, the ends require no treatment other than restoring their roundness.

The process causes no injury to the headers or tubes unless the flame is allowed to burn the metal, which can be avoided by proper care. The burner beats the hammer and chisel for this work as it is quicker and it may be also made very cheaply. It prevents the willing but careless or incompetent man from cutting the header to pieces while trying to remove a tube.

I suggested this method to the boss boiler maker of a large steel plant who



Oil Burner Outfit for Removing Boiler Tubes or Header

had the tubes to remove in 10 boilers. Out of the lot only two tubes were injured enough to prevent their being used again.

John F. Nagle.

Cos Cob, Conn.

Comment on a Study Question

Referring to the answer to Study Questions No. 31 in the July 2 issue, permit me to correct the statement that a direct solution cannot be obtained, as there are only four equations and five unknown quantities so that the solution must be obtained by trial.

The direct solution is obtained by subtracting 2 from the least common multiple of the divisors. For example, let x = the total number of lamps. Then x = L.C.M. of $(3, 4, 5, 6) - 2$ $\therefore x = 60 - 2 = 58$. Suppose he had lamps enough so that he could divide them in three rows and have one left over, in four rows and have two left over, in five rows and have three left over, in six rows and four left over, in seven rows and five left over, in eight rows and six left over, in nine rows and seven left over, in ten rows and eight left over, in eleven rows and nine left over. To find the number of lamps.

L.C.M. of $(3, 4, 5, 6, 7, 8, 9, 10, 11) - 2 = 27{,}718$

Proof

$$\frac{27,718}{3} = 9239 + 1$$

$$\frac{27,718}{4} = 6929 - 2$$

$$\frac{27,718}{5} = 5543 - 3$$

$$\frac{27,718}{6} = 4619 + 4$$

$$\frac{27,718}{7} = 3959 + 5$$

$$\frac{27,718}{8} = 3464 + 6$$

$$\frac{27,718}{9} = 3079 + 7$$

$$\frac{27,718}{10} = 2771 + 8$$

$$\frac{27,718}{11} = 2519 + 9$$

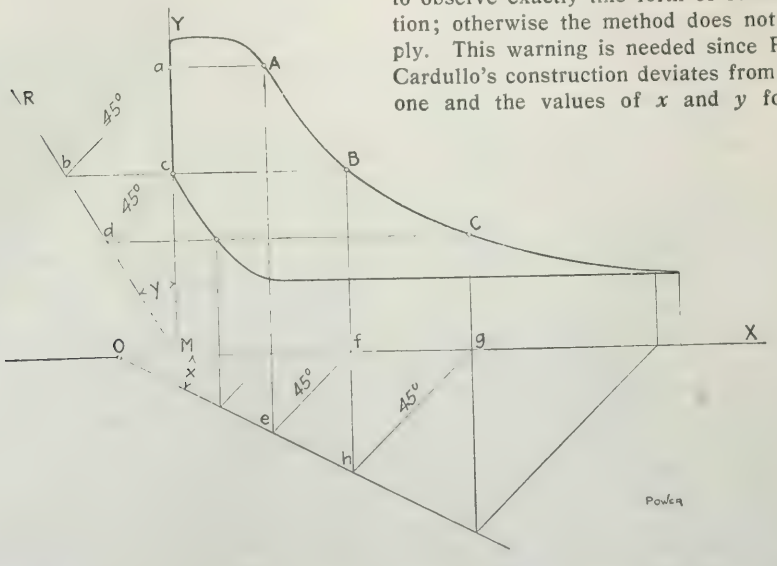
JAMES MCCLURE.

Fletcher, N. C.

Determining Clearance from Indicator Diagrams

The method of determining the clearance graphically from the indicator diagram when the curve is assumed to follow the law pv^n = constant, given on page 114 of the July 23 issue, is ingenious and useful. I developed a construction similar to Professor Cardullo's several years ago, and extended it to finding the value of n as well as the clearance.

The construction is shown in the diagram, where AC is a curve, assumed to follow the law PV^n = constant, and OX the line of zero pressure. Not knowing



the zero volume axis, draw any vertical line as MY, here drawn tangent to the indicator diagram. Lay off a construction line MR at any angle y to MY, preferably making y = 30 deg. Starting with any point A on the curve, draw the lines Aa, ab, bB, as indicated and thus determine a second point B on the curve. Repeat the same construction starting at B, using the lines Bc, cd, dC, and thus determine a third point C on the curve. Next draw vertical lines through A, B, C, and by a similar construction on the volume axis determine the points e and h. Join e and h and produce to O, which is the volume origin, and OM, the clearance desired on the indicator diagram.

This construction can be made easily and quickly and is theoretically exact. No matter what the starting point or the value of the auxiliary angle y, the origin O will always come at the same point, if the curve actually agrees with the assumed law PV^n = constant. This construction determines not only the clearance volume, but the value of n in the equation of the curve. The exact value of n is found from a rather complicated relation between angles x, y and n. A close approximation can be made by the simple relation $n = y \div x$, and as y was taken as 30 deg. to start with, it remains only to measure angle x when n can be found by division. The exponent n may be equal to, or greater or less than unity, depending on the value of x.

The theoretical error involved in the approximate determination of n is small. The approximate method becomes exact when $n = 1$. For all values of n lying between $n = 0.8$ and 1.4, the error in n is less than 1 per cent., if the angles x and y do not exceed about 40 deg. In using this construction it was recommended to use y as 30 deg., simply because this was an easy angle to draw with instruments and large enough to be measured accurately.

In finding n in this way it is necessary to observe exactly this form of construction; otherwise the method does not apply. This warning is needed since Prof. Cardullo's construction deviates from this one and the values of x and y found from his figure cannot be used in the simple way that is described here. Prof. Cardullo's method is correct for finding the clearance volume, but does not admit of using the above simple determination of n.

EDWIN H. LOCKWOOD.

New Haven, Conn.

Duchesne's Experiments on Superheat

An interesting investigation was described in the July 23 issue, under the above title, which pointed out that the indications of the mercurial thermometer, when used to determine the temperature of superheated steam, are unreliable. It is not the purpose here to comment upon the results of the investigation in so far as they affect our quantitative knowledge of superheated steam, but to call attention to the relation of these results to the use of the throttling calorimeter. Anyone who has handled this instrument to any extent will recall that the thermometer on the low-pressure side occasionally is erratic and the location of this thermometer in the steam's path seems to control its indications. This phenomenon likely is due to the two causes noted by Monsieur Duchesne to explain the discrepancies of his thermometers; namely, the low heat conductivity of superheated steam and continguous layers or strata of steam of very different temperatures. The latter cause would not be so pronounced in the restricted space of a throttling calorimeter as in a large steam pipe; nevertheless, the possibility remains, as does the need for an adequate explanation of the effect of location upon the calorimeter thermometer.

When operating from 100 lb. abs., the superheat shown by the throttling calorimeter is between zero and 115 deg. F., at a pressure between 15 and 20 lb. Duchesne's experiments do not extend to pressures as low as these, his limit being about 25 lb., under which condition an error of as much as 85 deg. C., or 153 deg. F., in the mercurial thermometer was noted, presumably at a superheat of about 300 deg. F. He says the error is augmented at the higher superheats, but, on the other hand, is higher at the lower pressures. Accordingly it may be inferred that the mercurial thermometer is untrustworthy when used to measure the superheat in a throttling calorimeter, especially if a thermometer cup is used instead of plunging the instrument directly into the stream. This accords with some observations of the writer in the use of the instrument, the conclusion being that it is better to omit the cup and use a perforated cork or wooden plug through which to pass the thermometer into the calorimeter chamber.

It requires an error of as much as 18 deg. F. in this measurement, however, to affect the result in the quality of the steam as much as 1 per cent. But it appears from Duchesne's experiments that there may be this much error. With such a possibility the refinements of stem corrections and water equivalents seem futile.

JULIAN C. SMALLWOOD.

Syracuse, N. Y.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Clearing Storage Batteries

How can the accumulation of sediment on the bottom of a storage-battery cell be prevented from short-circuiting the plates, without lifting out the plates?

C. H. B.

The sediment falls from the sides of the plates and is deposited in ridges. When short-circuiting first becomes noticeable there is usually sufficient depth below the plates for leveling the ridges down so they will clear the plates. This is most easily done by using a rod with a right-angled foot at one end long enough to reach a little more than half way across the cell. Such a leveling hook should be entirely wood and can readily be made of white pine about ¾ in. square having the angle of the foot formed with a half lap joint held by pegs of wood.

Addition of Heat to Dry Steam

If dry steam at the pressure and temperature of saturation were inclosed in a tight vessel and superheat applied, would the pressure thereby be increased?

W. L. S.

Yes, raising the temperature of dry steam will cause it to expand if it is not confined, and if the volume is maintained constant it will cause the pressure to increase.

Pounding After Resetting Valves

After setting the valves of an engine to secure a desired form of indicator diagram, a pounding set up in the crosshead pin and crankpin which could not be removed without restoring the original valve setting. What can be done to get good diagrams without the pounding?

B. N.

Undoubtedly the crankpin and crosshead pin are worn out of round into forms peculiar to the former load, point of cutoff and compression. Changing the cutoff or compression to different points of the stroke causes distribution of pressure on the partly worn pins to be made to new places which results in pounding caused in the same manner as is often the case for a change of load. Pounding usually may be obviated for any particular load by taking up the brasses and making the desired valve adjustments gradually. By continuing the operations only as fast as the pounding can be removed, the desired adjustment of valves should be secured after a few days running unlss the pins are so worn as to require re-turning.

Designing Girth Seams

In designing the girth seams of horizontal return-tubular boilers, what allowance, if any, is made for the reduction of effective cross-sectional area of the shell due to the presence of the tubes?

H. C.

Generally no such allowance is made, as in most boilers a considerable portion of the cross-sectional area above the tubes has no reduction of area, and parts of some of the girth seams are strained nearly as much as if no tubes were present. For practical considerations all girth seams of a boiler are usually made with the same size and pitch of rivets, employing the proportions suitable for resisting the greatest strain likely to come on any part of any girth seam.

Steam Drum vs. Steam Dome

Will the use of a steam drum on a boiler be as beneficial as equal steam space in a boiler dome, and if so, what is the best method of placing the inlet and outlet conections of a drum?

L. D.

A steam drum is fully as advantageous as a dome in affording steam room, and besides dispensing with expensive construction, eliminating a dome permits better and safer boiler construction. A good way to connect a steam drum of moderate length is by a saddle flange inlet placed in the middle of its length for connection to a flanged steam nozzle on top of the boiler, with a similar outlet flange in the opposite side of the drum.

Patching a Girth Seam

How is a patch applied to the girth seam of a boiler in a place which shows fire cracks at the rivets?

F. M.

Both sheets forming the girth joint have to be cut away and several girth-joint rivets removed on each side of the hole so made, for inserting a soft patch piece which will be inside of the outer course and outside of the inner course of the shell. The patch piece is jogged and scarfed where it comes between the shell plates, forming three thicknesses, which are drawn together by rivets inserted to re-form the ends of the original girth joint. The hole and the patch piece are laid out with rounded corners and so as to give good calking pitch of rivets and have all calking done from the outside of the boiler.

Flush Ends vs. Smoke Extension

What are the advantages of return-tubular boilers having flush front ends as compared with those having front ends extended?

J. P.

For settings with flush fronts, boilers with flush front ends are preferable, as serious leakage of heated gases direct from the furnace to the smoke connection is liable to occur from failure of fire-door arches, for when this happens in case of a boiler having extended end the lower part of the smoke connection is liable to be burned off; but when the boiler has a half flush front, with its smoke connection extending forward of the main front there can be no leakage of draft direct from the fire to the smoke connection and consequently no waste of furnace heat from lowering of the fire-door arch. The latter form of smoke connection has the disadvantage, however, of offering some obstruction to firing unless a boiler is set high enough or has the smoke extension beveled backward so as not to be in the way of the fireman.

Relative Heat Under a Boiler

With the ordinary proportions of furnaces and settings, which is the hotter place under a horizontal return-tubular boiler, immediately over the grates or over the bridge-wall?

L. J. D.

Over the bridge-wall.

Boiler Efficiency

What is the efficiency of a steam boiler and how is it estimated?

J. T.

The efficiency is the ratio between the heat units utilized in producing steam and the heat units contained in the fuel actually burned, and is therefore expressed by the fraction:

$$\frac{\text{Heat absorbed per lb. of combustible}}{\text{Calorific value of 1 lb. of combustible}}$$

The numerator would be the equivalent evaporation per pound of combustible from and at 212 deg. F. multiplied by 970.4 (the heat of vaporization, or latent heat of steam at 212 deg.). The denominator is taken as

$$14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right) - 4000\ S$$

in which C, H, O and S are the percentages of carbon, hydrogen, oxygen and sulphur, respectively, as determined by an ultimate analysis.

Study Questions

This Week's Questions
Last Week's Answers

(91) What is the total difference in pressure when a steam gage indicates 175 lb. and a vacuum gage 26 in. of mercury, if the atmospheric pressure is 14.7 lb.

(92) Two repair men are sent from a shop to fix an engine in a plant 12 miles away. *A* has a bicycle and agrees to let *B* ride it for alternate miles. *A* starts on the wheel and *B* on foot. At the end of the first mile *A* leaves the wheel for *B* and continues walking and when *B* reaches the wheel he rides for a mile and so on. Which reaches the destination first and how long does it take him if both ride at the rate of eight miles an hour and walk at the rate of four miles an hour?

(93) What is the weight of a wooden pipe 16 ft. long, 3½ ft. in external diameter, 9 in. thick, if the timber weighs 35 lb. per cubic foot?

(94) At what speed in revolutions per minute must an 8-pole generator be driven to give alternating current at 60 cycles per second?

(95) What are the acute angles in degrees, minutes and seconds of a right-angled triangle having a base of 35 in. and an altitude of 28 in.?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

(86) According to the Law of Charles, the volume of a gas under constant pressure, or the pressure of a gas at constant volume, varies as the absolute temperature. Absolute zero is 461 deg. below 0 deg. F. Then,

$$\frac{v_1}{v_2} = \frac{t_1}{t_2}$$

where

v_1 = Initial volume = 10 cu.ft.;
v_2 = Final volume;
t_1 = Initial absolute temperature = 32 + 461 = 493 deg. F.;
t_2 = Final absolute temperature = 212 + 461 = 673 deg. F.

Hence

$$v_2 = \frac{v_1 t_2}{t_1} = \frac{10 \times 673}{493} = 13.65 \text{ cu.ft.}$$

(87) Water weighs 62½ lb. per cu.ft. Then two tons is

$$\frac{2 \times 2000}{62.5} = 64 \text{ cu.ft.}$$

The area of the bottom of the tank will be

$$6 \times 4\tfrac{1}{2} = 27 \text{ sq.ft.}$$

Hence the inside height of the tank must be

$$\tfrac{64}{27} + \tfrac{1}{2} = 2.87 \text{ ft.}$$

The area of sheet lead required will be

$$\begin{aligned} 6 \times 4.5 &= 27 \\ 2 \times 6 \times 2.87 &= 34.44 \\ 2 \times 4.5 \times 2.87 &= 25.83 \\ &\hline \\ &87.27 \text{ sq.ft.} \end{aligned}$$

or

$$87.27 \times 144 = 12{,}566.9 \text{ sq.in.}$$

and the volume of lead

$$12{,}566.9 \times 0.125 = 1570.86 \text{ cu.in.}$$

Lead weighs 0.4106 lb. per cu.in., hence the weight required will be

$$1570.86 \times 0.4106 = 645 \text{ lb.}$$

(88) For the pump to discharge 400 gal. per min. it must be driven at

$$\frac{400}{3.2} = 125 \text{ r.p.m.}$$

The shaft speed will be inversely as the pulley diameters, then the line-shaft pulley diameter D may be found from the relation

$$\frac{D}{18} = \frac{125}{80} \text{ or } D = \frac{125 \times 18}{80} = 28\tfrac{1}{8} \text{ in.}$$

(89) A British thermal unit is defined as the amount of heat required to raise 1 lb. of water 1 deg. F. Then to raise 1 lb. from 50 to 120 deg. F. would require

$$1 \times (120 - 50) = 70 \text{ B.t.u.}$$

The mechanical equivalent of heat is

$$1 \text{ B.t.u.} = 778 \text{ ft.-lb.}$$

Then

$$70 \text{ B.t.u.} = 778 \times 70 = 54{,}460 \text{ ft.-lb.}$$

(90) In the formula for horsepower

$$Hp. = \frac{plan}{33{,}000}$$

where

p = Mean effective pressure in lb.;
l = Length of stroke in ft.;
a = Area of cylinder in sq.in.;
n = Number of strokes.

All of the quantities are known except a. Substituting

$$100 = \frac{36 \times 3.5 \times a \times 160}{33{,}000}$$

or

$$a = \frac{33{,}000 \times 100}{36 \times 3.5 \times 160} = \frac{3{,}300{,}000}{20{,}160}$$

$$= 163.69 \text{ sq.in.}$$

Since the area of a circle is

$$A = \tfrac{1}{4} \pi D^2$$

where

$\pi = 3.1416$

D = Diameter of the circle

$$163.69 = \frac{3.1416}{4} \times D^2$$

$$D^2 = \frac{163.69}{0.7854} = 208.416$$

$$D = 14.436 \text{ in.}$$

Over the Spillway

A large collection of the most eminent foreheads from the four quarters of Mother Earth recently gathered in New York for the International Congress for Testing Materials. There were princes, ministers, privy councillors, scientists, savants and royal and loyal Americans, all paying homage to old Father Science. It is their work which is making our jobs safer from fire and disaster, and they are spending $250,000 on this trip that they may learn more to tell more. Many things must be done in the rough before we get the finished product to warm up for operation, and these men in a quiet way are paving the broad and safe road for us.

Did you read Danny Hogan's "Daly and Doolin" story on page 366 of the Sept. 3 issue? Danny was right on the spot with a few predigested crumbs of thought on the boiler-rules situation and had a fine kernel of wisdom on "the mad desoire of engineers to monkey wid safety valves."

C. O'Two, says the *Fluegas Recorder*, is the new efficiency engineer in the Economy lighting station.

In the lively town of Brockton, Mass., lives a real live Italian countess who would rather wed a live-wire electrician than inherit a quarter million dollars. "There's nothing could buy my Andy!" says she. Now all you other "Andy" boys, buck up! In these days of high construction there is nothing out of reach of any ambitious Andy.

Marcel Meyer, chief engineer of the State Light & Power Electrical Station, a municipal plant in that dear Paris, has returned to La Belle France. Marcel is disappointed in Chicago, St. Louis and Buffalo. He seems to feel that America can't learn him nothing nohow. Another marcel-wave of disapproval for that poor America.

Thomas Fleming Day, a Buffalo yachtsman, has started for St. Petersburg in a 30-ft. boat having a two-cylinder, 16-hp. engine, and with 1200 gal. of fuel in the tanks. "Some Day," as the song goes. Here's hoping that Tommy does not drift across St. Peter before he reaches the "burg."

A politician had his greenhorn cousin appointed a smoke inspector, says the *Kansas City Journal*. The "inspector's" first monthly report read as follows:

"I certify that I have inspected the smoke of this city for the thirty days past. I find plenty of smoke and apparently of good quality. Respectfully submitted."

New Bundy Steam Trap

This is an improved form, but the same general principle in the construction and mechanism has been maintained, and the objectionable feature of the original trap, leaking trunnion joints, has been eliminated. The manufacturers state that there will be no trouble with packing or leaking joints in the new trap, as it is designed to be steam- and water-tight under all conditions and require no attention.

In the new trap the capacities have been increased as the trap works up to a much higher pressure which gives it a

FIG. 1. NEW BUNDY RETURN TRAP

larger range of usefulness. A side view of a return trap is shown in Fig. 1 and an end view of a tank trap in Fig. 2. A view of the new design of yoke is shown in Fig. 3. Connection is made from one bearing end to the bowl chamber; the other bearing end is connected to the cast outlet chamber.

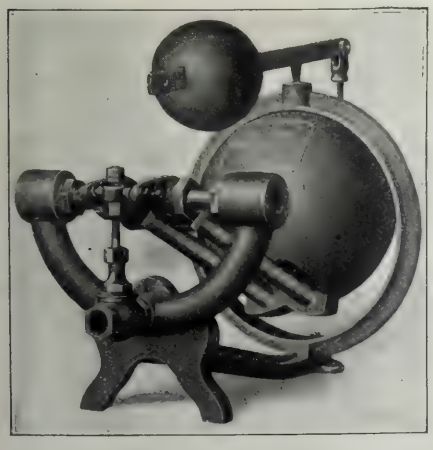


FIG. 2. TANK TRAP

The receiving bowl is cast round and of uniform thickness throughout, which, with its shape, offers a high factor of safety for operating under high pressure. The receiving bowl of the return trap has its steam inlet passage cast on the inside, Fig. 4, which eliminates the bent tube found in the old type of bowl.

In operation, condensation collects in the receiving bowl, which when filled tilts and opens the steam valve, admit-

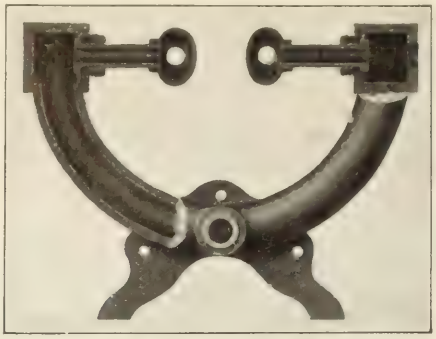


FIG. 3. THE NEW DESIGN OF YOKE

ting steam under boiler pressure to the bowl. The pressure in the trap and

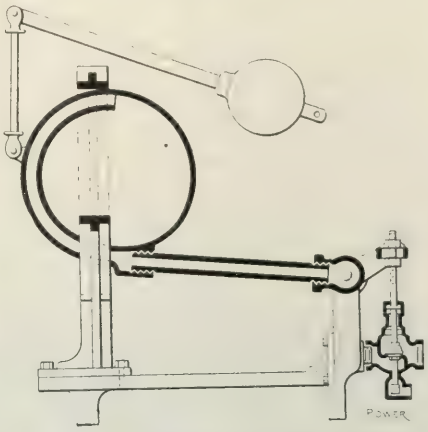


FIG. 4. SECTION OF RETURN TRAP

boiler is thus equalized and when placed above the boiler the water will flow into it by gravity. When empty, the receiving bowl tips back to its filling position and is ready for the next charge.

This trap is manufactured by the Nashua Machine Co., 127 Federal St., Boston, Mass.

Diesel Engine Ship Makes Good Record

The first of the Hamburg-American line's new oil-burning vessels to arrive at New Orleans was the "Christian X," which reached port on Sept. 7 from Hamburg after a successful voyage. It is of the new "stackless" type, equipped with Diesel engines using a low-grade of petroleum oil.

The "Christian X" left Hamburg on July 22, and on her first day, with good weather, made an average of 11½ nautical miles an hour. Later, under heavy weather and storm conditions, her general average for the trip was 11.01 nautical miles an hour. She spent four days in port at Havana and eight at Vera Cruz, and consumed 710 tons of oil. This is a creditable record for a vessel of this size driven by oil engines, for the "Christian X" is 370 ft. long by 53 ft. beam and, loaded, her tonnage is 9800. She has two heavy-duty engines, which can be controlled by a single lever, and reversed from full speed ahead to full speed astern in less than nine seconds.

The exhaust gases, after being cooled, pass out 49 ft. above the deck through the after mast, which is hollow.

Pension Plan for Employees

A pension plan has been announced by the General Electric Co. for its large force of employees. The pension will be 1 per cent. of the applicant's average yearly wages for the 10 years preceding retirement, multiplied by the number of years employed, and will be paid to men of 70 and women of 60, who have been 20 years or more in the company's service. Also provision is made for men of 65 and women of 55 who are incapacitated for further work.

The International Association for Testing Materials

The Sixth Congress of the International Association for Testing Materials, held at the Engineering Societies Building, New York, during the week commencing Sept. 2, called together between 800 and 900 delegates from all over the world, including many whose names are prominent in engineering, metallurgical and industrial circles.

Monday was devoted to registration and committee meetings, with a reception in the evening under the auspices of the American Society for Testing Materials, the American Institute of Electrical Engineers, the American Society of Mechanical Engineers and the American Institute of Mining Engineers.

On Tuesday morning the president and members of the council of the International Association for Testing Materials received formally the official delegates from governments, and at 10 o'clock the opening session of the congress was held. Addresses of welcome were delivered by Robert W. Hunt, president of the American Society for Testing Materials, by Gen. William H. Bixby, chief of engineers of the United States Army, on behalf of the President of the United States, John A. Dix, governor, on behalf of the State of New York, William A. Prendergast, controller, on behalf of the mayor of the City of New York, and Henry M. Howe, acting president of the International Association, delivered his presidential address.

In the afternoon the various sections held their professional sessions, and in the evening an opportunity was given to visit the New York Public Library under the guidance of the city officials.

Wednesday forenoon was devoted to professional sessions, and in the afternoon the party was taken by steamer to

the United States Military Academy at West Point, returning to New York late in the evening.

On Thursday sessions were held both forenoon and afternoon, and in the evening a reception was tendered to the foreign delegates and foreign members by the American Society of Civil Engineers at its house. During the evening, Alfred D. Flinn, member of the American Society of Civil Engineers, gave an illustrated lecture on the engineering activities of New York City.

On Friday the entire day was devoted to professional sessions, and the evening left free.

On Saturday the closing session of the Congress was held in the forenoon. In the afternoon a reception was tendered to the members of congresses of testing materials and of applied chemistry, which was also in session in New York at the same time, by the Museum of Natural History, and in the evening by the Metropolitan Museum of Art.

Following the congress a number of tours have been arranged, in which many of the delegates will participate.

Second Annual Meeting, Institute of Operating Engineers

The second annual meeting of the Institute of Operating Engineers, held in the Engineering Societies Building, 29 West Thirty-ninth St., New York City, Sept. 6 and 7, was highly successful.

Friday morning, C. A. Prosser, secretary of the National Society for the Promotion of Industrial Education, made the welcoming address which was responded to by President J. C. Jurgensen of the institute. After a reading of a summary of the minutes of the last annual meeting, the various committees were appointed. The report of the finance committee was then read, which showed the institute to be in favorable circumstances financially.

At 2:15 p.m. the professional session opened and continued throughout the afternoon. This session was very well attended and the papers read were received with much interest.

Alexander Gordon, chief engineer of the Boston Store, Providence, R. I., read a paper on the subject of the use of tungsten lamps as a commendably practical way of reducing the load on an overloaded plant; an abstract of which appears elsewhere in this issue. A paper by Robert H. Karl, chief erecting engineer for the Anheuser-Busch Brewing Co., of St. Louis, discussed the effect of varying the suction pressure on the economy of ammonia compressor machines. Interesting discussion led by H. W. Geare followed the reading. Mr. Karl's paper will appear in an early issue.

"Keeping Tabs in Industrial Plants" was the subject of a paper by Peter Lunbeby, which contained many excellent suggestions for establishing a system of record keeping, and will appear soon in POWER. Other interesting papers covered: "The Steam Engine Indicator," by C. S. Foster; "Steam in a Sugar Cane Factory," by W. A. T. Willink; "Watertown Engine Improved by Extra Exhaust Valve," by Wendolein Senge, and "Electric Drive versus Steam Drive," by J. F. Carman.

The commencement exercises, held Friday evening, were opened by Prof. Maxwell, of Pratt Institute, as Dr. Richards, Director of Cooper Union, who was to address the institute could not attend. Prof. Maxwell in his address on "Industrial Education and Its Relation to the Operating Engineer," emphasized the great advantage of the engineering student "finding himself" before he went too far in such studies, as only too often the student discovers, when too late, his inaptitude for such work. In this connection he condemned the proposed six-year college-engineering course. This course as proposed would require two years of practical work after the studies were completed. Prof. Maxwell contended that the practical work should precede the studies to determine the student's dormant qualifications as an engineer.

President Jurgensen presented diplomas to the class graduating as master operating engineers. Those honored were: Michael Furlong, James Fagan, John T. McDonough, John McGinley, Alonzo McLaughlin, George Pattberg and Harry W. Wright. The valedictory oration was made by Mr. Furlong.

The business meeting was called to order at 10 a.m., Saturday. In the afternoon the following officers were elected:

President, F. L. Johnson, chief of construction department, Edison Electric Illuminating Co., Brooklyn (one-year term); vice-president and director department of vocational statistics and employment, V. L. Rupp, chief engineer, Williamson Trade School, Williamson, Penn. (three-year term); vice-president and director of department branches and meetings, William J. Fair, chief engineer, Murray Hill Hotel, New York City (two-year term); vice-president and director of department of industrial education and training, William J. Freer, chief engineer, Citizens Central Bank Building, New York (one-year term). (Mr. Freer had been fiilling out the unexpired term of J. G. Ould, deceased); treasurer, W. P. F. Hill, chief engineer of Woman's Hospital, New York, reëlected (one-year term); commissioners at large, Lawrence Honmiller (one year), Darrow Sage (two years), Robert Langlotz (three years); commissioners, District No. 1, J. L. MacVicar, chief engineer, Boston City Hospital, Boston (three years) (reëlected); District No. 2, W. A. T. Willink, erecting and designing engineer, Arbuckle Brothers, New York (three years); District No. 3, D. B. Heilman, inspector, Philadelphia & Reading Ry. Co., Reading. Penn. (three years) (reëlected).

SOCIETY NOTES

Minneapolis Association No. 2, N. A. S. E., is arranging a "get together and booster" meeting for its last meeting in September, on Saturday evening the twenty-eighth. J. F. Flather, professor of mechanical engineering, University of Minnesota, will be among the speakers. It is the association's desire that every old member and every engineer interested in the work will be present and "get together" and boost the purpose and principles of the N. A. S. E.

Members of the American Society of Mechanical Engineers are urged to notify the secretary if there is any possibility of their taking part in the joint meeting with the Verein Deutscher Ingenieure at Leipzig in June of next year. It is the wish of the German society that as many representative American engineers shall accept the invitation as possible, and the invitation is broad enough to include not only the present members of the American society, but to permit those members to bring with them desirable guests. It will greatly facilitate the work of the committee on arrangements if they can have an early indication of the number for which they will have to provide.

PERSONAL

Thomas L. Delahaunty, who succeeds the late James J. Waters as corresponding and financial secretary and business manager of Marine Engineers' Beneficial Association No. 33, was born in Charleston, S. C., in 1865. Obtaining his first license in 1889, he sailed as an engineer in the old Pacific mail steamer "City of Para" between New York and Colon, going from there to the old Rio Line and serving as an engineer on all that line's steamers. He was afterward chief engineer in the Ward and Mallory lines. Shortly after the New York municipal ferries were inaugurated, Mr. Delahaunty was employed by the city. He resigned later and for the past four years has been chief engineer of the American Mail Steamship Co. on the steamers "Admiral Schley" and "Admiral Dewey." Mr. Delahaunty has been an enthusiastic member of the marine engineers' association ever since he obtained his original license as a marine engineer, and he was a representative to the thirty-fifth, thirty-sixth and thirty-seventh national conventions. He brings to his new position, to which he was unanimously elected by his fellow-members, a wide experience in matters affecting marine engineering.

POWER

Vol. 36 NEW YORK, SEPTEMBER 24, 1912 No. 13

A Bigger and Brighter Opportunity

AS the work of the operating engineer broadens and the value of the machinery in his charge increases, the day approaches when the manager will ask him to visit other cities and plants to gather data upon some important installation. This may be a difficult task at first, but by careful preparation one can do it.

When this opportunity comes, get right down to business and begin by reading up on the subject involved. Then your interest will be larger and you can ask intelligent questions when you get on the firing line.

Buy a serviceable loose-leaf book—you can get one for about a dollar—and put a medium-hard pencil or two and an eraser in your pocket. In the front of the book write the leading questions you mean to ask, numbering each to save the time of the man you interview when you put down his replies. Pretty homely advice? Surely, but then you won't come back with half your points overlooked.

Put several strong manila envelopes in your grip to hold any loose material. Collect enough live data to make your report thorough.

If you are a photographer, ask permission to take a picture of any apparatus, etc., that appeals to you.

Gather all the latest publicity matter bearing upon your subject. Ask if a description of the plant has ever been published. Buy the local newspapers and get the atmosphere of the place. Make a list of similar representative installations elsewhere, and aim to visit them enroute.

When you get home, having put down in your loose-leaf book under the proper headings all the business odds and ends of your trip, ask the Old Man to let you either dictate your report or have a typewritten copy made, and don't forget to keep a carbon copy yourself.

Never mind the literary style, just give the plain facts in straightforward order. Write your report while every impression is fresh and sign your name. Draft your report roughly on the way home.

If you work for a house that is worth while, you will be expected to travel in unpretentious comfort. Get a room at a favorably known hotel, and have three square meals a day. Travel in a parlor car on long trips; you can keep cleaner, and often get in touch with people that will benefit you if you sustain that personal reserve which demands the respect of all experienced travelers. But cut out the luxuries!

When you get home, itemize your expense account so that it can be audited easily. Every hour you are away, you personally represent your employer. Act for his interest all the time; it will pay in the long run.

Many times a trip is better than a post-graduate course to the man who takes it with his eyes open. When you hear the call of the Old Man, don't forget for a minute that it spells Opportunity.

It is a chance to become a bigger man and get away from the routine work. Other engineers have done it, are doing it; why not you?

Hydraulic Excavation at Panama

By Fred H. Colvin

An interesting pumping plant and dredging equipment used to remove the loam overlying the rock near the Pacific entrance of the canal.

One of the interesting power plants in the Panama Canal zone was that used for the hydraulic excavating on the Pacific end between the Miraflores lock and the Pacific entrance of the canal.

Time being an important element and it being impossible to assemble a sufficiently large dredging plant to complete this section within the time limit fixed, it was decided to install a hydraulic excavating plant as the most expeditious and cheapest means of handling the loam overlying the rock. An additional advantage was the utilization of the material removed for reclaiming the swamp land along the line of the canal and belonging to the canal commission.

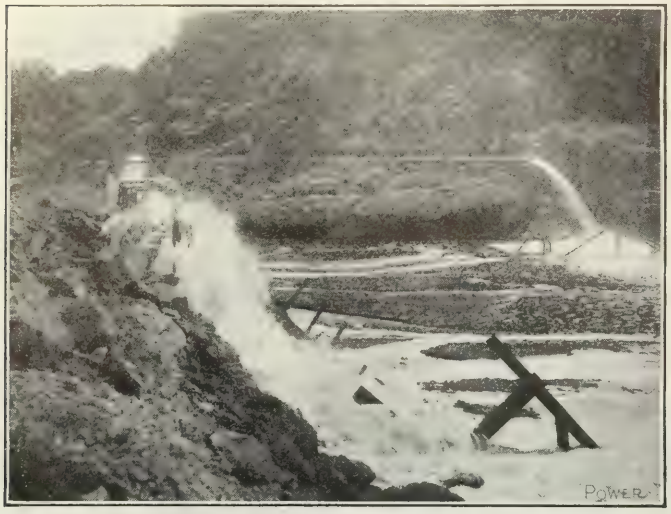


FIG. 1. DISCHARGE IN HYDRAULIC FILL AT COCOLI DAM

FIG. 2. ONE OF THE MONITORS AT WORK

By using a water jet under pressure, the earth overlying the rock was washed into sumps or low spots where centrifugal dredging pumps elevated and conveyed the material through flumes to the desired point of discharge. A central pumping station, shown in Figs. 5 and 6, was located on the west bank of the canal in the center of the area to be excavated. There were installed four Worthington, horizontal triple-expansion pumping engines with steam cylinders 19, 30 and 50 in., water cylinders 24½ in., and a 24 in. stroke. The condensers were of the surface type and each was provided with a 12 and 20 by 24-in. vacuum pump.

Steam was supplied by four Babcock & Wilcox water-tube boilers arranged in batteries of two. They were oil fired from two steel tanks of 2000 bbl. capacity, erected on a hill for gravity feed to the burners.

The pressure line running from the pumping station was 3600 ft. long, made up of 2000 ft. of 40-in. and 800 ft. of 32-in. lock joint pipe, and another 800 ft. of 24-in. spiral riveted pipe. The main was provided with valves and tees suitably located for connecting branch lines leading to the nozzles or monitors.

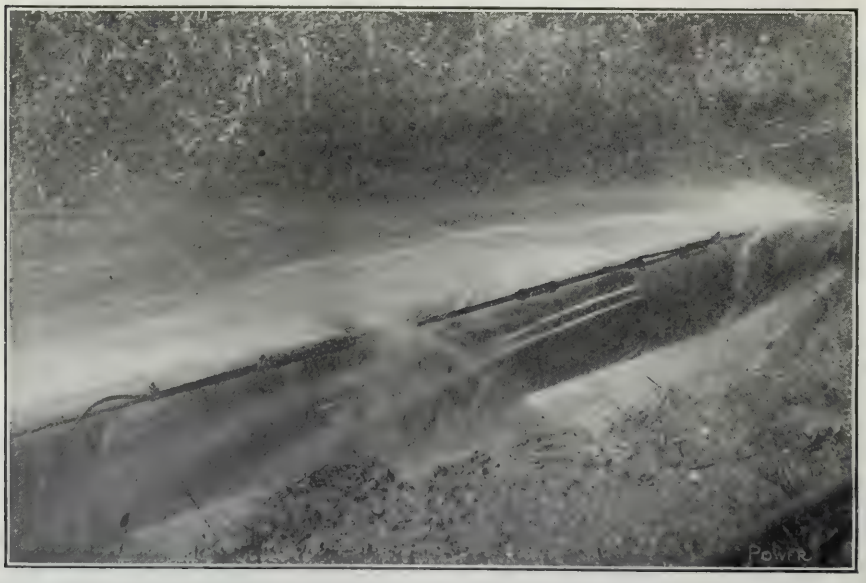


FIG. 3. ONE OF THE BAD SPOTS IN THE LINE

FIG. 4. MORE OF THE LEAKAGE ENCOUNTERED

The branch lines were of 16-in. spiral riveted pipe laid in groups of three so that two monitors might be continued at work while the third was being changed.

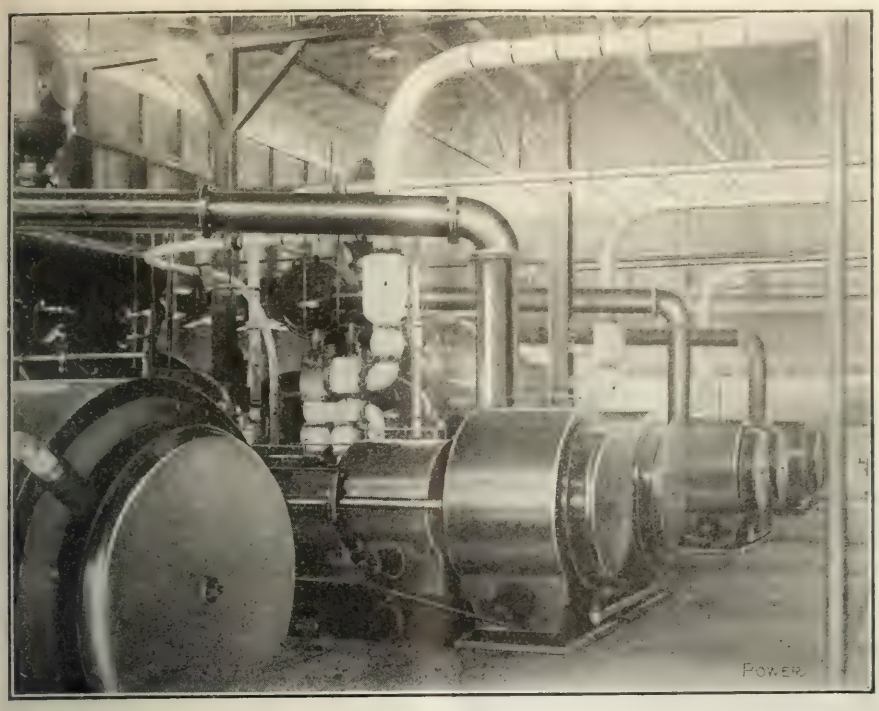


FIG. 5. CENTRAL PUMPING STATION AT AGUA DULCE

The monitors were fitted with special deflecting nozzles.

There were several dredges, each equipped with three 18-in. single-reduction centrifugal pumps, directly connected to a 655-hp. induction motor. These pumps, motors and auxiliaries were mounted on reinforced concrete barges specially designed and constructed for this purpose. After the removal of the earth overlying the rock by the hydraulic process, the rock was excavated by steam shovels.

Fig. 6 shows the pumping station under erection. The pumping engines and piping were first put in place and the station built around them.

As the pumping pressure was 125 lb. per sq.in., it was not easy to keep the lines tight, due to the settling of the earth and the strain put upon the riveted piping. Fig. 3 shows what happened in one of the bad places and the means taken for holding the pipe together so that the work could go on until repairs could be made. Fig. 4 is another view showing somewhat similar conditions and giving an idea of the leakage encountered.

Fig. 1 shows the discharge into the hydraulic fill of Cocoli dam, the same method of filling being employed at the large Gatun dam at the other end of the canal. One of the monitors is shown at work in Fig. 2, the nozzle being about 6 in. in diameter. These monitors could be swung easily into any desired position to wash the earth down into the sump from which it was readily handled by the dredging pumps. One of the concrete barges may be readily seen behind the stream from the nozzle. An iron grating over the end of the suction pipe kept out sticks and stones which would interfere with the working of the pump, but occasionally débris would accumulate across the grating to such an extent that it was necessary for a colored laborer, usually in his birthday suit, to dive into the muddy water and remove the obstructions. After seeking the consolation of a convenient shower bath he awaited the next call of duty.

The whole plant made an interesting installation but has now become obsolete as the earth has been removed and the steam shovels are at work on the underlying rock.

FIG. 6. PUMPING STATION UNDER ERECTION

A Large Pumping Engine

By C. W. Allcorn

Wheeling, W. Va., has just added to its water-works a new pumping engine having a capacity of 20,000,000 gal. per 24 hr. It is a vertical triple-expansion engine of the crank and flywheel type, having 42-, 74- and 110-in. steam cylinders and a 72-in. stroke. The low-pressure cylinder is the largest ever used on a pumping engine; its piston is shown in Fig. 2.

The steam cylinders are supported on double A-frames, one side resting on the water ends and the other on a pier. The cylinders are steam jacketed; the steam from the high- and intermediate-pressure cylinder jackets being tapped into the second receiver and utilized in the low-pressure cylinder. The first and second receivers are 5½x8½ ft. and 7½x9 ft., respectively, and are mounted on the engine frame in the space between the cylinders. A steam pressure of 110 lb. is employed, although the engine is designed for 125 lb. and a 27-in. vacuum. Both steam and exhaust valves on the high-pressure cylinders and the steam valves of the intermediate cylinders are of the Corliss type. The exhaust valves of the intermediate, and both the steam and exhaust of the low-pressure cylinder are cam-operated poppet valves. The steam and exhaust valves are operated by separate eccentrics mounted on a lay-shaft and driven by drag cranks and connecting rods on each end of the 27-in. main crankshaft. A flyball governor controls the high-pressure cutoff, but has a rod attached by means of which the speed can be adjusted by hand while running. The intermediate-pressure cutoff is hand controlled while the low-pressure valves are definitely set and not adjustable while running. The engine runs a little over 18 r.p.m. or at a piston speed of 220 ft. per min. The two flywheels weigh 90 tons. A surface condenser is provided, which receives its cooling water from a bypass in the discharge line. The condenser pump is driven direct from an arm on the low-pressure plunger rod. A jet condenser is to be installed to be used when repairing the surface condenser.

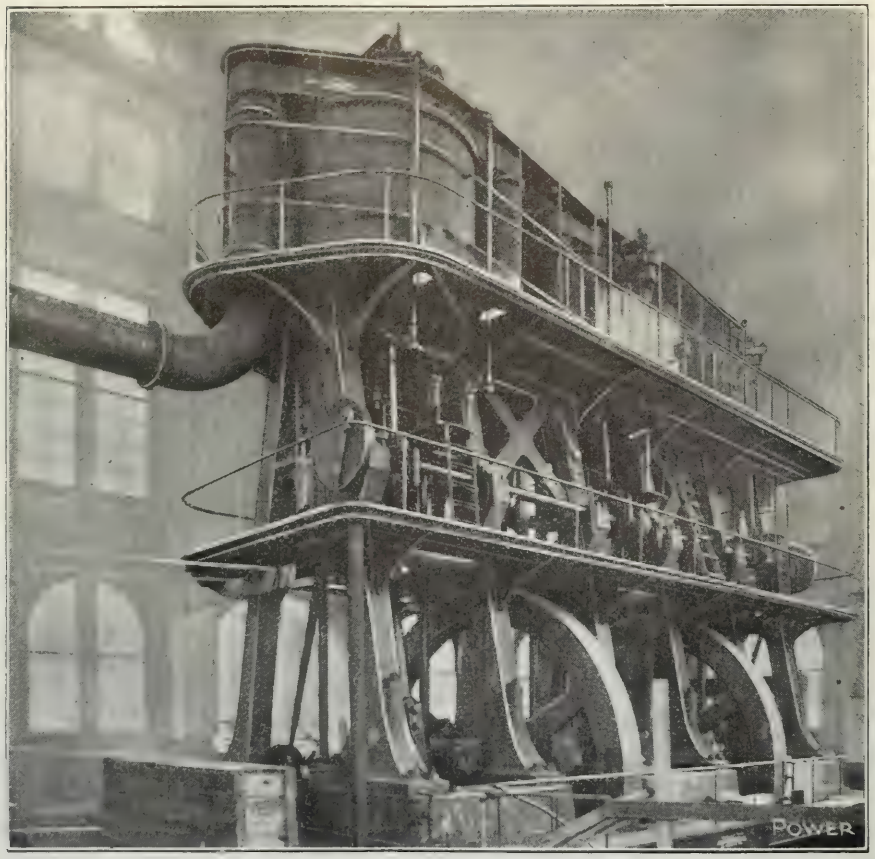


Fig. 1. View of Steam End

The pumps are of the single-acting outside-packed plunger type and are located directly under the steam cylinders. The plungers are 33 in. in diameter and each has its cast-steel crosshead with four distance rods extending to the engine crosshead, thereby deriving direct motion from the steam piston. The valve chambers are 72 in. in diameter, and the valve decks 79 in., containing in all 1116 valves. The valves are of the weighted type, 4 in. in diameter, and are arranged 186 in each deck. Located at the bottom of the pit which is 46 ft. from the engine floor is a six-basket 36-in. Lagonda strainer.

The pump gets its water direct from the Ohio River and pumps it through two miles of 36-in. pipe to the reservoir against a static head of about 250 ft.

The distance from the top of the steam end to the base of the water end is 84 ft. 8 in. A galley is placed at the pump-valve chambers and one above it gives access to the air pump and pump plungers. On the engine there are two galleries; the lower giving access to the crosshead eccentrics, etc., and the upper for inspection of the valve-gears, lubricator, and for applying an indicator motion. A platform at the top of the cylinders is arranged for the inspection of the valves and the receivers. The engine was built by the Allis-Chalmers Co.

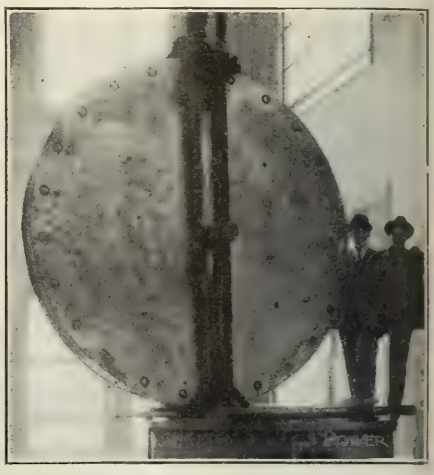


Fig. 2. Low-pressure Piston

Gold Metal for Safeguarding Life

The Allgemeine Electricitäts Gesellschaft. of Berlin, has cabled President Arthur Williams, of the American Museum of Safety, that the Rathenau gold medal has been placed at the disposal of the museum for award annually for the best device or process for safeguarding life and limb or promoting health in the electrical industry.

Competition is open to every country, the only condition being that the device or process must be exhibited at the American Museum of Safety, New York City.

The Rathenau medal is well known in Europe. It was presented to Dr. Emil Rathenau, president and founder of the Allgemeine Electricitäts Gesellschaft, on his seventieth birthday, with the felicitations of the Kaiser for his services in the field of electro-technics and having introduced incandescent lighting into Germany. One medal will be cast each year from the original die for the museum to award.

The water power of the River Rhone at a point 280 miles from Paris is to be used in supplying that city with electric light and power. Plans have been completed for the dam and power plant.

Limitations of Reducing Wheels

By Julian C. Smallwood*

A series of tests made with a special tester devised by the author upon the drum motion produced by various reducing wheels. The errors and limitations of reducing wheels are pointed out and some general operating rules given to insure accuracy.

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In the Aug. 20 issue the writer presented the results of tests on engine indicators showing the distortion of the drum motion at all parts of the stroke due to the stretch of the cord. The instruments tested were operated under various conditions such as might be met in practice when an accurate reducing motion as of the pantagraph or link type is used. From the results it appeared that the diagram obtained by the ordinary indicator may be seriously in error, both as to mean effective pressure and events of the stroke, when operated at speeds for which it is not adapted and under certain other adverse conditions.

Link-reducing motions, however, have been largely supplanted by reducing wheels on account of the latters' ready adaptability. To complete the drum-motion investigation, then, tests of this type of reducing motion are necessary. Since the cord used with this device is longer and the mass moved by the cord is greater than with link motions, the stretch of the cord and the inertia of the wheel may cause appreciable error.

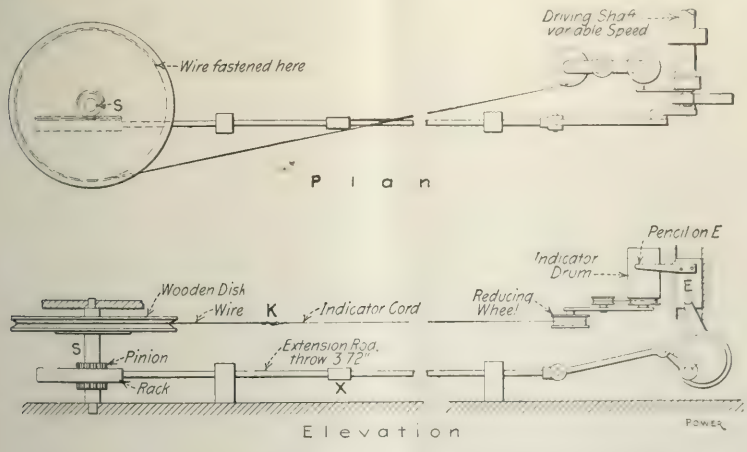


FIG. 1. DRUM-MOTION TESTER WITH ATTACHMENT FOR LONG STROKES

To test the drum motion produced by various reducing wheels, an addition was made to the drum-motion tester devised by the author and previously described. The apparatus is shown diagrammatically in Fig. 1, and as photographed in Fig. 2.

The drum-motion tester, as originally used, carried a bracket at *X*, to which the indicator cord was fastened, thus duplicating the condition of a link-reducing motion. As the motion of the pencil point carried on the vertically traveling crosshead *E* is exactly proportional to the motion of the bracket, the diagram produced by the tester is one of crosshead motion shown vertically against drum motion horizontally. When the drum and crosshead motions are exactly proportional the diagram is a straight inclined line and there is no error. The stretch of the cord, however, causes lack of proportionality and a closed curve results. The departure of any point on this curve from the straight line adjoining its uppermost and lowest points marks the error of the drum motion. Fig. 3 shows an error diagram and is self-explanatory.

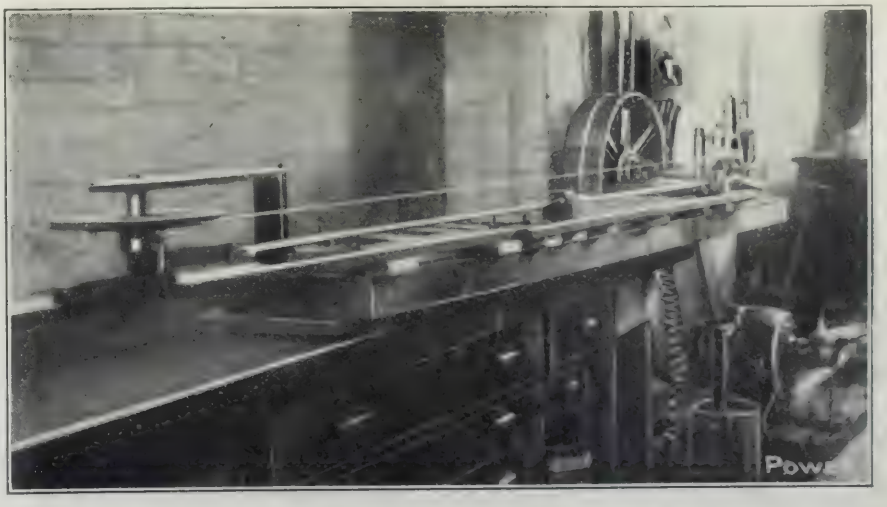


FIG. 2. THE DRUM-MOTION TESTER AS SET UP FOR A TEST

To adapt this device to reducing wheels, a bracket, or some equivalent attachment is necessary with a long stroke to duplicate an engine crosshead motion; and, to make the apparatus cover all operating conditions, the stroke should admit variation. This is conveniently accomplished by coupling an extension rod to the motion tester, carrying at its outer end a rack meshing with a pinion on a vertical shaft. This shaft *S*, Fig. 1, carries a wooden disk which was replacable by others of different diameters. When the rack is reciprocated, the disk is given an oscillating motion. An extension of the indicator cord from the reducing wheel being fastened to a point on the circumference of the disk, the cord will be reciprocated through a stroke the length of which depends upon the diameter of the disk. By using different disks, a reducing wheel may thus be operated through any desired stroke.

The apparatus was made with great care to avoid back lash and other imperfections. The rack and pinion were cut with involute teeth and were held closely in mesh by an adjustable roller at the back of the rack. The wooden disks were turned on the shaft to make them perfectly concentric and were held by a 5-in. faceplate which also located centers. For the long strokes the effort of the reducing wheel spring was partly balanced by another spring working oppositely on the disk. The pulley driving the apparatus was large enough to act as a flywheel and make the speed of rotation uniform, as a tachometer held to the pulley shaft gave steady indications. A calibrated tachometer was used for all speeds above 100 r.p.m.

In the operation of a reducing wheel the same forces are set up as in the indicator drum discussed in the previous article. These are spring tension, friction and force due to inertia. Since the drum and wheel masses are connected

by a short cord having inconsiderable stretch, they may be regarded as one mass producing a single inertia effect. Likewise, the two springs may be considered as exerting a single force. In Fig. 4, the component forces are shown as they are felt at the cord at three parts of each forward and return stroke. The arrows represent roughly by their length

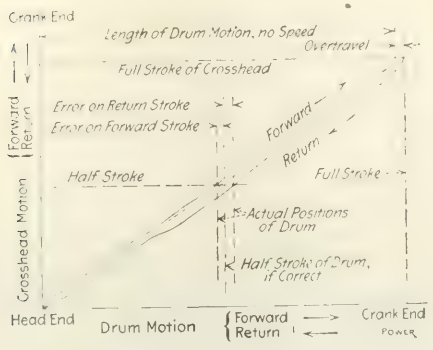


FIG. 3. DRUM-MOTION TESTER DIAGRAM. DOTTED LINE REPRESENTS CORRECT MOTION

the comparative magnitude of the forces.

At the beginning of the forward stroke, the cord exerts a maximum force in overcoming friction, inertia and spring tension, but as most of the cord is wound on the wheel, its stretch and the consequent lag of the drum are inappreciable.

As the crosshead advances the length of cord free to stretch becomes greater, but, as the velocity of the moving parts approaches its maximum in the neighborhood of midstroke, the effect of inertia is to lessen the force exerted by the cord and therefore its stretch. This continues throughout the forward stroke; as the length of the cord increases its stress decreases, and the net effect is to maintain a reasonably uniform total stretch and consequently to give a correct drum motion. On the return stroke, however, the spring tension is required to accelerate the wheel sufficiently so that the cord does not slacken, and it must do this not only against inertia, but against the frictional resistance.

As the friction is large, the spring may not be able to do this at a rate equal to that at which the crosshead is being accelerated, particularly at the beginning of the stroke, as here the acceleration is greatest. The stretch of the cord is therefore released at the beginning of the return stroke, if not entirely slackened, as evidenced by whipping. Beyond midstroke, the moving parts of the wheel have gained sufficient momentum to draw up the cord again, and this momentum keeps it taut to the end of the return stroke. Under ordinary circumstances, then, a marked distortion of the drum motion may be expected only through the return stroke and greatest at its beginning.

Obviously the length of the cord free to stretch, i.e., that part of it not wrapped around the wheel, has a decided influence upon the correctness of the motion. In practice this length at its minimum and maximum is determined by the engine stroke. Normally, at the head end, the length of cord free to stretch is about equal to the stroke, and at the crank end twice as long.

The experiments were made, except where specifically noted, with a length of cord to duplicate this condition. This was accomplished by tying the indicator cord, when adjusted to the desired length, to a piece of stranded wire which was then connected to the disk, Fig. 1. The wire being practically inextensible, the effect was to reciprocate the cord through the desired lengths as by an engine crosshead at the point K.

The cord used was of medium grade, unless otherwise noted, showing a stretch of 0.02 in. per foot per pound when stressed between 1 and 4 lb.

The total spring tension referred to the line of the cord at the crank-end dead center of motion was recorded in each series of tests. This was found by fastening the cord to a spring balance and drawing it slowly past the dead center so that the total tension in the cord could be observed at the moment of passing. This tension is equal to the force of the spring S plus friction F. The cord was then allowed to rewind on the wheel, and another reading taken upon passing the dead center in the reverse direction. The value so found equals the spring tension minus friction. Then, using numerical values,

$$S + F = 48\ oz.$$
$$S - F = 24\ oz.$$

Adding,

$$2S = 72\ oz.$$
$$S = 36\ oz.$$

and

$$F = 48 - 36 = 12\ oz.$$

or 33 per cent. of the spring force.

In general the friction is a fixed percentage of the other forces, and as it imposes a serious limitation to successful operation, the importance of this determination should be emphasized.

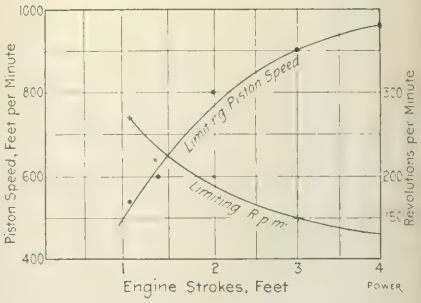


FIG. 5. CURVES SHOWING HIGHEST SPEEDS AT DIFFERENT ENGINE STROKES; WHEEL NO. 1, INDICATOR "C"

Four wheels were tested after being put in best working order by an expert instrument maker. They were run with indicators A and C, described in the article on drum motion and a third indicator D. Brief descriptions follow in which the wheel used for adjusting the motion to different engine strokes is referred to as the stroke wheel, and the one to which is attached the cord from the crosshead as the main wheel:

Wheel No. 1, used with indicator C. Spiral (clock) spring. Spring barrel carries the stroke wheel. Stroke wheel connected to main wheel by a short cord.

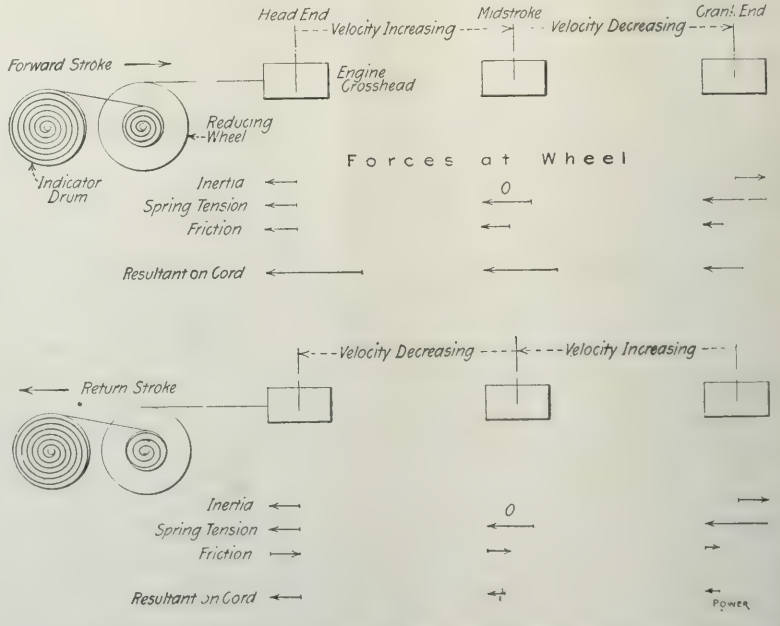


FIG. 4. FORCES ON A REDUCING WHEEL AND INDICATOR DRUM DURING OPERATION

Total friction of wheels and drum, 20 per cent. of the other forces.

Wheel No. 2, used with indicator C. Spiral spring. Same arrangement of wheels as No. 1 except that the main and stroke wheels are connected by gears. Friction, 50 per cent.

Wheel No. 3, used with indicator A. Helical spring. Same arrangement of wheels as No. 2. Ball bearings. Friction, 33 per cent.

Wheel No. 4, used with indicator D. Spiral spring. Stroke and main wheels are combined in a single small light wheel whose shaft is geared to the indicator drum. Light spring behind wheel, but drum spring furnishes the greater force to return the parts. Drum, 2 in. in diameter and comparatively heavy. Friction, 23 per cent.

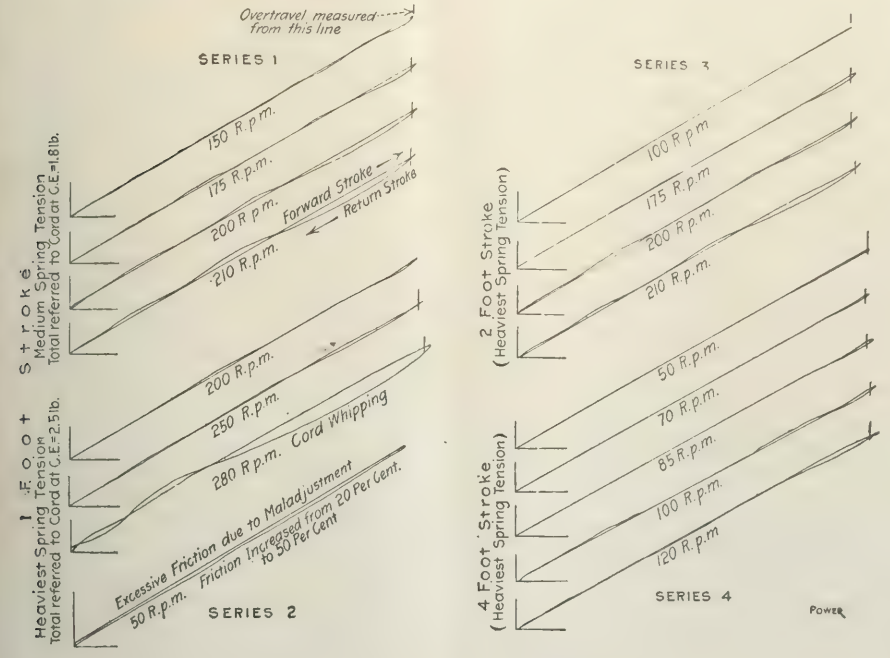


CHART 1, WHEEL NO. 2, INDICATOR "C"

For the presented results from the first three wheels, the drum-spring tension was kept constant for both indicators, and equal to 2¼ lb. at midstroke. This tension was found by previous tests enough to accelerate the mass of the drum only; the acceleration of the reducing-wheel masses was taken care of by varying the tension of its spring. Some tests were made with greater drum-spring tensions, but the results are not presented for reasons to be noted later. At each engine stroke, drum-motion tester diagrams were taken at increasing speeds of revolution until the effect of inertia was apparent to the eye through the whipping of the indicator cord, which condition would not be tolerated in good practice.

The results from wheel No. 1 and indicator C are such as to be met in the operation of good, if not equal to the best, apparatus, and are therefore presented more fully than the others. Tests were made of this combination at 12-, 16-, 24-, 36- and 48-in. engine strokes with many variations of the operating conditions, quality and lengths of cord and the like; but only the representative results are presented. These are given by Chart 1, the diagrams being the same as that indicated in Fig. 3, but closely spaced for convenience. For a 12-in. stroke, the motion is correct up to 150 r.p.m. when a medium wheel-spring tension is used (Series 1). At this speed there is a slight lag at the beginning of the return stroke which becomes more pronounced at higher speeds until at 200 r.p.m. there are considerable errors during most of the return stroke. This accords with the reasoning in the preliminary analysis, that so much of the spring tension is absorbed in overcoming friction that enough is not left to promptly accelerate the wheel after passing the crank-end dead center.

Increasing the wheel-spring tension, as shown by Series 2, enables the wheel to be run at higher speed; the limit is reached at 280 r.p.m., or a piston speed of 560 ft. per minute with the total spring tension, referred to the line of the cord, 2.5 lb.

The other diagrams of Chart 1 were obtained with this same total tension, the highest obtainable without increasing that of the drum spring. A study of the chart brings out the following subjects for discussion:

First, the overtravel of the drum is small at both ends of the stroke.

Second, the distortion of the return-stroke motion is very marked and follows a characteristic wave. Notice that the forward motion is in all cases very closely true.

Third, the piston speeds producing whipping and therefore limiting the operation of the wheel, are much greater at the longer strokes.

The first effect is largely due to friction which helps the springs to check the velocity of the wheel at the end of the forward stroke. Much overtravel at the crank end would be evidenced by whipping of the cord. For instance, if the drum motion were 4 in. and the engine stroke 24 in., an overtravel of the drum of ⅛ in. would mean a slackening of the cord six times that amount, or ¾ in., since any distortion of the drum motion is multiplied at the wheel by the ratio of the motions. This is enough to cause whipping unless the cord is very stretchable. At the head end the overtravel is subdued not only by friction but by the positive resistance of the cord, the stretch of which is a minimum at this end because of its reduced length.

The second effect is due not only to friction, but is increased because an indicator cord varies very considerably in length for changes of stress below 1 lb., although above that amount the stretch is comparatively small and uniform with increasing stress. Also the cord will contract markedly when the stress reduces to zero, and sometimes has been observed to continue contracting after the stress was entirely removed. If the frictional resistance to motion of the reducing wheel prevents the spring from properly accelerating it upon the return stroke, the crosshead motion gains upon the wheel motion, thus reducing the stress in the cord, perhaps to only a few ounces. The cord accommodates itself to this

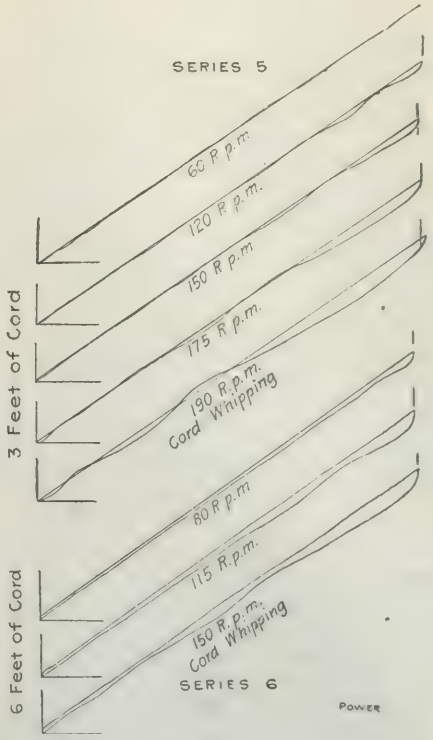


CHART 2, WHEEL NO. 2, INDICATOR "C"

change in length because it contracts markedly at low stresses, and not until the distortion of the drum motion is quite pronounced is the speed of contraction exceeded by the crosshead speed, as evidenced by whipping.

The lag of the drum motion just after passing the crank-end dead center is only momentary at moderate speeds because the force required to accelerate the wheel is a maximum at dead center; when it has once started upon the return stroke less force is needed to continue the motion and this the spring is able to deliver. At higher speeds the action of the spring is further delayed, so that the late acquired momentum of the parts causes them to overshoot the mark and fetch up

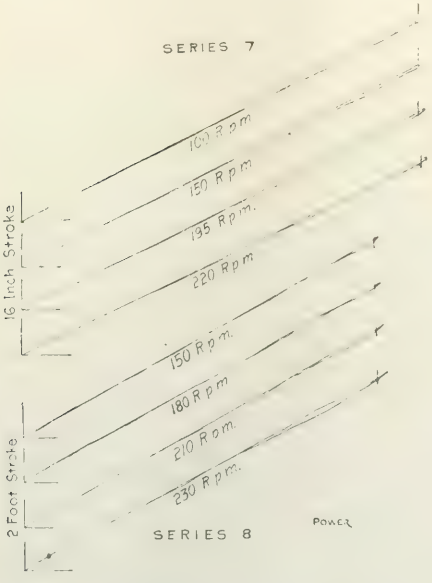


CHART 3, WHEEL NO. 3, INDICATOR "A"

against the positive resistance of the cord, resulting in the wave previously noted.

The third effect follows from the limitations caused by inertia. It is shown more clearly by the curve, Fig. 5, of highest piston speeds plotted against engine strokes. The speeds plotted were those observed to cause a slight vibration of the cord; the highest spring tension was used in each case. The effect is explained by the fact that at a given piston speed the oscillations per minute of the wheel are greater, the smaller the engine stroke, and therefore the effect of inertia is greater with the small strokes. Obviously, it is more difficult to oscillate a mass moving at a fixed average speed, the more frequent are the oscillations. This also follows from the mathematical value for the force required to accelerate a reducing wheel at the ends of the stroke (assuming harmonic motion) which may be expressed as follows:

$$A = W \times L \times N^2 \times a \text{ constant}$$

or

$$A = W \times LN \times N \times a \text{ constant}$$

in which

A = Force of acceleration referred to the cord;
W = Weight of moving parts referred to the cord;
L = Length of engine stroke;
N = Revolutions per minute.

This may be interpreted "Inertia of a reducing wheel varies directly with the stroke and the square of the r.p.m.," or "directly with the stroke and the piston speed." It follows that wheels arranged as are Nos. 1, 2 and 3 will operate less successfully at the shorter strokes.

This analysis does not consider the inertia of the indicator drum. The throw of the drum is practically constant at all strokes, so the force required to accelerate it at the ends of the stroke is

$$a = w \times l \times N^2 \times a \text{ constant}$$

or

$$a = w \times N^2 \times a \text{ constant}$$

in which w is the weight of the drum and l its stroke. Thus, a varies with the square of the r.p.m. only.

The total force of acceleration of drum and wheel is the sum of these forces a

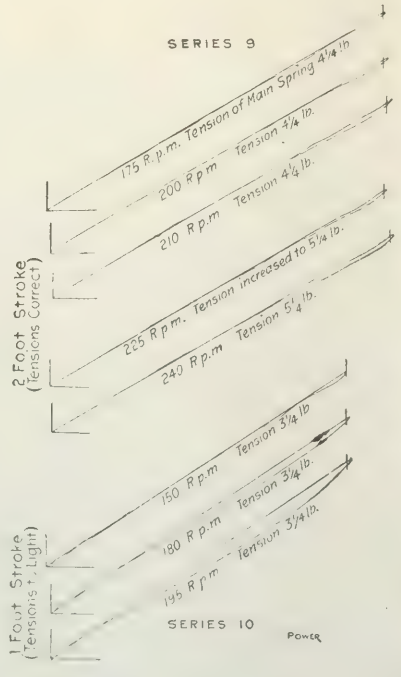

SERIES 9
175 R.p.m. Tension of Main Spring 4¼ lb.
200 R.p.m. Tension 4¼ lb.
210 R.p.m. Tension 4¼ lb.
225 R.p.m. Tension increased to 5¼ lb.
240 R.p.m. Tension 5¼ lb.
2 Foot Stroke (Tensions Correct)
150 R.p.m. Tension 3¼ lb.
180 R.p.m. Tension 3¼ lb.
195 R.p.m. Tension 3¼ lb.
1 Foot Stroke (Tensions Light)
SERIES 10
POWER

CHART 4, WHEEL NO. 4, INDICATOR "D"

and A, but as the drum spring is strong enough to meet the drum inertia at the highest speed of oscillation possible with the usual reducing wheel, the force a generally need not be considered.

If the wheel velocities are low so that inertia is not material, the frictional resistance to motion, if excessive, may cause a material lag of the drum behind its correct position throughout both strokes, since it effects a stretch of the cord upon the forward and a slackening upon the return stroke. This is shown by the lowest diagram of Series 2, Chart 1, the additional friction being introduced by a maladjustment of the wheel which might easily be made in practice. The thumbscrew holding the main wheel upon its spindle was turned a shade too far so that it cramped the bearing.

If the variation of stress in the cord is the same for different engine strokes, the distortion of the drum motion is the same. This is because any error in the motion of the wheel is reduced at the drum in the ratio of the drum travel to the wheel travel. For example, if the stresses are the same, the stretch of the cord on a 4-ft. engine stroke is double that of a 2-ft. stroke, since the cord is twice as long, but as the resulting drum distortion is one-twelfth of the stretch in the one case (drum stroke being equal to 4 in.) and one-sixth in the other, the effect is the same.

The results upon the other wheels are therefore presented for the short strokes only (although others were made), and these, it must be remembered, represent adverse conditions.

Wheel No. 2, although the same in general arrangement as No. 1, ran with excessive friction, partly because of the gears used, but mainly because of the arrangement for guiding the cord. On this account it could be run at no higher piston speed than 470 ft. per minute, 175 r.p.m., at a stroke of 16 in., without marked inertia effects. Chart 2 shows that the errors are very considerable, particularly when a long cord is used. It is noticeable that whipping occurred at a lower speed with the long cord; this is due to its weight overcoming the tendency to contract under the reduced external force. Noticeable, also, is that practically no overtravel occurs at any of the speeds except 190 r.p.m.

This wheel could be run at a piston speed of 720 ft. per minute at a 4-ft. stroke. At this speed, using a cord of 8 ft. maximum length, the errors were very much the same as the 80 r.p.m. diagram, Series 6, Chart 2.

The spiral or clock type spring increases its tension but little upon being wound. To accommodate a reducing wheel to higher piston speeds, either its friction must be decreased or its spring tension in the out position increased. The helical type spring is more readily adaptable to the latter requirement as its tension increases more rapidly with twist. From this it was foreseen that a wheel having such a spring would be more accurate at higher piston speeds, and accordingly Wheel No. 3 was tested. The results, Chart 3, show comparatively little error up to 200 r.p.m. or a piston speed of 800 ft. per minute at a 2-ft. stroke, and that it could be run as high as 920 ft. This wheel was new, which probably accounts for the distortion shown by Series 7 for the 16-in. stroke. The friction of the gears became less after a little use, so that Series 8 shows much more accurate results.

This wheel was driven by a fine fish-line, instead of indicator cord, which

was furnished by the wheel makers. The pitch of the cord-guiding device was such that regular indicator cord could not be used. The resilience and consequent stretch of a high-grade fishline, however, are not desirable for indicator work; and this line was found to be the poorest of many samples tested, having a stretch of 0.04 in. per foot per pound between 1 and 4 lb. But for this, Wheel No. 3 would have yielded still better results.

Wheel No. 4 showed different characteristics from the others, the limit to operation being controlled by the speed of engine revolution and not materially by the engine stroke, due to the peculiar combination of wheel and drum, for the indicator drum itself took the place of the usual wheel-spring barrel. The combined stroke and main wheel was so light and small that its inertia was insignificant compared to that of the drum which was particularly large and heavy. This combination gave unexpectedly good results, Chart 4, as it showed a nearly correct motion at a piston speed of 900 ft. per minute and a 2-ft. stroke (see Series 9, diagram at 225 r.p.m.) and could be run as high as 960 ft. This wheel's merit is due to the spring actuating both drum and wheel being large and strong, and to lack of friction, the gears and bearings running freely. The motion may be very erroneous, however, if the spring tension is too light, as shown by Series 10 for a 12-in. stroke. With the heavy tension and this stroke, the limiting speed of revolution was practically the same as with the 2-ft. stroke, and the errors were the same at the same speeds.

A general conclusion from the investigation is that the reducing wheel is reasonably accurate on short engine strokes and moderate piston speeds, and on long engine strokes at all usual piston speeds, *if properly designed to avoid friction.* The drum motion resulting from its use has its maximum error, when the friction is small, at a part of the stroke where usually it would affect the indicator diagram little if any, namely, at the beginning of the return stroke where either the steam line on the crank end or the exhaust line on the head-end diagram is described. As these lines are generally horizontal, or nearly so, errors upon them would not appear, except at their ends.

How to judge when the conditions of operation are such as to cause material distortion, cannot well be foretold unless the given wheel is tested by some such device as the drum-motion tester herein described. Overtravel of the drum motion does not indicate error with reducing wheels, as it is always small, exists for even low speeds when the motion is actually accurate, and may not exist at all when the friction is such as to cause large error, as shown by Chart 2. A vibrating cord should not be relied upon as a danger signal, for a speed causing large inertia errors will often not slacken the cord on account of its property to contract greatly under low values of a diminishing stress. This, however, applies more particularly to the short strokes, as the greater length of cord used otherwise, and its consequent weight, will cause vibration if it is not stressed with a force sufficient to accelerate the wheel.

The general rules to observe in operation are: Keep the wheel-spring tension as high as possible if its ability to accelerate the wheel is doubted. Amply lubricate, with watch oil, all moving parts of both drum and wheel and see that their motion is free from mechanical obstruction. Avoid guide pulleys, as these materially increase the friction, as was shown in the article on drum motion. Use only the best grade of cord, and be sure that it is the best not by trying its stretch with the unaided hand, but by measurement with a spring balance.

A stretchable cord, on account of its ability to contract as well as stretch, may prevent whipping under conditions that would slacken a stiff cord. This, however, is no reason for using it; whether or not the cord whips, the distortion exists, and it is desirable not to conceal the evidence of it.

A deficiency of wheel-spring tension may be corrected by tightening the drum spring to a limited extent only, because the drum spring acts upon the wheel at a mechanical disadvantage on account of the reduction of motion, and therefore has only a small effect to accelerate it. On the other hand, tightening the drum spring increases the reaction on the drum spindle, and therefore the friction to be overcome throughout the stroke, and this may cause even more distortion.

The faults of design of the wheels tested are two: First, the springs are not strong enough at short strokes, and are not properly proportioned in that instead of exerting a slightly increasing force as they are wound, they should furnish a much higher tension at the crank end than the head end. Second, friction is not avoided as much as it might be. This, however, is difficult. Wheel No. 3 has ball bearings, but on account of the gears and an unfortunate arrangement for connecting the stroke wheel to the drum, its friction is above the average. In conclusion, it is hoped that these tests will inspire confidence in the reducing wheel as an instrument of moderate precision and that they may indicate its limits of operation.

Iceland possesses a large amount of power in the shape of waterfalls. Quite recently a French syndicate purchased a large extent of ground in the neighborhood of Thorlakshaven, as well as several large waterfalls situated in the mountain regions in the interior. It is stated that the falls will be able to furnish as much as 200,000 hp.

Measuring Emulsified Oil in Condensation

Ordinarily it is not difficult to detect the presence of free oil in water, since given sufficient time, the oil will make its appearance on the surface. It is not always so easy to detect emulsified oil, which is nearly always present in the condensed exhaust from reciprocating engines, pumps and other apparatus where a good oil separator is not installed in the exhaust-steam line.

The Martin test tube, an instrument specially designed for this purpose, is made of clear glass, except at the bottom, which is opalescent or milky, with a small black dot in the center. The manner of using the instrument is to collect in a glass or cup a sample of the condensate

MEASURING OIL IN CONDENSATE

to be tested, and then taking the tube to a good light from a north exposure, that is, neither direct sunlight nor electric light, but rather to strong, diffused daylight, pour the water slowly into the tube, until the black spot just disappears from view. The graduation on the tube opposite the level of the water will show the amount of oil in emulsion, in grains per U. S. gallon.

The results obtained by these tubes have been repeatedly compared with analyses made by chemists and are reported to be substantially correct. The tubes are sold by the Harrison Safety Boiler Works, of Philadelphia, who report that they have proved very useful in several instances in ferreting out the sources of oil in boiler-feed supplies, in testing the efficiency of different types of oil separators in determining the degree to which different grades of cylinder oil emulsify, etc.

Taking Care of Bearings and Brasses

By J. C. Hawkins

Numerous methods are used by engineers in keying up and taking care of bearings, some of which are just guesswork, especially on crank and wristpin brasses. All bearings must have a little clearance to allow for expansion of the metal as it warms up, but the amount of play is often guessed at, sometimes with disastrous results. It is a good plan to calculate how much a box will be tightened by one turn of the wedge bolt, in driving the key ⅛ in., or by the thickness of a liner, such as is used on the marine type of crankrod.

It is much better to be sure of the adjustments before the start than to have to fight a hot bearing, or shut down and slack up on the wedge bolt later on. Fig. 1 shows the connecting-rod of a high-speed engine. Each wedge bears directly against one brass and is held by two adjusting bolts. To tighten the brasses the top bolt is loosened and the wedge is drawn down by the bottom bolt; the top one is then tightened. Assuming that the wedge has a taper of 10 deg., or 2⅛ in. per ft., and a 1-in. tap bolt with eight threads per in., drawing the wedge down 1 in. will close the brasses

$$2\tfrac{1}{8}\ in. \div 12\ in. = 0.177\ in.$$

One turn of the tap bolt will close the brasses

$$0.177 \div 8 = 0.0221\ in.$$

The cap bolt has a hexagonal head, and turning the screw one section will close the brasses nearly 0.0037 in.

In keying up this type of bearing, if the brasses and pin are in good condition, the writer loosens the top bolt and draws the wedge down tight, then slacks back about 0.004 in. for an 8-in. pin and 0.003 in. for a 4-in. pin, depending on the condition of the pin, the speed of the engine and the load. After the first trial one can usually tell just how much to slack back to get the best results. The wristpin is adjusted in the same way, but is given about 0.001 in. less clearance.

With the crankrod, Fig. 1, the cylinder clearance is not changed by the wear of the stationary brasses. In some rods of

Fig. 1. Rod for High-speed Engine

this type the wedge is placed back of the brass on the crank end, in which case the rod becomes longer with each adjustment, thereby reducing the head-end clearance. This wear is adjusted by placing shims of soft steel between the brass and the end of the slot in the rod on either end, but usually on the crank end.

In Fig. 2 is shown a rod using gibs and a key on the crank end and a wedge and bolts on the crosshead end. The key on the crank end has a taper of ⅜ in. in 1 ft. Driving this wedge 1 in. will close the brasses

$$0.375\ in. \div 12\ in. = 0.0312\ in.$$

In keying up brasses, place a rule across the wedge and gibs and make a mark

Fig. 2. Rod with Gibs, Keys and Wedge Adjusted

with a lead pencil, not with a scriber, which leaves a mark each time and is apt to be misleading. Then drive the key down tight and slack back to give the proper clearance, which would be $\frac{1}{16}$ in., $\frac{3}{32}$ in., or ⅛ in., depending on the size of the pin and the amount of clearance desired.

The inner brass is set against the end of the rod and the brass *A* is held by the strap. The gibs prevent the strap from spreading and in keying up the strap it is drawn further on the rod.

If the key drives all the way down, take it out and make a ⅛-in. steel feather to fit between the key and gib. A soft hammer should be used in driving keys to prevent upsetting the head. In keying up, turn the crank to the back center; this brings the wedges to the most accessible position. If the crankpin is not round, this position will cause the brasses to be adjusted on the largest diameter of the pin. To adjust the wristpin brasses, the crank would have to be at one-half stroke, thus making the top bolt inaccessible.

Fig. 3 shows a marine type of crankrod in which thin liners are used between the

Fig. 3. Marine Rod

brasses. There should be several liners in a set, one or two thick ones and several 0.002 in., 0.004 in. and 0.008 in. in thickness.

In this type of rod the thickness of the liner removed is the amount the brasses are set up. To adjust, loosen the bolt nuts and remove three or four liners, being sure to have the same number on each side. Then put in a piece of soft wire (fuse wire is good) and draw up tight against the pin, but do not spring the boxes. Then slack off and measure the thickness of the wire with a caliper, or, better still, with a micrometer and put in enough liners to make up this thickness plus the desired clearance. These liners should be made with a slot from the bolt-hole to the end so that they can be taken out without removing the bolts.

In some small vertical engines it is impossible to get at the crosshead brasses, in which case the pin is taken out after blocking up the crosshead; the crank is then turned to the end of the down stroke, so that the pin can be fitted to the brasses below the crosshead. In this type of rod the wear of the inner brasses will shorten the rod, which will decrease the crank-end clearance. This is remedied by placing a steel liner between the brass *B* and the end of the rod.

When taking up on the eccentric, the writer usually takes out a thin liner, say 0.003 in., and after tightening up, disconnects the eccentric rod from the valve-gear and moves it sideways, also up and down, to see if it is too tight. It is often necessary to do a little guesswork here, being sure not to take up too much at one time.

In keying up the valve-gear, the bearings are so small that very little clearance is required, and at the same time, an excessive amount in proportion to the size of the pin will do no harm. It is usual to set the brasses up tight, then slack off a little.

To key up the main bearings, the wear is usually taken up by some form of a wedge which is either drawn up or forced down between the quarter-box and frame.

As this bearing is stationary it is best adjusted when the engine is running just enough to make the bearing run quietly. Wear on the back quarter-box will throw the shaft out of alignment with the cylinder, and this will make the crankpin brasses bind and wear one-sided.

Modern engines have "spots" planed on the inside of the frame opposite the crank and parallel to the center line of the cylinder. Caliper between one of these spots and the crank, then turn the crank over until the same point on the disk comes opposite the caliper again. The difference, if any, will show how much the shaft is out of alignment. Then remove the main bearing wedge and push the shaft and back quarter-box ahead with a bar if the engine is small. On a large engine put the crank on the head-end dead center and admit a little steam through the bypass and then place thin liners between the back quarter-box and the frame. The quarter-boxes withstand the thrust of the piston and, therefore, wear very much faster than the bottom box, which only carries the weight of the shaft and wheel.

Any wear of the bottom box can be found by placing a level on the shaft, which will show whether the bearing is down or not. If it is very much out of plumb it should be raised and a thin liner placed under the bottom box.

In most modern engines the main bearing is so constructed that by raising the shaft an inch or two the bottom bearing

can be removed for examination or repair. In the center-crank engines the wear of the back quarter-box does not throw the shaft out of alignment with the crank; it decreases the head-end piston clearance. In placing liners behind the back quarter-boxes care must be taken to use the same thickness of liners for each to prevent throwing the shaft out of alignment. Some of the improved engines have a wedge under the bottom of the main bearing which allows the shaft to be raised without the use of liners.

As modern engines are so constructed that the bearings can be flooded with oil, run a stream of it to the crank, crosshead and main bearings, the used oil being drained from the bed, filtered, cooled and used over again. On the eccentric, compression grease cups are used unless the eccentrics are well inclosed, in which case oil can be used without throwing too much on the floor.

Grease is used exclusively on our governor bearings and links and valve-gear, as it is cheaper, cleaner and requires less attention.

Gas Explodes Tar Tank

By F. S. Hadfield

A tank used for storing crude tar at the plant of the Globe Soap Co., St. Bernard, Ohio, exploded, July 19, by the ignition of accumulated gas fired by a small flame used to melt out the tar draw-off pipe.

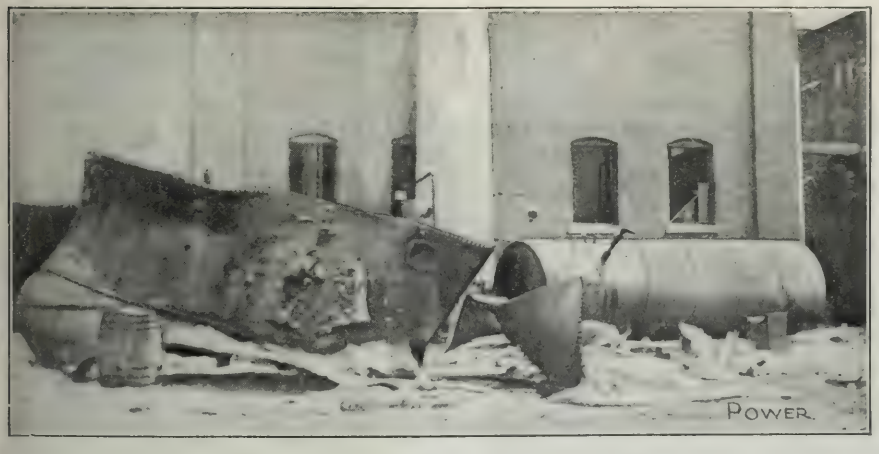


Fig. 1. Part of the Exploded Tank.

Fig. 2. Another View of the Shell

The tank was formerly an old return-tubular boiler, the tube sheets of which had been replaced by dished heads. The plates were originally $\frac{7}{16}$ in. thick but had wasted to $\frac{13}{32}$ in. The longitudinal joints were single-riveted, butt-strap, a peculiar joint, and the girth seams single-riveted lap joints.

From the direction of the explosion and the appearance of the plates the tank failed by the rivet holes pulling out in the longitudinal joints, the tank opening up at this point and straightening the sheets out practically flat, tearing themselves loose from the heads, which were blown about 20 ft. The top sheet containing the manhead was blown about 200 ft. and the manhead cover about 600 ft.

Although two men were working at the tank and several were near-by at the time, no one was injured.

Fig. 1 shows the exploded tank and another tank which was beside it. The second tank, full of tar, weighing some 30,000 lb., was lifted off its foundations and moved about 8 ft.

Fig. 2 shows the second tank and the wreck of the foundations, and Fig. 3 the rivet holes pulled out. The seam at the extreme top of the picture was one of the ends of the tank. The view also shows how the tank was unrolled and partially bent in the other direction.

The sheet that failed by pulling out of the holes was originally the fire sheet of the boiler and shows evidences of crystallization. The lap on these seams was 1½ in.

Fig. 3. Showing Torn Rivet Holes

The line of the sheet, indicated by the arrow, shows where the plate was ripped apart for some 10 ft. of its length, the fracture showing a rather crystalline structure.

S. & S. Stationary Tachometer

This stationary tachometer, manufactured by Schuchardt & Schutte, 90 West St., New York City, indicates on a dial the rates of revolutions per minute of the machine being tested.

The result, either revolutions per minute, speed per minute, periods, or miles per hour, is instantly shown without calculation or timing with a watch. Any variations occurring during a fraction of a minute in either right or left-hand revolutions or speeds are indicated, thereby enabling the engineer to control and regulate the speed of the machine.

The instrument is equipped throughout with ball bearings and operates by weights which are acted upon by centrifugal force counteracted by springs. The device is driven by pulley and belt or a spring coupling. The housing is of aluminum alloy.

The "Manual for Engineers," a handy flexible leather-bound pocketbook containing metric-conversion tables, cubes and roots, weights, logarithms, tensile strains, flow of steam, and other engineering data, may be obtained by writing to Greene, Tweed & Co., New York City, and inclosing 10c. in stamps.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Electric Motor Gear Drives

By John S. Leese

Operators frequently complain of the unsatisfactory performance of motors which are gear-connected to the driven shafts. Usually the motor manufacturer is undeservedly blamed for the noise and vibration, whereas the trouble is almost invariably due to the straight-cut gears being operated at excessive pitch-line speeds, setting up noise and vibration, the latter causing bad commutation. Upon looking into the subject carefully, the writer has found that generally the motor manufacturers had strenuously objected to furnishing high-speed motors and large gear reduction, offering instead slow-speed direct-connected machines.

A slow-speed motor for direct connection is slightly more expensive than a high-speed motor of the same output. First cost, however, is not the only consideration, and most experienced owners and engineers will prefer to pay a little more at the outset and avoid continued noise and constantly recurring troubles during operation.

There are five alternatives to straight-cut gearing for transmitting power to the driven element, that is, when the centers are too short for belts or ropes. These are worm gears, helical gears, double helical or herringbone gears, bevel gears and chain drive.

Worm Gears

The perfection of gear-cutting machinery and processes has led to the greatly enhanced efficiency of this means of power transmission and the worm gear is now fully as efficient as the straight-cut gear. Some end thrust occurs along the worm shaft, it is true, but this is provided for by some form of ball or roller thrust-bearing. Worm gears will wear well and be almost as silent after long service as at the beginning. There is nothing about a well cut worm gear to set up chattering or vibration; in fact, the writer has sometimes observed that their application has appeared to have a distinctly steadying effect on both the motor and the driven shafts.

That worm gears are not more widely used for transmitting power from electric motors may be due to their high first cost and suitability only for comparatively large speed reductions. The first objection prevails against the worm in most small motor applications, and the second makes it necessary, as a rule, to employ a very high motor speed and slow speed for the driven element. When this driven part is a line-shaft, either the size of the pulleys on it must be increased or the size of those on the driven machines decreased.

Single Helical Gears

Helical gears are much more expensive than straight-cut gearing, although fairly silent in operation at considerably higher speeds. This is because the teeth come into contact gradually and with less impact than do straight-cut teeth. On the other hand, helical gearing requires heavy thrust bearings, equal thrusts being imposed in opposite directions along both shafts.

Double Helical Gears

Double helical gearing may be used without noise or vibration with high pitch-line velocities and obviates the objectionable feature of thrust which is prevalent with simple helical gearing. Many drives of this class are in use for all sizes and high speeds. The sole objection to this class is that of first cost.

Bevel Gears

This type need not be considered for motor drive except in special cases. As a rule, it is noisier than straight-cut gearing and sets up more vibration, as well as putting end thrusts on both the driving and driven shafts.

Chain Drive

With silent chains applied to industrial drives, the driving and driven shafts must be very carefully aligned and where the power exceeds about 150 hp.,* the width of chain necessary renders such alignment difficult. When the chain exceeds 10 in. wide an outboard bearing for the motor shaft and closely located bearings for the driven shaft are usually required. Again, chains cannot, as a rule, be used for pitch-line speeds exceeding about 1250 ft. per min. with any more chance of success than straight-cut gearing will have at pitch-line speeds above 1000 ft. per min. This is because centrifugal force tends to throw the chain away from the sprockets at the high speeds.

For straight-cut gears a satisfactory way to reduce the pitch-line speed might seem to be to reduce the diameter of the driving pinion. This cannot be carried beyond a certain point for three reasons, namely: 1. The minimum limit to the number of teeth in the wheel must not be passed, and reduction in pitch involves widening the wheel. 2. The diametrical thrust on the shaft is rapidly increased as the diameter of the wheel is reduced. 3. The hole in the wheel boss limits the extent to which metal can be cut away with safety.

From the foregoing, it will be seen that for all ordinary circumstances and low powers, using straight-cut gearing with pitch-line speeds which exceed 1000 ft. per min. is disadvantageous. This does not mean that such speeds cannot be exceeded, for in a number of good drives it is; but in these cases care is taken to keep the gearing quiet, the loads are steady, and the shafts are particularly well supported. All that can be said definitely is that if the user specifies higher pitch-line speeds than 1000 ft. per min. for straight-cut gearing to use a cheap high-speed motor, he must not blame the motor manufacturer if noise and vibration result.

*Chain drives have been used successfully for transmitting considerably over 1000 hp., but different conditions prevailed from those herein discussed. Furthermore, Morse chains are operated at speeds of 2000 ft. per min.—Editor.

Lessons of Some Recent Electrical Fires

By H. S. Knowlton

Recent examination into the causes of a number of fires of electrical origin showed most of them to have been preventable. There are, of course, failures of insulation which cannot be anticipated without prohibitive expenditures for testing, but, on the other hand, a large percentage of the fires and accidents occurring yearly as a result of improper electrical conditions result from those operating plants or engaged in construction in close proximity to electric circuits failing to exercise the proper amount of care.

Loose connections are always fertile causes of trouble, as illustrated in the case of a low-potential arc lamp having a loose binding post. The contact was so poor that the insulation of the wire was ignited, and about 18 in. was burned off, charring the floor timber from which the lamp was suspended and threatening the entire building.

In another instance, a fire loss of over $10,000 resulted from placing an oily piece of canvas over a motor operating in a coal-hoisting tower. Sparks from

the contacts of the controlling switch were communicated to the canvas, and the adjacent woodwork was immediately set afire. The use of an asbestos guard or a rubber blanket covering would have prevented the trouble.

Again, careless handling of tools and metal around wiring often results in serious damages. In one case men working in a cellar of a building partially torn down brought an iron pipe into contact with the live terminals of a service switch, current passing from this pipe to a gas pipe in front of the building, melting a hole in the gas pipe and setting fire to the gas. A loss of $225 resulted from this apparently trifling carelessness in handling the pipe.

Dampness is always a foe to insulation, even on low-potential circuits. A slight fire occurred in one plant, due to the insulation of a wire in molding deteriorating on account of damp conditions. Electrolytic action took place, the wire becoming corroded to a fracture point, and when it broke, an arc established the incipient blaze.

Poor mechanical construction in supporting a steam-heating line led to the pipes falling across the service wires which were burned off, with subsequent melting of a hole in the steam piping and a small fire in clothing hanging nearby. In another case a staging used in the construction of a new building was brought in contact with a series arc-lighting wire; on account of rain the staging and the insulation of the wire were wet, and when the circuit was closed at night, arcing and burning occurred between the wire and the staging.

Combinations of circumstances which result in the starting of many fires are highly suggestive. In one instance a short-circuit occurred in the wires leading to a rheostat which had been temporarily placed above a large belt. The "short" was insufficient to blow the fuses or to open the circuit-breakers, but continued as a small arc with the result that several drops of melted copper fell upon the belt and were carried along with it and thrown among the papers on the chief engineer's desk near-by; the loose papers being set on fire and destroyed.

The wires of a city arc machine and the conduit from the magnetite arc system were so close on the basement ceiling that after whitewashing the ceiling the insulation became wet and allowed the current to jump across to the conduit, starting a fire. The trouble was discovered by flashing on the magnetite loop, which attracted the attention of the operating engineer, and immediately afterward smoke was seen coming from the basement. Sand extinguished the fire.

The floor of a power house was also set on fire by excessive heating of battery wires connected with an engine-stop device, the wires having become crossed with the local 500-volt service. The incident demonstrated the vital importance of separating low-potential circuits from lines carrying dangerous voltages and of avoiding all loose wiring in interior work unless of approved flexible construction.

Sand quickly extinguished another fire which was found under the ratchet of the slide-rail on a three-wire direct-current generator. The rails rested on foundations about 6 in. below the floor level, and the flooring had to be cut out to allow the ratchet to be worked. This open space had become filled with an accumulation of waste ravelings, sweepings, etc., and a spark from the brushes apparently fell into the hole and started a smouldering fire.

In another plant of rather out-of-date design a fire was caused by the commutator on a belted generator flashing over and setting fire to a transmission rope which ran directly above the commutator. The burning rope carried the fire to the oil-soaked floor and to the ceiling of the engine room; the floor and roof timbers being burned and scorched, and one transmission rope so badly scorched that it had to be removed. The entire plant was shut down for two hours. Later a noncombustible shield was placed between the commutator and the rope.

CORRESPONDENCE

Putting Generators on the Line

In the article on "Inspecting and Testing Electrical Apparatus," by A. L. Cook in the Aug. 6 issue under "Putting Generators on the Line," the author says "if a new compound or interpole generator will not take its share of the load, the machines should be tested separately to see that they give the same voltage for corresponding loads, and, if necessary, the shunts on the series coils should be adjusted so they will."

Replying to this, I would say that a shunt across the series field coils of any one machine also shunts the series coils of any other machines in parallel with it, and lowers the compounding, or rise in voltage with the load, of all the machines in parallel. To insure each machine taking its proper share of the load, the drop across the series field coils—in other words, the resistance of the series field coils—must be in proportion to the capacity of the machine as referred to the resistance of the field coils and the capacity of the remaining machines. If two generators of the same capacity or rating differ in the resistance of their series fields, the one having the least resistance will carry more than its share of the total current around its series field, thus building up a higher voltage than its mate and taking more than its share of the load. Adding a shunt across the series field terminals provides another path for the current and will lower the station voltage in proportion to the amount of current passed by the shunt.

The remedy in such a case is to place a resistance in series with the series field of right amount to cause both machines to have the same drop across the series field terminals. If the machines are of different capacity, the drop should be proportional to the capacity.

P. L. Werner.

McKeesport, Penn.

Mr. Werner's communication with reference to the adjustment of shunts on the series windings of compound machines, discusses a phase of the problem which the author intentionally omitted as it is a matter belonging more properly to the design of the wiring. Mr. Werner's statement that a shunt across the series field of one machine changes the compounding of all, should be qualified somewhat since the effect of the shunt is greatest upon the series field to which it is connected. As the shunts are connected directly at the terminals of the series coils and not at the busbars, there is an appreciable resistance between a shunt on one machine and the series windings of the other machines; this resistance being that of the cables which connect the machines to the busbars.

If two or more compound machines are to be run in parallel, it is essential, in the first place, that they have the same degree of compounding; in other words, that they give the same voltage at corresponding loads. This is secured by running the machines separately and adjusting the series shunts. With machines of similar design and with a proper layout of main cables, the machines should then operate satisfactorily in parallel. Assuming the shunts to be adjusted to give the proper degree of compounding when operated separately, then in order that the machines may run properly in parallel, the resistance of each circuit between the positive and the equalizer busbars through the series winding and shunt, if there is one, must be inversely proportional to the rating of the machines; this is so that each series winding may take its proportion of the total current. In some cases this requires additional resistance inserted in the leads between the machine and the busbars if the difference in the lengths of the leads is considerable for the different machines.

This is generally most easily accomplished by inserting an extra length of cable in the machine with the short leads. A matter like this, however, is supposed to be taken care of in planning the station wiring and would not properly be included in an article on electric testing. The writer considers this method of adjusting compound machines preferable to placing a resistance in series with the series winding of the machine.

A. L. Cook.

Montreal, Can.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Before and After Taking in a Charge

By Cecil P. Poole*

Figures intended to tell what happens to a gas and air mixture inside an engine cylinder are easy to work out; but figures that tell what really does happen never have been worked out accurately, unless by accident.

Changes of temperature within a cylinder must be largely guessed at, the guess modified by an estimate based on meager experience and the net result viewed with suspicion, because no thermometer yet made will record both instantaneously and accurately and also withstand the high temperatures occurring in a gas-engine cylinder.

While getting into the cylinder the gas and air are subjected to two influences which produce uncertain results; throttling, due to friction in passing through the passages leading into the cylinder, and heating, due to contact with the cylinder walls, with the hot spent gases left in the clearance, etc. Both reduce the weight of air and gas actually taken into the cylinder. It is frequently stated that the volume is reduced, but this is not true; the volume, figured back to atmospheric pressure is less than the number of cubic inches of space in the cylinder, but the actual volume is the same as the space in the cylinder. This expansion lowers the pressure so that at the end of the suction stroke it nearly always is slightly below atmospheric pressure.

A "stop" indicator diagram shows very clearly this drop in pressure due to throttling; Fig. 1 is such a diagram and indicates an absolute pressure of 13¼ lb. per sq.in. in the cylinder just before the piston starts on the compression stroke. The pressure first drops sharply as the piston starts on the suction stroke, producing a "bag" in the pressure line between *A* and *B*; then it rises gradually to the end of the stroke.

The fluctuations in pressure during admission are due to a combination of inertia and heating. When suction begins, the inertia of the mixture prevents it from entering instantaneously; therefore the pressure in the cylinder falls sharply, the burned gases in the clearance expanding as the piston moves forward. As soon as the difference between the pressure in the cylinder and that of the atmosphere is enough to overcome the inertia of the mixture, it enters with a rush. As the piston speed increases toward the middle of the stroke, the velocity of the incoming charge increases also, and when the piston has passed mid-stroke and its speed decreases, the inertia of the moving charge tends to keep up the velocity and there is a slight piling up behind the piston which raises the pressure slightly toward the end of the stroke. This is shown by the gradual rise of the suction pressure line from *B* to *C*, and the sharper curve from *C* to *D*, Fig. 1. The heating of the charge also helps raise the pressure; hence it is wrong to suppose that the difference between the pressure at the end of the suction stroke and atmospheric pressure is an accurate measure of the effect of throttling.

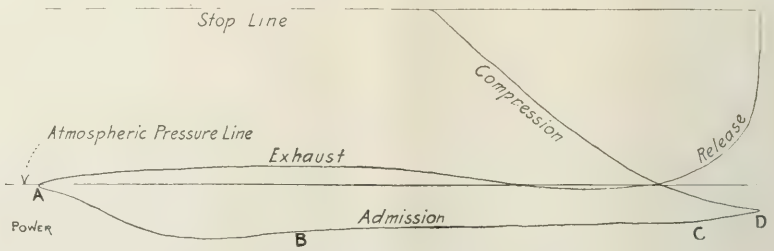


Fig. 1. A Representative Stop Diagram

The weight of mixture taken into the cylinder being less than the weight of the same volume at atmospheric pressure produces two unfavorable effects on the operation of the engine; it reduces the available energy in the mixture and the effective compression ratio. For example, suppose the piston displacement were 100 cu.in. and the clearance volume 25 cu.in.; then the total volume behind the piston before compression begins would be 125 cu.in., and when compression is complete, 25 cu.in., making the mechanical compression ratio

$$125 \div 25 = 5$$

With this ratio, under ordinary working conditions, the compression pressure would be about 8.1 times the pressure in the cylinder just before compression begins. Therefore, if the displacement efficiency were 100 per cent.—that is, if the piston drew in exactly 100 cu.in. of mixture during the suction stroke, making the pressure just before compression begins equal to atmospheric pressure (14.7 lb.)—the final compression pressure would be 119 lb. per sq.in., abs. If the mixture contained 2¼ B.t.u. per cu.in. at atmospheric pressure and temperature, and if its temperature were not changed during admission, the charge taken in would contain 225 B.t.u.

If the displacement efficiency were only 80 per cent., still assuming the temperature of the mixture constant during admission, the piston would draw in only 80 per cent. of the weight of mixture which occupied 100 cu.in. at atmospheric pressure, and the total B.t.u. taken in would be 180 instead of 225. The pressure in the cylinder at the end of the suction stroke (always assuming no change of temperature in the mixture) would be 11.8 lb. per sq.in. abs., instead of atmospheric pressure, and the piston would have to travel through 20 per cent. of its compression stroke before the pressure rose to atmospheric. Therefore, reckoning from the basis of

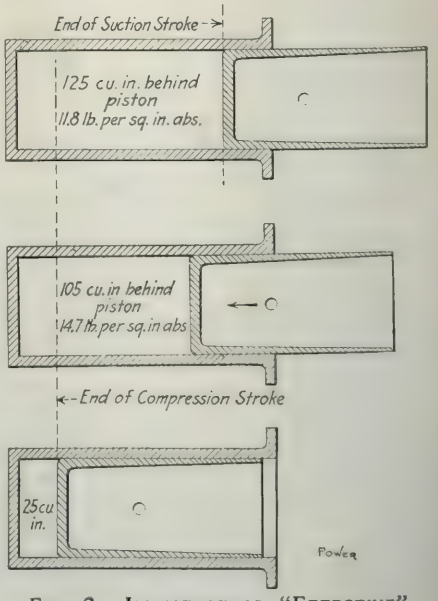

End of Suction Stroke
125 cu. in. behind piston
11.8 lb. per sq. in. abs.
105 cu.in behind piston
14.7 lb. per sq. in abs
End of Compression Stroke
25cu in.
Power

Fig. 2. Illustrating "Effective" Compression

atmospheric pressure, compression would really not begin until 20 per cent. of the compression stroke had been traveled. This condition is illustrated in Fig. 2.

*Consulting engineer, Atlanta, Ga.

The volume behind the piston just before the pressure began to rise above atmospheric would be 105 cu.in. instead of 125; consequently, the effective compression ratio would be

$$105 \div 25 = 4.2$$

instead of 5. With this ratio, the compression pressure would be about 6½ times the atmospheric pressure instead of 8.1 times, making the compression pressure 95½ lb. per sq.in. abs., instead of the 119 obtained in the first example, in which 100 per cent. displacement efficiency was assumed.

However, if a stop diagram is available, making known the pressure at the end of the suction stroke, the mechanical compression ratio (the volume behind the piston before compression begins divided by that when compression is completed) can be used, and it is sufficient to remember that the scale of pressures is lowered. That is, the pressure just before compression being lower than atmospheric, the pressure at the end of compression will be correspondingly lower than if no pressure had been lost during admission.

For example, starting from the reduced pressure of 11.8 lb. per sq.in. abs., at the end of the suction stroke with 80 per cent. displacement efficiency, and taking the full mechanical compression ratio of 5, the compression pressure would be

$$8.1 \times 11.8 = 95.6$$

practically the same as that computed on the basis of *effective* compression ratio and atmospheric initial presssure.

For simplicity the foregoing illustrations were based on the assumption of a constant temperature of the mixture during admission; but such is never obtained under normal running conditions; the temperature is always increased more or less above atmospheric, and this raises the pressure slightly above what it would be with the same displacement efficiency. The rise in temperature may be as small as 10 per cent., or as great as 25 per cent. of the absolute temperature of the atmosphere, or even more. Although the temperature in the cylinder cannot be measured accurately, approximate measurements have shown the temperature rise during admission ordinarily to be about 15 per cent. of the absolute temperature of the mixture before entering.

It is important to distinguish between the effects of throttling and heating. Both reduce the weight of gas and air taken in, but very differently; throttling does it by opposing the flow mechanically, heating by expanding the mixture. If it were not heated, the weight (and therefore the heat content) of the charge actually taken in would be equal to the weight (and therefore the heat content) of a cylinderful of mixture at atmospheric pressure multiplied by the displacement factor. (Often referred to as displacement efficiency.) But as there is a rise in temperature, some other factor must be used to apply to actual conditions, and this may appropriately be called the "intake factor." This is a number by which the theoretical capacity of the cylinder must be multiplied to get the actual weight or heat content of gas and air taken in per stroke under operating conditions.

For example, suppose the piston displacement were 100 cu.in. and that quantity of mixture contained 200 B.t.u. at atmospheric pressure and temperature. Also suppose that the intake factor of the engine were 77 per cent.; then the piston would draw in 77 per cent. of 200, or 154 B.t.u. at each full suction stroke.

If the rise in temperature during the suction stroke could be ascertained, this, with the pressure in the cylinder just before compression begins, would enable one to determine with sufficient accuracy for all practical purposes, the intake factor or efficiency of the engine; that is, the proportion of the theoretical weight of mixture that the piston actually draws into the cylinder. Suppose, in the example just given, the intake factor had not been known and temperature rise had been 17 per cent. with the pressure at the end of the suction stroke 13 lb. Referring to Table 1, in the column headed 13 and in line with 17 per cent. temperature rise will be found the intake factor, 77 per cent.

The displacement factor of an engine can be determined practically by driving it from an external source at its rated speed, having it take in air from a small holder and measuring the air by the holder drop; this is simple and easy, but expensive with a large engine, unless a holder happens to be at hand and the plant idle. But the displacement factor does not cover the rise in temperature during the entrance of the mixture into the cylinder, and there is no direct relation between it and the intake factor, which does.

The intake factor can be determined experimentally by operating an engine at full load and supplying it with gas and air taken from calibrated holders or delivered through meters for a certain length of time during which the number of strokes made by the piston are counted. Multiplying the piston displacement in cubic feet per stroke by the number of strokes made during the observation period and dividing this product into the total number of cubic feet of mixture taken in during that period will give the intake factor. For example, suppose an engine with a cylinder 10x16½ in. were running at full load and making 200 r.p.m. The piston displacement per stroke would be ¾ cu.ft. (approximately) and, assuming that the engine worked on the four-stroke cycle, it would make 100 suction strokes per minute, or 6000 per hour; the piston displacement per stroke multiplied by the number of suction strokes per hour would be 4500. Now suppose that the gas and air meters or holders showed that a total of 3300 cu.ft. of gas and air was delivered to the engine in one hour. Then the intake factor would be

$$3300 \div 4500 = 0.7333$$

or 73.33 per cent.

In considering the behavior of gases, absolute temperatures must be used. (The absolute temperature of a gas is 460 deg. greater than the Fahrenheit thermometer temperature, because absolute zero is 460 deg. below the Fahrenheit zero.) Therefore, in speaking of the percentage of temperature rise during admission, the percentage of the *absolute* temperature of the atmosphere is meant. For example, if the atmosphere is at 80 deg. F., or 80 + 460 = 540 deg. absolute, and the temperature of the mixture increases 17 per cent. during admission (17 per cent. of 540 = 91.8 deg.) the absolute temperature at the end of the suction stroke will be 540 + 91.8 = 631.8 deg. or practically 632, and the thermometer temperature will be 631.8 — 460 = 171.8 deg. Table 2 gives the Fahrenheit tem-

TABLE I. SHOWING RELATION BETWEEN FINAL INTAKE PRESSURE, TEMPERATURE RISE AND INTAKE FACTOR

(Figures in Body of Table are Intake Factors, Expressed in Percentage.)

Per cent. absolute temperature rise	Absolute Pressure at End of Suction Stroke														
	12	12¼	12½	12¾	13	13¼	13½	13¾	14	14¼	14½	14.7	15	15¼	15½
10	74.2	75.8	77.3	78.8	80.4	81.9	83.5	85.0	86.6	88.1	89.7	90.9	92.8	94.3	95.9
11	73.5	75.1	76.6	78.1	79.7	81.2	82.7	84.3	85.8	87.3	88.9	90.1	91.9	93.5	95.0
12	72.9	74.4	75.9	77.4	79.0	80.5	82.0	83.5	85.0	86.6	88.1	89.3	91.1	92.6	94.1
13	72.2	73.7	75.2	76.8	78.3	79.8	81.3	82.8	84.3	85.8	87.3	88.5	90.3	91.8	93.3
14	71.6	73.1	74.6	76.1	77.6	79.1	80.6	82.1	83.5	85.0	86.5	87.7	89.5	91.0	92.5
15	71.0	72.5	73.9	75.4	76.9	78.4	79.9	81.3	82.8	84.3	85.8	87.0	88.7	90.2	91.7
16	70.4	71.8	73.3	74.8	76.2	77.7	79.2	80.6	82.1	83.6	85.0	86.2	88.0	89.4	90.9
17	69.8	71.2	72.7	74.1	75.6	77.0	78.5	80.0	81.4	82.9	84.3	85.5	87.2	88.7	90.1
18	69.2	70.6	72.1	73.5	74.9	76.4	77.8	79.3	80.7	82.2	83.6	84.8	86.5	87.9	89.4
19	68.6	70.0	71.5	72.9	74.3	75.7	77.2	78.6	80.0	81.5	82.9	84.0	85.7	87.2	88.6
20	68.0	69.4	70.9	72.3	73.7	75.1	76.5	78.0	79.4	80.8	82.2	83.3	85.0	86.5	87.9
21	67.5	68.9	70.3	71.7	73.1	74.5	75.9	77.3	78.7	80.1	81.5	82.6	84.3	85.7	87.1
22	66.9	68.3	69.7	71.1	72.5	73.9	75.3	76.7	78.1	79.5	80.9	82.0	83.6	85.0	86.4
23	66.4	67.8	69.1	70.5	71.9	73.3	74.7	76.1	77.4	78.8	80.2	81.3	83.0	84.3	85.7
24	65.8	67.2	68.6	69.9	71.3	72.7	74.1	75.4	76.8	78.2	79.5	80.7	82.3	83.7	85.0
25	65.3	66.7	68.0	69.4	70.7	72.1	73.5	74.8	76.2	77.5	78.9	80.0	81.6	83.0	84.4

peratures in the cylinder corresponding to different percentages of absolute temperature rise. The numbers in the left-hand column are Fahrenheit thermometer temperatures of the mixture before entering.

One further example should suffice to make clear the application of the tables. Let the temperature of the atmosphere be 70 deg. (the atmospheric pressure being 14.7 lb. abs.), the pressure in the cylinder at the end of the suction stroke 13½ lb., and the intake factor of the engine as found by experiment, 80 per cent. What is the temperature in the cylinder at the end of the suction stroke?

Referring to Table 1, follow down the 13½-lb. column to the number 80, which is the intake factor; then following across to the left-hand column, 17 will be found as the percentage of absolute temperature rise. Now locate in the left-hand column of Table 2 the atmospheric temperature (70 deg.) and trace across until the 17 per cent. column is reached; in this column is the number 160, which is the thermometer temperature of the mixture in the cylinder.

Referring to Table 1 again, it will be noted that the range of absolute pressures at the end of the suction stroke is carried above atmospheric pressure. This is for the purpose of making the table apply to those infrequent cases in which the temperature rise is so great that the final suction pressure in the cylinder is above atmospheric pressure.

These two tables will be found interesting, if not of highly practical value, in showing the relation between the pressure drop and temperature rise during admission, and the intake factor of an engine. The relation is not rigidly accurate as computed, because the figures are based on the law of perfect gases (of which there are none), but they are more nearly accurate than the average observations with an indicator and a pyrometer.

TABLE 2. THERMOMETER TEMPERATURES IN CYLINDER CORRESPONDING TO DIFFERENT OUTSIDE TEMPERATURES AND ABSOLUTE TEMPERATURE INCREASES DURING ADMISSION

(Numbers in Body of Table are Thermometer Temperatures in Cylinder at End of Suction Stroke)

Air Temp. at Intake Deg. F.	Percentage of Absolute Temperature Rise During Admission to the Cylinder															
	10%	11%	12%	13%	14%	15%	16%	17%	18%	19%	20%	21%	22%	23%	24%	25%
40	90	95	100	105	110	115	120	125	130	135	140	145	150	155	160	165
41	91	96	101	106	111	116	121	126	131	136	141	146	151	156	161	166
42	92	97	102	107	112	117	122	127	132	137	142	147	152	157	162	168
43	93	98	103	108	113	118	123	128	134	139	144	149	154	159	164	169
44	94	99	104	110	115	120	125	130	135	140	145	150	155	160	165	170
45	95	101	106	111	116	121	126	131	136	141	146	151	156	161	166	171
46	97	102	107	112	117	122	127	132	137	142	147	152	157	162	167	173
47	98	103	108	113	118	123	128	133	138	143	148	153	159	164	169	174
48	99	104	109	114	119	124	129	134	139	145	150	155	160	165	170	175
49	100	105	110	115	120	125	130	136	141	146	151	156	161	166	171	176
50	101	106	111	116	121	127	132	137	142	147	152	157	162	167	172	178
51	102	107	112	117	123	128	133	138	143	148	153	158	163	169	174	179
52	103	108	113	119	124	129	134	139	144	149	154	160	165	170	175	180
53	104	109	115	120	125	130	135	140	145	150	156	161	166	171	176	181
54	105	111	116	121	126	131	136	141	147	152	157	162	167	172	177	183
55	107	112	117	122	127	132	137	143	148	152	158	163	168	173	179	184
56	108	113	118	123	128	133	139	144	149	154	159	164	170	175	180	185
57	109	114	119	124	129	135	140	145	150	155	160	166	171	176	181	186
58	110	115	120	125	131	136	141	146	151	156	162	167	172	177	182	188
59	111	116	121	126	132	137	142	147	152	158	163	168	173	178	183	189
60	112	117	122	128	133	138	143	148	154	159	164	169	174	180	185	190
62	114	119	125	130	135	140	146	151	156	161	166	172	177	182	187	193
64	116	122	127	132	137	143	148	153	158	164	169	174	179	185	190	195
66	119	124	129	134	140	145	150	155	161	166	171	176	182	187	192	198
68	121	126	131	137	142	147	153	158	163	168	174	179	184	189	195	200
70	123	128	134	139	144	150	155	160	165	171	176	181	187	192	197	203
72	125	131	136	141	146	152	157	162	168	173	178	184	189	194	200	205
74	127	133	138	143	149	154	159	165	170	175	181	186	191	197	202	208
76	130	135	140	146	151	156	162	167	172	178	183	189	194	199	205	210
78	132	137	143	148	153	159	164	169	175	180	186	191	196	202	207	213
80	134	139	145	150	156	161	166	172	177	183	188	193	199	204	210	215
82	136	142	147	152	158	163	169	174	180	185	190	196	201	207	212	218
84	138	144	149	155	160	166	171	176	182	187	193	198	204	209	215	220
86	141	146	152	157	162	168	173	179	184	190	195	201	206	211	217	223
88	143	148	154	159	165	170	176	181	187	192	198	203	209	214	220	225
90	145	151	156	162	167	173	178	184	189	195	200	206	211	217	222	228
92	147	153	158	164	169	175	180	186	191	197	202	208	213	219	225	230
94	149	155	161	166	172	177	183	188	194	199	205	210	216	221	227	233
96	152	157	163	168	174	179	185	191	196	202	207	213	218	224	229	235
98	154	159	165	171	176	182	187	193	198	204	210	215	221	226	232	238
100	156	162	167	173	178	184	190	195	201	206	212	218	223	229	234	240
102½	159	164	170	176	181	187	193	198	104	209	215	221	226	232	238	243
105	162	167	173	178	184	190	195	201	207	212	218	224	229	235	241	246
107½	164	170	176	181	187	193	198	204	210	215	221	227	232	238	244	249
110	167	173	178	184	190	196	201	207	213	218	224	230	235	241	247	253

Minor Defects in Suction Gas Power Plants

By James A. Seager

It is not proposed here to discuss either the gas producer or the engine, but to consider some of the accessories and their minor troubles. For instance, in the wet form of scrubber it is most important that attention be paid to the joints on the clean-out doors. In one instance, leakage occurred past the flanges on the bottom door and resulted in corrosion of the metal. As the water was charged with impurities this corrosion proceeded so rapidly that it was decided to paint the exposed surfaces with anti-corrosive paint. It was thought that this would settle the difficulty, but after 12 mo. it was found that the corrosion had extended to such an extent that it became necessary to thoroughly scrape the metal and cover it with a thin layer of cement.

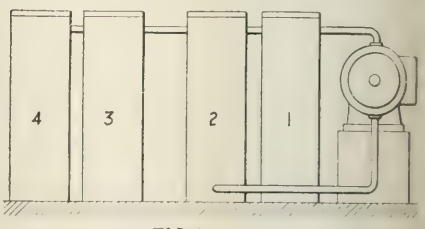


FIG. 1

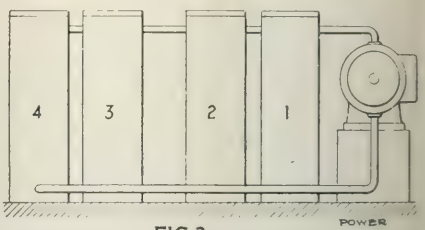


FIG. 2

Incorrect and Correct Connections

Among other minor troubles may be mentioned those occurring in the piping. In one case the exhaust pipe became cracked, due to the hot gases causing an old defect to open. An attempt was made to keep the gases from escaping into the room by putting clamps around the pipe, but this proved ineffective. Therefore, to cure the trouble, a sheet of asbestos packing was softened in water and after becoming flexible it was tightly clamped over the crack.

Many pipe-work troubles occur in connection with the engine cooling-water system; for instance, in a certain suction-gas installation the circulating water was at first cooled by the aid of two cylindrical tanks in series. When the engine was fully loaded, however, these proved inadequate and two similar tanks were added, and connected as in Fig. 1. Tanks 1 and 2 formed the original installation and formed a complete circulating unit, but in connecting tanks 3 and 4 no provision was made for returning the water to the engine; hence the original circulation remained unchanged. The defect was finally corrected by connecting the tanks as in Fig. 2.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Influence of Suction Pressure on Ammonia Compressors*

By Robert H. Karl†

An ammonia compressor is constructed to take in ammonia gas at a low pressure and temperature and discharge it at a higher pressure and temperature. The size and speed of the compressor, or strictly speaking, its piston displacement, is the measure of the capacity of refrigeration.

To say that a machine has a capacity of so many tons of refrigeration means that a certain quantity of liquid ammonia circulated by it will, by evaporating, extract as many heat units from the brine tank as will melt so many tons of ice at 32 deg. F. into water at 32 deg. F. This capacity measurement is not constant and varies considerably under different conditions of operation. A 50-ton machine may produce only 30 tons of refrigeration under certain conditions, while under more favorable conditions it may produce 70.

If a compressor discharges the greatest volume of ammonia gas for a given expenditure of work, that is all the machine can do. The work expended depends on the pressure of the gas entering the compressor, and the pressure against which the gas must be discharged.

A pound of ammonia can either absorb or give off a certain amount of heat by changing from a liquid to a gas or from a gas to a liquid. This transition can occur through a great range of pressure, and since a cubic foot of ammonia gas increases in weight when its pressure increases, and therefore a greater weight of gas can be handled by the same refrigerating machine at higher suction pressure, it is evident that more refrigeration can be performed. Since the amount of work that must be expended increases with the pressure against which the gas must be discharged, the lower this pressure, the less work is required. In other words, the smaller the difference between the condenser and suction pressures the greater the capacity and economy. The suction pressure is governed by the temperature and pipe surface of the refrigerator, and the condenser pressure by the cooling surface and the temperature and quantity of the cooling water. These conditions should be given proper consideration when the plant is installed, and because this is seldom the case many refrigerating plants suffer in capacity and economy. Sometimes for the sake of a few dollars worth of piping, the entire plant is crippled, while on the other hand the compressor may be bought entirely too large or too small.

It may be interesting to see how the pressure of the gas as it passes from the cooling coils to the compressor cylinder, exerts such a marked influence upon the capacity and also upon the economy of the refrigerating machine. The formula for capacity is:

$$F = 0.00505\ N\ M\ [R - (T - t)]$$

where

F = Ice melting capacity in tons per 24 hr.;
N = Strokes per minute;
M = Weight of ammonia per stroke;
R = Latent heat at refrigerator pressure;
T = Temperature in condenser;
t = Temperature in freezing coils;
0.00505 = Reciprocal of B.t.u. per minute per ton of refrigeration.

Since the number of strokes remains constant, it is evident that the capacity depends upon the weight of ammonia, latent heat at the refrigerator pressure and the temperatures in the condenser and the freezing coils.

A table of properties of ammonia, will show that as the suction pressure increases, the corresponding latent heat becomes less, and the corresponding temperature or heat of the liquid rises; consequently the sum of the latent heat and the heat of the liquid in the condenser and freezing coils changes but little with different suction pressures, so that the capacity is practically proportional to the weight of ammonia circulated. For instance, at a suction pressure of 18.45 lb. abs., a cubic foot of gas weighs 0.0689 lb., while at 38.55 lb. abs. pressure it weighs 0.1384 lb., or twice as much.

As the same volume of ammonia gas passes through the compressor in a given time, whatever the pressure, then with the greater suction pressure the machine will circulate double the amount of ammonia and the capacity will be nearly double that at the lower pressure.

It requires more power to operate the compressor at a higher suction pressure, but the power does not increase as rapidly as the capacity. The power is proportional to the heat delivered to the condenser minus the heat taken from the cold body, and the capacity is proportional to the heat taken from the cold body.

The examples following may serve to make things a little clearer. For simplicity, take a saturated gas compressor, 12x24-in. and 70 r.p.m., that works without superheating, the suction pressure to be 10.31 lb. gage and the condenser pressure 200 lb. gage. Each cubic foot of ammonia at the suction pressure would weigh 0.0921 lb., and each stroke of the piston would displace 1.57 cu.ft. of gas weighing 0.1446 lb. Then assuming an efficiency of 80 per cent.

$$F = 0.00505 \times 140 \times 0.1446\ (560.39 - 108) \times 0.80 = 37\ tons$$

requiring about 76.2 hp. The machine would thus fall short about 13 tons of the supposed capacity. If the same compressor were operated at the same speed with a suction pressure of 27.28 lb. gage and a condenser pressure of 170 lb. gage the weight of ammonia discharged per stroke would be 0.2366 lb., and the capacity

$$F = 0.00505 \times 140 \times 0.2366\ (546.88 - 77) \times 0.80 = 62.88\ tons$$

requiring 83.2 hp.

The previous examples show the foolishness of buying a machine of 50 tons capacity without knowing under what conditions it will have to work, for this compressor will develop its rated capacity only with a suction pressure of 18.68 lb., and a condenser pressure of 180 lb. at the same speed and same efficiency. Thus

$$F = 0.00505 \times 140 \times 0.1903\ (553.04 - 90) \times 0.80 = 49.84\ tons$$

and the horsepower would be about 80.4.

The same compressor may, however, produce a great deal more. In the case of wort cooling in a brewery where the suction pressure is 44.72 lb. and the condenser pressure 180.43 lb. the weight of ammonia discharged per stroke is 0.3295 lb. and the capacity becomes

$$F = 0.00505 \times 140 \times 0.3295\ (536.91 - 64) \times 0.80 = 88.13\ tons$$

requiring 91.2 hp.

It is evident then that the suction pressure greatly affects the capacity even when the machine is running at the same speed. Take the case where the compressor is doing 50 tons of refrigeration with about 1.6 hp. per ton, and compare it with the first case where the same machine working at the same speed at a lower pressure requires 2 hp. per ton.

*Abstract of paper read before the Institute of Operating Engineers, New York City, Sept. 6 and 7.

†Chief erecting engineer, Anheuser-Busch Brewing Co., St. Louis, Mo.

The capacity is reduced 26 per cent. and the horsepower per ton increased 25 per cent.

Now compare the third with the second case, where the suction pressure is increased from 18.68 lb. to 27.28 lb. The capacity of the compressor is increased to 62.88 tons, an increase of practically 13 tons, or 26 per cent. The horsepower per ton has been decreased from 1.6 to 1.32, or 17.5 per cent.

By raising the suction pressure of 10.31 lb. in the first case to 27.18 lb. in the second, the capacity of the compressor was increased 70 per cent. at an expenditure of only 9 per cent. more power. In other words, the extra 25.8 tons of refrigeration required only 7 hp.

Setting the Expansion Valve

By C. E. Anderson

Among engineers operating refrigerating plants, opinions seem to differ concerning the proper setting of the expansion valve. Some claim it best to open wide the expansion valve for 10 or 15 min., depending on the size of the coils, flood the coils with liquid and then close the expansion valve and shut down the pump until the temperature of the ammonia gas returning to the machine is 10 or 15 deg. below the temperature of the cooler or brine tank, when the operation is repeated.

Others contend that the coils should be pumped down to a vacuum before the flooding begins. This works fairly well with rooms that are hard to cool, if it is used once or twice in 24 hr. and the expansion valve is adjusted for continuous feeding. With top expansion on the coils it is not satisfactory. Best results are obtained when the coils are fitted for bottom expansion.

Still others claim that no liquid should pass the expansion valve and be allowed to enter the evaporating coils. Where only gas is being admitted to the coils the latent heat of the liquid ammonia is entirely expended at the expansion valve and the coils do no work. As refrigeration is brought about only at the point of expansion, obviously the liquid must be admitted to the coils.

The first cooling done by the liquid ammonia is upon itself until it reaches the cooler temperature; hence the lower the cooler temperature the less useful refrigeration will be produced per pound of liquid ammonia. In every transfer of heat certain losses are unavoidable; first, because a temperature difference is necessary to cause a heat transfer, and it takes energy to maintain a temperature difference; second, to effect a heat transfer, requires certain apparatus, and with this heat is lost by absorption.

Assume the condensing water at 60 deg. F. and the condenser pressure 126.5 lb. gage, which gives the condensed ammonia gas, or the liquid, a temperature of 75 deg. F. The specific heat of liquid ammonia is variously given from 1 to 1.22. Take a mean value, 1.1. To reduce 1 lb. of ammonia liquid from 75 deg. to zero requires removing

$$75 \times 1.1 \times 1 = 82.5 \ B.t.u.$$

This only reduces the liquid ammonia temperature and not the room or brine temperatures. In practice this is done by opening the expansion valve and allowing the liquid ammonia to pass from a higher to a lower pressure, or, in this case, from 126.5 to 15 lb. gage, or from 75 to 0 deg. F. On reaching the lower pressure, the liquid will give up 82.5 B.t.u., and 82.5/556 of the pound of liquid ammonia will vaporize. The pound of ammonia is now in the cooling coils, 82.5/556 of it vapor and 473.5/556 lb. still liquid.

Suppose the evaporating coils are in a cold-storage room whose temperature is 10 deg., the ammonia temperature being zero. Every heat unit the evaporating coils absorb from the room will evaporate a definite amount of the liquid. When 473.5 B.t.u. has been absorbed, all of the liquid ammonia will be evaporated, because

$$473.5 + 82.5 = 556 \ B.t.u.$$

which is the latent heat of liquid ammonia at 15 lb. pressure or 0 deg. F.

Evidently, therefore, the liquid must pass through the expansion valve to the coils to derive the benefit of the latent heat, or one of the very important properties for which ammonia was chosen as a refrigerating agent will be lost.

CORRESPONDENCE

Data on Absorption System

Some data on absorption machines by Fred Ophuls, published in the issue of Apr. 23, has just come to the writer's attention. From a scientific standpoint the data is interesting and particularly that on moisture in rectifier vapors, but without a little study some of the conclusions drawn may be misleading.

Steam Consumption—A standard absorption machine has a steam consumption of from 30 to 40 lb per hr. per ton, depending upon conditions, making a total variation of 30 per cent., but Table 10 shows a variation from 32 to 58 lb., or 90 per cent.; hence the tests are not representative.

Rectifier Heat—In test No. 1, Table 9, the figures show a tremendous heat elimination in the rectifier, namely, 43,882 B.t.u., and only 47,603 for the absorber. This discrepancy accounts for the excess steam consumption. The following are some figures of test made by Prof. J. E. Denton on a well known absorption machine on July 1 and 2, 1897; duration 24½ hr:

Water inlet, 68.6 deg.; brine outlet, deg.	13.5
Steam consumption per hour per ton, lb.	32.76
Heat eliminated in rectifier, B.t.u.	290,264
Heat eliminated in absorber, B.t.u.	1,063,876
Per cent. rectifier of absorber heat	27

Test No. 4 more nearly approaches good practice. The rectifier heat is 30 per cent. of the absorber and the steam consumption is 32 lb. per hr. per ton, which is still high, considering the very favorable conditions.

Water Consumption—The illustration indicates that water flows separately over the condenser, absorber and rectifier. Under these conditions the consumption would be excessive. Condenser water should be used over the absorber and then the rectifier, so that the same water would do for all purposes. An absorption machine will use excessive water under the former conditions just as a compound engine will use extra steam if live steam enters each cylinder instead of running from one cylinder to the other.

It is practicable to run absorption machines with 1½ to 2 gal. of water per min. per ton. A compressor should have at least 2 gal. and 3 gal. is better. The tests described by Mr. Ophuls show water consumptions over double what is necessary in good practice.

Standard Practice—Results of tests on regular absorption exhaust-steam machines are produced in the following figures, which show what is to be expected from machines of this type under various conditions and quantities of water:

STANDARD PRACTICE FOR ABSORPTION MACHINES

Steam Press. Lb	Water Temp. Deg.F.	Outlet Brine Deg.F.	Gal. water per min. per ton	Steam required per hour per ton
3	60	8	2.2	29.9
3	60	0	2.4	32.4
3	65	8	2.4	29.9
15	60	—15	2.7	36.5
30	70	—15	3.2	40.2

(Ammonia pumps exhaust included)

Henry Torrance, Jr.

New York City.

A steam-heated pond is obtained by a lumber company owning a mill pond near the Oregon-California line. This was done that no interruption might be met in the work of the employees at that place. During the winter pipes injected steam into the pond. No ice is allowed to form, no matter how low the temperature may fall. In this way about 200 men are kept steadily at work in the logging camps, on the railroad from the timber, and in the sawmill. During the day the engines exhaust steam into the mill pond, and at night dry steam is forced into the water.—*Ambition.*

The Italian government has ordered from Burmeister & Wain, of Copenhagen, four boats for its navy, to be fitted with oil engines.

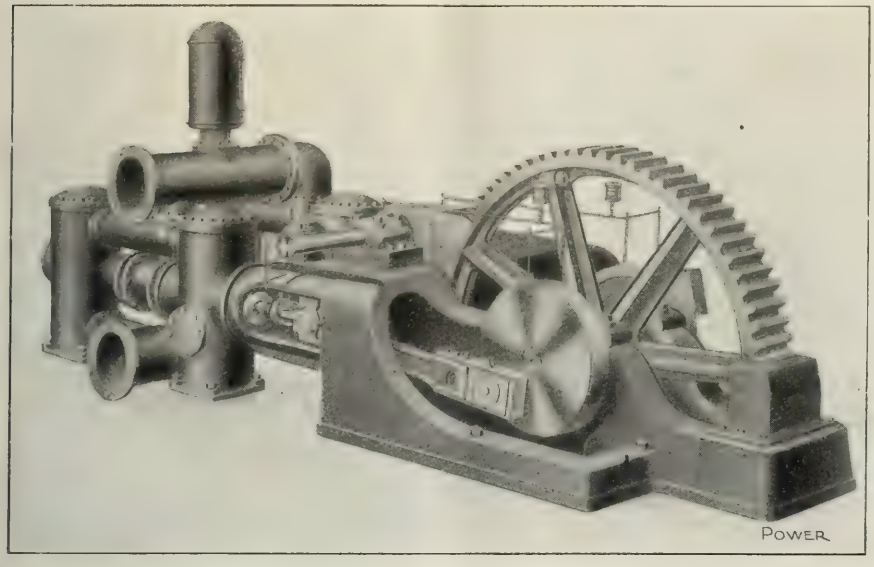
Large Capacity Duplex Pump

A new type of horizontal duplex double-acting power pump for large capacity service in water-works and isolated plants is being placed on the market by the Goulds Manufacturing Co., Seneca Falls, N. Y.

The pinion shaft is extended for direct connection to an electric motor, steam or internal-combustion engine, or other driver.

The power end has a single split cast-iron gear and forged-steel pinion. The teeth in both are machine cut from the solid metal. Three heavy cast-iron girder sections, each carrying a main bearing and crosshead, are doweled and bolted together to form the frame. The crossheads are cast iron with babbitted shoes which have wedge and screw adjustment. The guides are bored.

The water end consists of four castings, each containing a plunger, a discharge and a suction chamber. These castings are held together by large flanged suction and discharge connections, and are bolted to the main frame through the front cylinder.



GOULDS LARGE CAPACITY DUPLEX PUMP

Imperial Mechanical Lubricators

The imperial mechanical lubricators herewith illustrated are manufactured by the Imperial Brass Manufacturing Co., Harrison St. and Center Ave., Chicago. Model A, Fig. 1, is designed for pressures up to several thousand pounds, for all speeds up to 2000 r.p.m. and can be supplied with any number of individual feeds desired. Model B, Fig. 2, has a glass body and is made in 1-pt. size for small units.

Fig. 3 shows the construction of model A. The pump is operated by the ratchet *C*, which turns the cam *D* to which the crosshead *L* is connected. The amount of oil fed depends upon the length of the stroke of the pump plunger *J* which is adjusted by the nuts *M*. On the upstroke of plunger *J* a definite amount of oil is drawn from the suction passage *I* into the pump. This causes a partial vacuum in the oil well *N* and consequently an amount of oil equal to that drawn out by the pump flows from the reservoir into the oil well *N* through the supply tube *G*. On the down stroke of plunger *J* the oil passes through the discharge valve *K* and is forced from the lubricator through the oil pipe and through the pressure valve *F* to the point of lubrication.

This arrangement avoids having the sight-feed glass under pressure. All wearing parts are inclosed and work in a bath of oil. The delivery of oil being accomplished by a single-plunger pump the lubricator is simple and has a small number of parts. Another feature is the use of ball-and-socket joints *Y* on the transmission rod *X*, fitted with springs to take up wear. With this construction perfect alignment is not necessary.

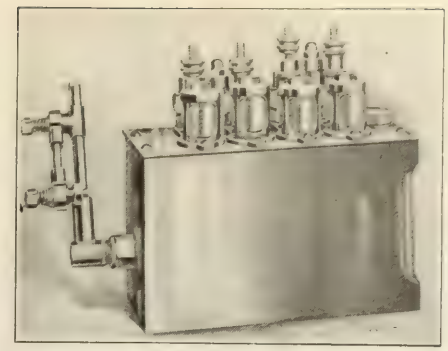


FIG. 1. IMPERIAL MECHANICAL LUBRICATOR, MODEL A

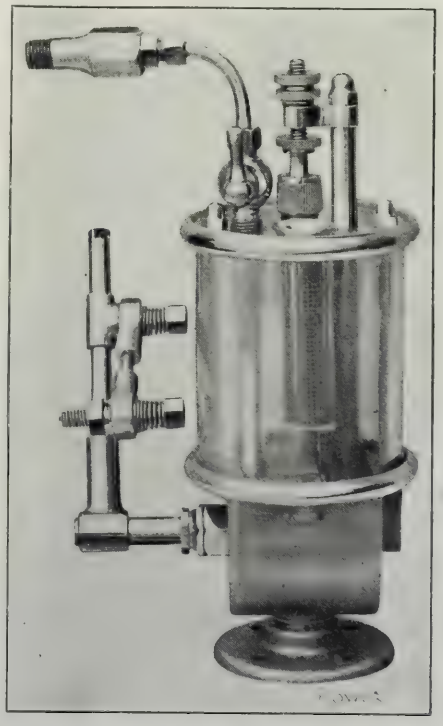


FIG. 2. IMPERIAL MECHANICAL LUBRICATOR, MODEL B

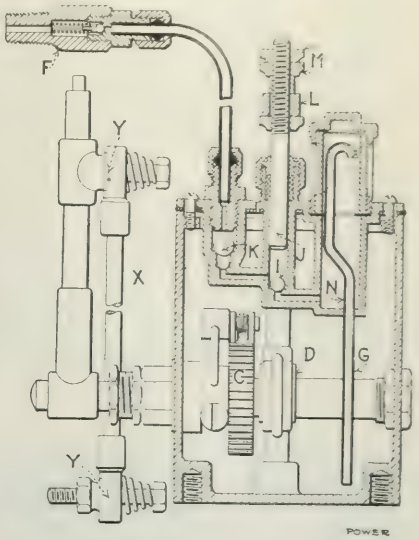


FIG. 3. VERTICAL SECTION OF MODEL A LUBRICATOR

Rucker Tube Blower

A tube blower, for use on return-tubular boilers, in which the steam is given its spiral motion through the tubes has been placed upon the market by the Rucker Manufacturing Co., 15 South First St., Minneapolis, Minn. Three jets of steam are blown through the tube, one from a central outlet in the head of the blower, and passes through the tube in a straight line. The other two jets are given a spiral motion, due to the form of the two tubes attached to the inner head and connecting with its steam supply. The blower is made in sizes from 1½ in. up to 4½ in.

Water Turbines for Panama Power House

By Fred H. Colvin

The illustrations show one of the turbines for the new power house at Gatun which will supply electric power for the locomotives used in handling vessels through the locks at both ends of the canal and also all other power required in the canal zone. There are three of these machines being installed at present, each turbine giving 3600 hp. at a 75-ft. head and a speed of 250 r.p.m. There are also provisions for doubling this installation so that a total of 21,600 hp. can be had if necessary.

Fig. 1. Setting Up First Unit

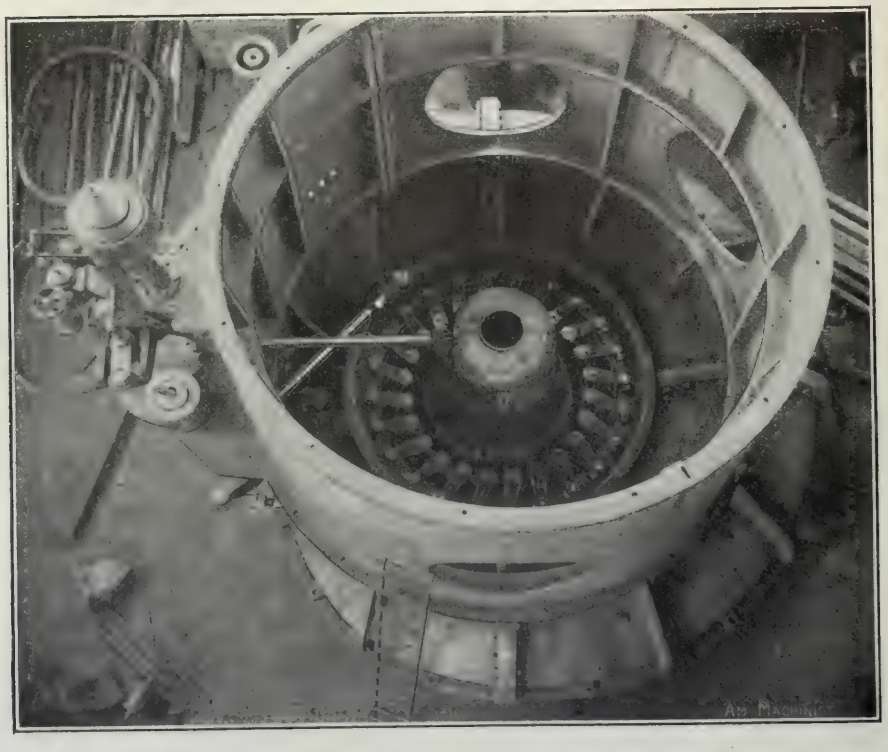


Fig. 2. Looking Down at the Wicket Governing Gates

The outside diameter of the casing is about 35 ft. and weighs between 30 and 32 tons. The total height is 21 ft. above the floor when in position in the power house, and the outlet at the bottom will be 3 ft. below this. The generator, made by the General Electric Co., is mounted at the top of the turbine and is 12½ ft. in diameter.

The first unit set up is shown in Fig. 1, which gives some idea of the size of the openings. The inlet is 80 in., being supplied by a pipe line 10 ft. 6 in. in diameter, and the outlet is 76 in. The speed of the turbine will be controlled by 24 wicket gates, whose controlling arms can be seen in Fig. 2.

Each of these wicket gates is 23 in. long, all being controlled from the governor by these arms, which move the outer rings to which the wicket cranks are connected, and in this way increase or decrease the flow of water to the bronze runner. This runner is of government bronze and weighs 7500 lb.

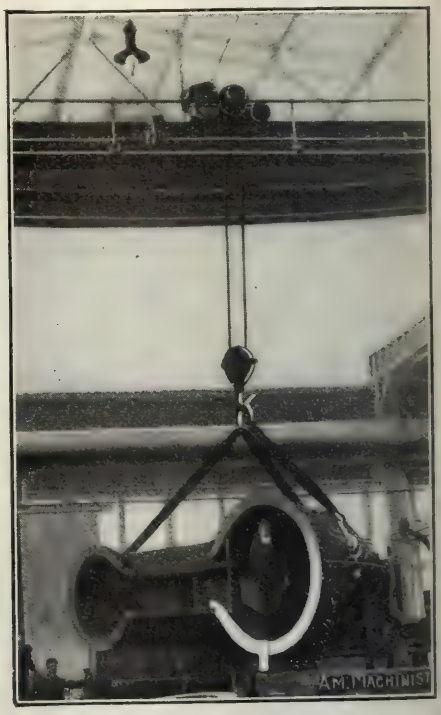


Fig. 3. Showing Size of Casing

Part of the annular casing is shown in Fig. 3, where the piece is swung from the 15-ton crane and a good-sized man is standing in the bore.

The turbines are the design of the Pelton Water Wheel Co., and were built by the Harrisburg Manufacturing & Boiler Co., Harrisburg, Penn.

Not only did James J. Farrell, the engineer who saved the lives of six men by sticking to his post while a heavy electric current passed through his body, display admirable courage, but his presence of mind is equally worthy of note. His devotion and fortitude were typical of the untold instances of intelligent self-sacrifice that enter into the daily life of men who make no pretense to being cast in a heroic mould.—*New York Sun.*

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POWER

Treatment of Boilers

Boiler abuse in small steam plants is more prevalent than in large ones, for several reasons. In large plants the men in charge are, on the average, of a higher grade, have a better understanding of the effects of dirt in boilers, deteriorating effect of expansion and contraction, caused by changes in furnace temperatures and know what to do and what not to do. The methods employed in caring for boilers by the men in the two classes of steam plants vary as conditions dictate.

Often, men in small steam plants are forced to do things they know are detrimental to the life of the boiler, but their better judgment counts for naught, repairs must be made and the boiler must be put into service again, when there is not time to properly do the work.

In one instance, one of two boilers demanded partial retubing. The boiler was in service from 4 p.m. until 7:30 a.m. daily. During the intervening period it was cooled, emptied, one or two tubes removed and replaced by new ones, refilled and steam raised to the working pressure. As the tubes were badly scaled it was slow work removing them, and the eight hours in which the boiler was out of commission was not enough in which to do the work.

As it was the boiler was emptied under pressure, giving it no chance to cool, and was filled with cold water so late that proper time could not be given for slowly heating the water and generating steam. Frequently steam was raised to the working pressure from a cold boiler in less than 90 minutes.

Poor practice certainly, and the work was done against the engineer's better judgment. It was a case of obeying orders or of cutting out one-half of the lighting load when it was most needed. This much is evident, the boiler was injured by the abuse and its life materially shortened.

As a contrast, it is the practice in some power plants to cut out a boiler by closing the valve in the connection between it and the steam main and allow the brickwork and boiler to cool naturally, by letting the fire gradually die out with all draft doors and damper closed. When the boiler is to be put into service again, the fires are started and made to burn slowly for several hours before steam is raised.

Quite a difference in the two methods employed, and after a service of a few years the boilers would speak for themselves, one condemning, the other commending the treatment received.

Friendliness and Familiarity

We give space to the communication under this title on page 466 not because we sympathize with the sentiments it expresses, but because we do not wish to be accused of refusing a critic the chance to present his views.

In a certain manufacturing establishment the memory persists of a highly polished self-important individual thoroughly imbued with the ideas of rank, caste and discipline, which he had contracted at Annapolis, who became its chief executive. One day, as he was showing a party of visitors over his little realm the superintendent of one of the departments spoke to him pleasantly about some matter of shop interest. The calling down which that superintendent got is still one of the shop traditions. In ringing tones he was reminded that he was a subordinate, an inferior, and that never again was he to approach and address his supreme officer in so informal a manner.

This man was universally regarded as a prig and a snob, and outside of a few bread-and-butter flunkies who took his part and spoke him fair, but who would have kicked him over without compunction if his fall would have served their interests, there was not a man in the shop who would not have thoroughly enjoyed any mishap which would put a crimp in his stiff spine or have belittled and embarrassed him.

A short time ago the writer hereof was conducted through another shop by one of its executive officers. No sly warnings, no simulated activity preceded him, no surly glances, supercilious remarks nor mocking grimaces followed in his wake. He gave a cheery word wherever he stopped. He knew most of his men by name, asked after their babies, how the new house was coming along, told another to call at his home and see what "mother" had for his sick wife, knew how long they had been there, and their fathers before them, knew the ambitions and the capabilities of each. He could pick the best man for a given job and the man would sit up nights planning how he could make good to "Uncle Charlie."

Do you think one of them would have slapped him on the back, or asked him to have a drink? Whose dignity was the more secure, that of the prim and pompous ex-naval executive or that of Uncle Charlie, whose men bring their babies around on Sunday to see him and are proud and happy in his notice of them, as he is simple and hearty and natural in his interest in and enjoyment of them?

In which shop is there likely to be the better *esprit de corps?* Which shop is likely to turn out the best work at the least cost? In which shop is working most likely to be like living, with a pleasant, wholesome, craftsman-like interest in what is being created by and about one?

It is unfortunately true that there are individuals who, as our correspondent says, "are inclined to take advantage always of each and every point." Give them an inch and they will take an ell. Speak to them pleasantly today, they will be calling you "old chap" tomorrow, and boasting of their familiarity with you. Of such are the Goops, and they should be sternly repressed to their own level, which is the same whether they work in the fireroom, at a lathe, or over a mahogany desk.

And in closing may we remark that if you will watch the man who is so grandiose and overbearing toward his inferiors, you will probably see an exhibition of the most abject toadyism whenever he is in the presence of anybody who is higher than he in authority or social standing.

When Condemnation Is Justified

On page 463 a correspondent expresses the belief that engineers as a class are egotistical and delight in speaking unfavorably of their predecessors.

In many instances this is true, but more often the unfavorable comment is justified. If the truth were known, the shortcomings of the predecessor are due, not to his incompetency but to the reluctance of the management to supply the necessary material to keep a plant in proper running condition. Almost any engineer, however good he may be, will become discouraged when his requests for essentials to good service have been repeatedly denied by the shortsighted manager. Then he loses interest in the plant and things get worse instead of better.

The engineer who must tolerate such conditions long, eventually loses heart in all things. Such a situation breeds the attitude that precedes the loss of self-respect and of the approbation of his fellow men and induces the pessimism that precedes degeneration.

So when a man in this frame of mind assumes charge of a run-down plant when he expected to find it in good condition, and meets with opposition in his efforts to put it on a paying basis it is likely he will condemn his predecessor, who undoubtedly worked hard to improve it also.

In engineering, as in all channels of human activity, the drone persists. He deserves the condemnation he receives. A man taking charge of a plant left in bad condition by a shirk is justified in protesting.

In Restraint of Education

Eighty thousand children were denied their right to attend school all day at the opening of the fall sessions of the public schools of New York City. This is disgraceful, if not criminal. The school laws compel the daily attendance of children up to the age of sixteen, and prohibit their doing manual labor. What a shame, then, to deprive them of half the time they need to get beyond the pale of ignorance and illiteracy.

Thirty-four million dollars was apportioned the Board of Education of the second largest city in the world for the education of its children for 1912, and the sectarian and private schools lifted the burden of educating more than two hundred thousand children, and yet eighty thousand are unable to attend school full time. The Board of Education should use some of the money it now pays for what are aptly called "frills"—music, dancing, "art," "specialists," etc.—in renting temporary quarters and increasing its staff of teachers.

We hope this matter will be taken up by the Institute of Operating Engineers and the local operating engineering organizations. They realize what the young engineer of the future is losing; they know how the young engineer has to struggle today, because in his early youth he had no chance to properly learn the necessary three R's. These organizations should voice a loud protest against an injustice that is working a hardship the gravity of which is almost impossible to estimate.

Given the opportunity of studying at the age when his mind is most receptive and his help as a bread-winner can best be spared by his parents, the boy will not suffer humiliation when he goes to work. He will not have to burn the midnight oil in his endeavor to acquire the education denied him at a time when he might far better be gaining strength in sleep and healthy recreation.

"To earn more, learn more" will prove a feeble slogan indeed if the local organizations in New York City do not demand the education for the city's children that is their right. And if this condition obtains in other cities, even in a lesser degree, its seriousness should be given cognizance and remedial measures taken to correct it.

In American schools there are nearly seven hundred foreign students, Chinese, Koreans, Turks, Chileans, Russians, Finns, Ceylonese and others. The high cost of living here is no deterrent to the foreigner's ambition to acquire knowledge that will make him a useful citizen in his own country and a credit to us.

We are glad to be enlightened by one of our esteemed contemporaries that "a turbine is composed of two essential parts, first the fixed part or cylinder, and second, the moving part or rotor."

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

The Engineer and His Predecessor

As a boiler inspector I meet so many engineers ready to dissect the record of their predecessors that I am inclined at times to believe that long association with cranks and eccentrics, not to mention blowing apparatus, has made engineers as a class very egotistical.

Even in POWER, sometimes the correspondent begins his letter by relating the deplorable conditions he found when he took charge—the pumps were out of order, engines out of line, boilers full of mud and scale, etc. Then he tells what wonderful changes he brought about by a few simple expedients. Human nature is much the same the world over, and most of us like to be complimented. Some go so far as to pat themselves on the back when they accomplish something notable.

The average engineer has a hard enough time without his successor throwing "bricks" at him. I have seen thoroughly competent men so held down by their superiors that the plant could not be kept in even fair condition. When repeatedly refused supplies and repairs, the best men are likely to become indifferent. Sometimes a superintendent or manager is convinced of the justness of the engineer's requests, but is too hard-headed to relent and lets things go along until the engineer quits in disgust. Then the new engineer gets everything needed to put the plant in order.

It is true there are engineers who take no interest beyond "putting in" the day, but they are few, and when an engineer's letter tells of exceeding his predecessor's record in fuel and other expenses, I wonder if there are not other reasons than incompetency.

A. C. TERLENE.

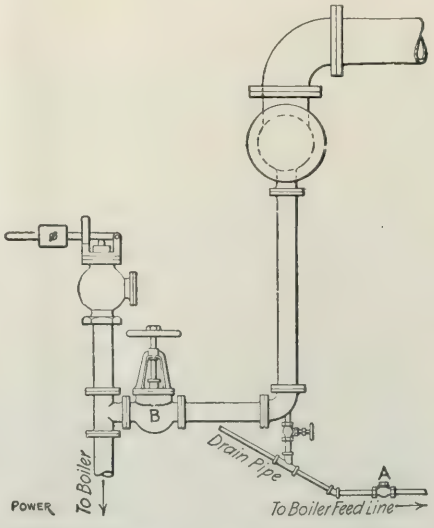
Chattanooga, Tenn.

What Broke the Stop Valve?

This plant has three horizontal return-tubular boilers, 150, 100 and 75 hp. respectively. No. 1 boiler had been cleaned and inspected and was steaming again with No. 3, while No. 2 was out of service to be cleaned and inspected.

When I first went into that boiler I noticed that the 5-in. stop valve was leaking enough to be annoying when cleaning. I notified the chief engineer, who tightened the valve down with a big monkey-wrench, and the leak stopped. At the time, the boiler had been off for 24 hours and had cooled. The boiler was then cleaned and found in good condition by the boiler inspector and was afterward made ready to be connected with the others.

The chief always connected the boilers himself. Before going on top of the boilers to connect No. 2 he closed the steam valve on No. 1. The drain-pipe connection was as shown. The check valves, one of which is shown at *A*, were leaking more or less and as the stop valve *B* had been screwed down tight when the boiler was cool, the chief had to use considerable force with a big wrench to open it, which occurred suddenly, opening the valve wider than he expected.



SHOWING BROKEN STOP-VALVE ON STEAM MAIN OF BOILER

A terrible racket indicated that something had gone wrong. The body of the valve burst, hurling the chief against the wall. With presence of mind, he crawled toward the front of the boiler, but fell over its edge upon the ashpit-door and broke a hip bone, besides being scalded badly. Eight months afterward he died.

A boiler inspector stated that the cast-iron body of the stop valve had a flaw in the casting. He ordered the drain pipe which was run into the feed line changed into the blowoff pipe. A tee was placed behind the blowoff valve and a 1½-in. pipe screwed into it, which was connected with the 1-in. valves on top of the boiler. The inspector said that these valves should be left open all the time. There was a valve put in the vertical 1½-in. line, which was kept closed. Each time the boilers had been blown down this valve was opened for a little while to let the condensate out of the line. When a boiler was connected the engineer opened this valve before he connected the boiler and closed it afterward. When No. 1 was shut off, the gage showed about 90 lb. and just that much on No. 2 and No. 3.

Why the chief shut off No. 1 I do not know. Do POWER readers think that closing the valve on No. 1 had anything to do with the explosion? What was the real cause of the trouble? Was it the wrongly connected drain pipe or tightening down on the valve when the boiler was cool? The valve had been on the boiler about 15 years. The boiler inspector called our attention to the thickness of the casting of the valve body, which was ⅛ in. thinner where it probably started to give way than it was in other places.

WILLIAM L. KEIL.

Philadelphia, Penn.

Hydraulic Elevator Indicator Diagrams

Indicator diagrams from hydraulic elevators are taken or published so seldom that those shown below should be of interest.

The diagrams were taken from a 22x 90-in. elevator cylinder operating under a tank pressure of 100 lb. The load was

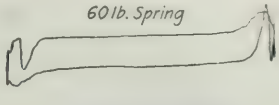


FIG. 1 Partial Stroke

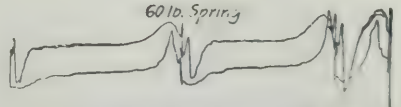


FIG. 2 Full Stroke

DIAGRAM FROM CYLINDER OF HYDRAULIC ELEVATOR

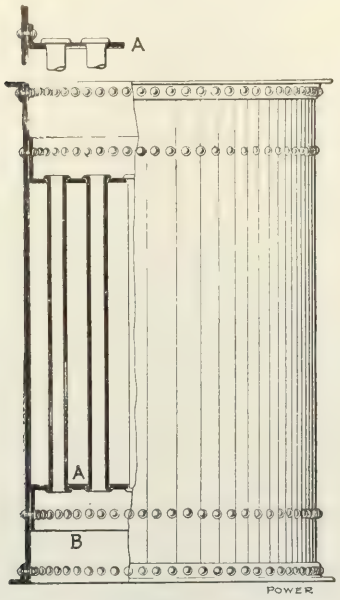
the same on the up as on the down motion; they are friction diagrams only. Fig. 1 is a record of only a part of the stroke. Fig 2 represents a full stroke and shows that a stop was made at a little less than half stroke. It also shows that the pump did not maintain full pressure throughout the stroke.

HARRY D. EVERETT.

Fort Apache, Ariz.

Saving Partly Injured Condenser Tubes

The illustration shows how a set of condenser tubes were saved. These tubes are of bronze and very expensive. The beading had given out from constant expansion and contraction and to save the condenser the tubes were all removed from the old heads, the ends measured and cut off square to an exact length. Then two new condenser tube heads were



CONDENSER REPAIRED BY SHORTENING TUBES

made with extended flanges, as shown at *A*. The distance from *A* to *B* was made great enough so that the tubes could be cut off and the heads riveted back in place, using the old rivet holes in the outer shell, thus saving the outer shell as well.

This condenser is used for cooling or condensing in an ice plant. Several of them were saved in this manner and do the work as well as new ones

C. F. McGAHEY.

Baltimore, Md.

Corliss Versus Four Valve Engine

Being much confused over the following, I would appreciate answers from those interested.

How is it that the four-valve nonreleasing engine is represented to show as good economy as the Corliss releasing engine—at a speed of 225 r.p.m.? Does the clearance not have to be increased? How is it explained that clearance is regarded as almost a total waste? Have not nearly all the improvements in Corliss engines been along the line of reducing the clearance?

The diagram factor of a good Corliss engine is about 95 per cent. and of a four-valve nonreleasing engine only about 87 per cent. How does this affect the economy of an engine? Would you not have to have a larger cylinder in a four-valve engine to do the same work as a Corliss? The four-valve engine makers claim that the steam is superheated by this wire drawing, but with the design and operation I fail to see how it is possible to guarantee the performance and economy of a four-valve engine to equal a Corliss.

Is it possible to make a showing of low steam consumption per indicated horsepower with the four-valve engine and still not do it? Does not the valve movement depend on the time element in the four-valve engine and in the Corliss on the load developed; does not this affect the economy? Is not the wear on a four-valve nonreleasing engine very rapid?

How does the angularity of the connecting-rod affect the diagram of an engine or bear on the mean effective pressure of the crank end? On a four-valve engine it will be noticed that the connecting-rod is very short as compared with a Corliss.

I realize that I am asking for considerable information, but there are many others who would appreciate a discussion on these questions.

B. E. DONOVAN.

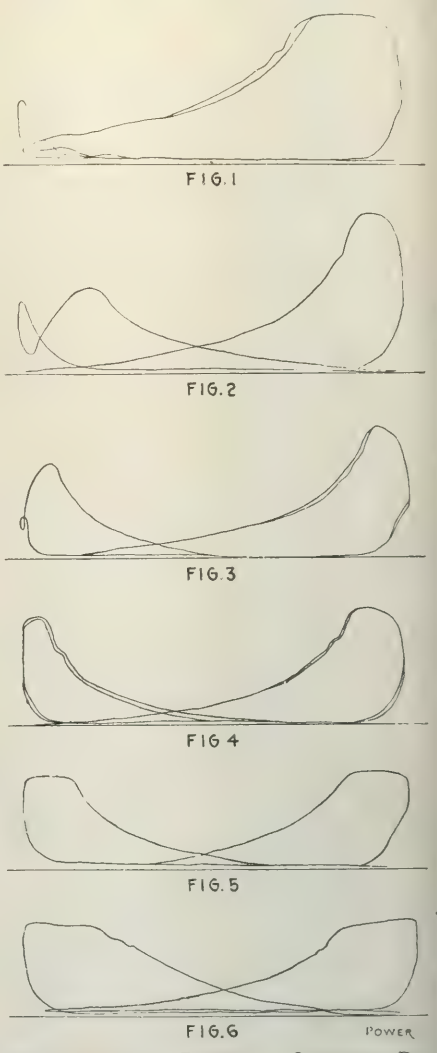
Omaha, Neb.

Adjusting a Brown Engine Valve Gear

A mixed-up valve-gear had caused an engine to be condemned as "no good." When diagram Fig. 6 was taken it was as good as new. Some unknown means, apparently continual tampering with the valves, caused the steam pressure to be raised from 80 lb. to 125 lb. to get the engine to do even a small portion of the shop load. The valve-gear had numerous scratch-marks, one of which I could use, so I put on the indicator and obtained diagram, Fig. 1.

I then set about to get the proper steam lap and length of valve stems. The old method of getting the lengths of the valve stems is to take off the valve chest and valve and measure from the top of the guide-block to the bottom edge of the bottom port. The valve stem is the length from the bottom mark on the stirrup block to the bottom of the valve. These should correspond when the valve stem is the right length. The mark made by the builders can be seen by taking off the guide-block, and is found scratched on the engine bed. If the valve is down on the bottom of the steam chest the valve-stem end will come even with it, or it can be made to do so by screwing the stem in or out of the valve. I used the indicator pipe to get the puff of steam, noting the marks at the same time. With a bar I raised the valve until I got steam from the indicator and noted the distance the valve stem had traveled. The engine was known to be practically steam-tight. Changing the valve stem until the travel corresponded to the marks on the stirrup block, having now got the laps, I proceeded to set the valves.

Fig. 1 showed the head-end events to be late. I advanced the head-end eccentric to get earlier admission, and obtained Fig. 2. This caused the opposite end to shorten its cutoff automatically.



BROWN ENGINE DIAGRAMS SHOWING RESULT OF VALVE ADJUSTMENT

I next advanced the crank-end eccentric and got an earlier opening, as shown in Fig. 3. Then I threw both exhaust eccentrics back to delay their closing, giving less compression, and obtained Fig. 4.

I next proceeded to equalize the cutoff, making the crank end $\frac{7}{16}$ in. and the head end to correspond by loosening the setscrews and pushing the taper pin in and out until I got a very close adjustment. There are only setscrews on the crank end, no taper pin being used. After the changes, a load was put on the engine and diagram, Fig. 6, was taken with 80 lb. steam. After blocking the governor

a trifle lower than its highest position, I so adjusted the trip shaft that by turning the top of the handwheel away from the cylinder the valves would not pick up enough to uncover the steam lap in case the engine tried to race.

This engine had been running with a large amount of lap on the steam valves, causing most of the eccentric's throw to be taken up before the valves were open and decreasing the range of cutoff. By decreasing the lap, the load was carried with greater ease and economy.

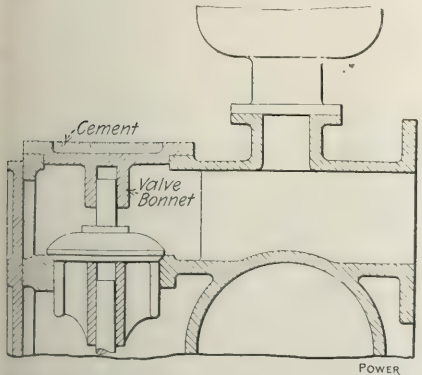
I would like the reader's comment on the diagrams.

A. C. WALDRON.

Revere, Mass.

Keeping Valve Chamber Covers Clean

One of the little jobs done to lessen the labor of cleaning in the plant is shown below. Each of the valve-chamber bonnets of the feed and stoker pumps had



CEMENT-FILLED VALVE-CHAMBER COVER

a depression about 2 in. deep and 3 in. in diameter, which afforded a resting place for oil, water and dirt. There were 96 of these unsightly holes and a visitor's suggestion to fill them with portland cement was adopted. A little manganese dioxide was added to color it black, and when it was painted it looked like the solid metal.

T. W. ZERMEIN.

Brooklyn, N. Y.

A Troublesome Injector

The makers of our injector specify that the steam, water delivery and suction pipes must be of 1½-in. diameter. Its capacity is 55 litres (12.1 gal.) of water per minute, and it is supposed to start at any pressure between 45 and 120 lb. It will lift the water all right and allow it to escape at the overflow and will inject water into any of the boilers when there is no pressure in them, but the moment a boiler is under anything above 30 lb. pressure the injector will not work. It has been sent back to the makers and they report it in working order. The lift from the supply tank is 6 ft., as shown, and the temperature of the water is 59 deg. F.

The injector is for use only on the 90- and 120-lb. boilers, as per specification, and has never been tried on the 195-lb. boiler. The injector takes steam from the main header, which is common to all three boilers, but only takes steam from one boiler at a time because no two boilers are in service simultaneously. I tried giving the injector steam from the 120-lb. boiler and injecting water into the 90-lb. boiler while the pressure in it was being increased, but it would not work. The same thing happened when I tried it with steam from the 195-lb. boiler and injecting water into the 120-lb. boiler, and again when I tried injecting water into either the 90- or 120-lb. boiler by using steam from the same boiler into which the water was being forced. With our duplex pump or engine pumps no trouble is experienced, so it cannot be due to faulty piping unless it is that there are too many ells and tees in the feed line. The bends in the line at the well have a radius of 10 times the pipe diameter. The distance from the injector to the check valve on the 120-lb. boiler is 16 ft., and to the 90-lb. boiler check valve, 42 feet.

Can readers tell me what causes the trouble?

E. S. WYLDE.

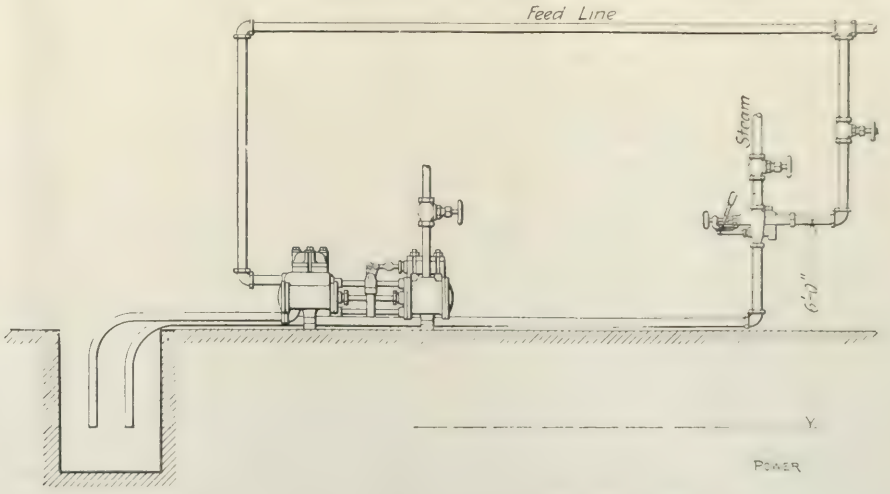
Pinerolo, Italy.

The Tilt Trap as a Boiler Feeder

Some consider the tilt trap the most efficient and economical boiler-feeding device. It may be when condensation must be removed from long steam lines, heating systems, etc., and returned directly to the boiler. It is claimed that the tilt trap is equally efficient whether the initial temperature of the feed water is 40 or 250 deg. and that its operation will satisfy all conditions and mean a considerable saving of coal, even though the existing boiler-feeding apparatus may be of the highest order with respect to the saving of heat units in the available exhaust steam.

I do not believe that the operation of tilt traps for boiler feeding means all "velvet," and would be interested in the opinions of POWER readers who have given this matter considerable study.

Operating engineers insist that the injector is a "steam hog" because it "pulls the steam down" and if it does there must be some reason for it. However, the injector has a very high thermal efficiency and since the heat in the steam used is returned to the boiler in the feed water it ought to be rather efficient and economical as a boiler feeder. But supposing that the injector is a real "steam eater," would not the tilt trap come under the same heading? Like the injector the steam trap depends upon live steam for its operation, and both may be classed as live-steam feed-water heaters. This brings up the question whether it is desirable to use live steam for heating feed water when exhaust steam is available for that purpose. If a tilt trap is installed to replace a steam or power pump, would greater economy result?



DISCHARGE AND SUCTION PIPES OF PUMP AND INJECTOR

The following questions have a direct bearing upon the economic operation of tilt traps for boiler feed:

Will the evaporation be the same whether the feed water is admitted regularly or intermittently, assuming that all other conditions remain exactly the same?

Would the intermittent admission of feed water retard the circulation, especially in a tubular boiler having the feed-water intake through the blowoff connection? And if the circulation is retarded, would it not affect the heat transmission which in turn would affect the evaporation and the coal pile?

The operation of tilt traps is such that the feed water is discharged at intervals.

A. A. WICKLAND.

Chicago, Ill.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Purchased Current for Summer Months

The arguments of W. A. Blonck, of Chicago, in favor of shutting down office-building power plants during the summer months and purchasing electrical energy from a central station and the editorial criticism which appeared in the Aug. 20 issue, interested me.

As I disagree with Mr. Blonck on some points and POWER on others and agree with both in still others, I wish to give what I consider an impartial statement of some of the more important factors entering into this problem.

Every office building presents a separate problem, and whether a power plant is more economical than a heating plant and central-station electric service depends entirely on the many and varied factors that enter into the particular building and can only be determined by an unbiased study of the cost of operation with and without an isolated plant. One thing certain is that few office buildings could afford to maintain their own plants if exhaust steam was not available for heating purposes during the winter. Also quite certain is that during the summer months making electricity costs as much if not more than purchasing it from a central station. As to which is more economical, isolated plant the entire year or isolated plant during the heating season and central-station service during the summer, is again determined only by local conditions. For each building the question must be decided on its own merits.

Mr. Blonck's statement that an isolated plant takes four tons of coal to develop the same amount of electrical energy that can be produced with one ton in a central station is absurd. The efficiency of the boilers would be about the same in either case. The efficiency of the engines might be 50 to 75 per cent. in favor of the central station, which would be offset to a small degree by the losses in the distributing system, but at the worst the isolated plant would hardly burn more than double the coal for the same amount of electrical energy. Where the central station gains is in its ability to burn cheaper grades of coal, the special price obtained by buying in large quantities and more economical systems for handling the coal and ashes; also, in many plants the lower cost of labor due to the use of stokers. Mr. Blonck's statement in regard to the elimination of radiant heat from the boilers, engines, piping, etc., is well taken. Almost all office-building power plants are regular Turkish baths during the hot summer months and an ounce of prevention, by having no heat to radiate, might be worth the pound of cure suggested in the foreword in the Aug. 6 issue.

POWER states that during the summer months electric elevators and motor drive for all pumps and auxiliaries would be required, and no exhaust steam from these pumps would be available during the winter for heating the feed water and that the steam would have to be taken from the engines. Most office buildings have electric elevators, and if they have hydraulic elevators, the elevator pumps can be changed to electric. The only other pumps that would have to be replaced by electric are the house and fire pumps (very often the same pump is used for both purposes) so that this is not such a serious objection as it might seem at first. As for the loss of exhaust steam, even if all electric pumps were required, the energy has to come from the boilers whether it goes direct to the pumps as steam or first to the generating unit as steam and then to the pumps as electricity. Also as the engines are more efficient than steam pumps it would be cheaper to operate the electric pumps unless the heating demands required more live steam from the boilers than the difference between that necessary to generate the electricity in the generators and to operate the pumps. In the latter case the steam pumps would be cheaper as the initial and maintenance costs are less.

The point that many buildings are fitted with absorption refrigerating systems which require exhaust steam is well taken, and such buildings would hardly find it profitable to use central-station service at any time.

The labor question is probably the most important item entering into the problem and would go a long way toward making the final decision. The statements by POWER on this point admirably cover the ground.

While central stations would probably prefer to have a yearly contract it should be to their advantage to increase the summer load by giving office buildings especially attractive rates for this kind of service. The summer load of any central station is much lower than the winter load, due to the exodus of people from the city, and the longer hours of daylight.

Now that all is said it still remains a problem for each building to solve independently, taking into consideration the points mentioned above.

No doubt in some cases a great saving in the yearly expense could be made by following the method suggested by Mr. Blonck, but in other cases just the reverse would be true. Still the question is so important that it would pay all office power-plant engineers to take this idea under consideration and see how it would affect their own plant.

EDMUND LEAVENWORTH.

Brooklyn, N. Y.

Back Water from Open Heater

Relative to the question by G. W. in the Aug. 13 issue, I would advise him that all open heaters have an overflow connection which takes off water at a lower elevation than the steam inlet; so that even if the float valve failed to work there would be no chance of water getting into the engines from this source.

THEODORE HAIGHT.

Brooklyn, N. Y.

Friendliness and Familiarity

One is not supposed to criticize nor to comment on the articles known as "editorials," because the editor is supposed to be the last word on any subject. Yet, when one sees where an editor has gone wrong, it is only a "duty well done" to have pointed out the error of his ways.

The article to which I take exception is found on page 94, July 16 issue, and entitled "Know Your Employees." Your article starts with the statement "that it was once believed, etc.," and that old-time belief, though wrong as regards the "dignity" part of it, was right as regards the discipline. When a manager, superintendent, or even chief engineer gets the habit of being familiar with, or allowing the men under him to become familiar, right then and there the wheels of good management start to clog, and in time a new crew, or a new head man will become necessary.

Of course the manager, or superintendent should go into the plant, or factory frequently, see how things are going and if all is not well, should call the head of

the visited department into his private office, and talk the matter over with him; and never jump on one of the crew, nor the head while in the presence of the men.

Man is a very peculiar sort of creature, and while we may, and do, all blow and bluster, yet we know way down in our hearts that we are inclined to take advantage always of each and every point. Should the superintendent come to the plant, and allow me to get to calling him by some given nickname, or to slap him on the back, and invite him to go drink with me, then he would either lose his respect for me, or he would be unfit for the position held by him, because were he to allow this, it would be but a matter of a very few days or weeks until I would lose my respect for his authority, and then I would be worse off than if he had been strict, and formal and shown no friendliness toward me.

The manager of the plant where I am employed as chief engineer is one of the best men I have ever known. One or two things in our relations, however, are peculiar. As a manager, he is as strict and unbending as an army officer and holds me to account for everything that takes place in the plant. He does not jump on my crew, but as he should, calls me into his office and "stands me on the carpet." I respect him for this and he knows it. One of the other departments had a head man, who, because the manager asked him a few questions about his private affairs in a kindly manner, took it for granted that he had a "stand in" with him, and so became negligent in his work (as all men will do, should the management allow themselves to become familiar) and in two weeks was calling him by his given name. The third week he was so friendly that he attempted to slap the manager on the back. As a result he was shortly looking for another job, while the manager felt that he could not afford to get close to his men.

In the army, discipline must be maintained, and here or in a civil office when a superior gets so that his men know him too well, it is bad both for the employee and the employer.

JOHN PIERCE.

Alva, Okla.

Air Compressor Indicator Diagrams

The indicator diagrams offered for comment by C. E. Anderson in the June 18 issue, interested me. Instead of the air-cylinder diagram, Fig. 6 (here reproduced as Fig. 1), I would have preferred to have had the real, full-sized card. If it was really taken with a 40-lb. spring, and the air delivery pressure 75 lb., then it was about as long as any standard indicator will make.

I am continually more certain that all indicator diagrams are not absolutely truthful in the story they are supposed to tell. The number of diagrams which do not carry some suggestion of incorrect indicator action is not as large as it might be. The drum movement should be actually coincident with that of the piston, and this it too frequently is not. A connection between the drum and the crosshead should be as absolutely non-elastic as that of the connecting-rod with the crank, and the necessary flexibility of the former connection should not be permitted to excuse it. The indicator should be as rigidly mounted as a New York skyscraper; there should be no elasticity and no lost motion anywhere in the connections. The drum spring should have sufficient tension to command the drum inertia, and friction should be nowhere in evidence.

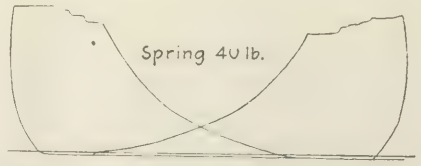


FIG. 1

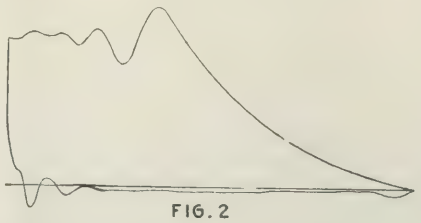


FIG. 2

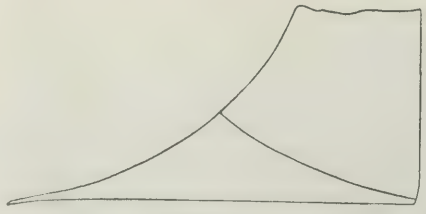


FIG. 3 POWER

DIAGRAMS FROM AIR COMPRESSOR

This diagram, Fig. 1, was taken from a slow-running machine, 50 r.p.m., yet it seems to be lacking in the precision of movement which is so essential. In air diagrams the reëxpansion line is very important, and as it is made so quickly, or in such a small part of the total drum movement, precision of movement is here most essential; where it is lacking the inaccuracy is noticed at once.

I notice that the drum reached the limit of its travel, ostensibly representing the end of the compression stroke, when the pencil was at *a*. But the card reveals that reëxpansion, representing the beginning of the return stroke, had actually begun before that, and the pressure in the end of the cylinder had fallen more than one-third before the indicator began to record the return stroke at all; of course, the indication was incorrect. That this is a frequent fault in air diagrams does not excuse it, because cards can be and are taken, on faster-running machines where the beginning of the return drum movement and the beginning of the reëxpansion curve occur coincidently at the upper corner.

The indicator inaccuracy appearing in the reëxpansion line of Fig. 1, was due to the arrival of the indicator drum at the end of its travel, and then the beginning of its return too late for the compressor piston. The latter had evidently begun its return, and the pressure in the clearance space had dropped to a considerable degree before the drum began to move.

Fig. 2 is a card, typical of a dozen which recently came to me, with a slightly different peculiarity. Here the indicator drum seems to have reached the end of the compression stroke on time, and the pencil shows the correct position for the start on the return stroke, only the drum does not start promptly. The compressor piston starts and the pressure in the clearance space drops nearly to atmosphere, and then the drum starts suddenly as shown by the sharp crook. The reexpansion line should have been a smooth curve clear down to the suction line, beyond which point the indicator action seems to have been normal.

Nothing is told about the make and style of the compressor or the type of valves used, especially the inlet valves which seem to be positively operated, as shown by the compression line of Fig. 1, where the pressure remains below atmosphere for the first quarter of the stroke, and then the intake valve closes quickly and compression begins. This control of the inlet valve and the closing of it at varying points of the compression stroke, is the arrangement employed in certain compressors for varying the volume of air compressed according to the varying demand for the air. If the inlet valve is closed by positive mechanical means it also might be opened in the same way, and it might be adjusted to open too soon, which would give the peculiarity spoken of in the compression line of Fig. 1, except that at the moment when the compressor center was passed the drop should have been more decidedly vertical than it was.

A vertical straight line for some distance should have been expected in the case. Notice the vertical drop of the reëxpansion line in Fig. 3. This was not caused by a prematurely opened inlet valve but by a very leaky piston, but the vertical drop should have been practically as abrupt with the open inlet valve, if that was the actual condition.

I hope that Mr. Anderson will tell us all about it.

FRANK RICHARDS.

New York City.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Increasing Pressure of Forced Draft

How can the delivery pressure of forced draft from a fan blower be increased? Cannot the pressure and volume be raised by increasing the speed of the fan shaft?

J. M.

Pressure can be increased by partly closing the outlet but the volume of free air delivered will be decreased. Both pressure and volume will be increased with increase of speed, though the pressures will not be in proportion to the speed of the fan, but to the square of its speed. To increase the pressure 50 per cent. would require the fan to be speeded up to 2¼ times as fast; and to double the pressure, would require the fan to be driven four times as fast.

Do Not Wet Firebrick

Is any serious damage done to firebrick of a furnace by wetting down before pulling the fire?

F. H.

No serious damage may be done if water is confined to the ashes and clinker, but it is bad practice to wet down in such a manner that any water reaches hot firebrick, as it results in rapid failure of the furnace lining.

Miner's Inch

What quantity of water or rate of flow is a miner's inch?

G. H. C.

A miner's inch of water may be roughly defined as the quantity which will be discharged by a vertical orifice 1 in. square. The term originally signified a unit of measurement in designation of the size of apertures employed for gaging water supplied for mining purposes, regardless of kind or proportion of orifices. which were usually employed under heads ranging from 4 to 7 in. above the centers of the apertures. The average head employed is about 6½ in. and the discharge averages about 1.5 cu.ft. of water per min. per sq.in. of aperture. As the quantity of water passing through an aperture per square inch of area is modified by the proportions of the aperture, the velocity and head of water presented, the form of channel of approach, the shape of the edges of the aperture and freedom of discharge, the quantity represented by one miner's inch is very indefinite unless all conditions are stated. In view of the uncertainty, California and Montana have established by law that 40 miner's inches are equivalent to 1 cu.ft. per sec., making one miner's inch equivalent in those states to 1.5 cu.ft. per min.; while in Colorado 38.4 miner's inches is made equivalent to 1 cu.ft. per sec., or one miner's inch is equivalent to 1.56 cu.ft. per min. In many of the states, by common agreement, 50 miner's inches are assumed as equivalent to a discharge of 1 cu.ft. per sec., making the miner's inch equal to 1.2 cu.ft. per min.

"Cataract"

What is a "cataract" regulator?

F. H. K.

In single-acting engines the exhaust valve and the steam valve are not opened directly by the plug rod, but by a piece of mechanism called the "cataract," of the nature of a pump brake. Coming down from the name applied to a form of regulator used on early Cornish engines, in which motion of a rocking bucket alternately filling and tilting was employed for controlling the period of admission of injection water, the term "cataract" continued to be applied to hydraulic mechanism used for regulating the motion of engine valves and more recently to other forms of governors that operate by displacement of oil or other liquids.

Reactance and Impedence

What is the difference between reactance and impedence in an electric circuit?

T. P.

In direct-current work the difference in potential is equal to the current multiplied by the ohmic resistance, i.e.,

$$E = I \times R \quad (Ohm's\ law)$$

In alternating-current work, however, there is more than the straight ohmic resistance; there is self-induced electromotive force which opposes the impressed electromotive force. This is called "reactance." It is not in phase with the current; hence must be added vectorially to obtain the total resistance which is called "impedence," hence,

$$Impedence = \sqrt{(resistance)^2 + (reactance)^2}$$

Boiler Brackets

Why are present-day return-tubular boilers usually provided with two supporting wall lugs or brackets on each side, in place of three, as formerly?

G. H. P.

When three pairs of brackets are employed, expansion of the furnace walls or height of the bedding of the wall bearing plates are likely to cause the boiler to be mainly supported by the middle pair of lugs and thus throw excessive weight on the side walls of the furnace, making them more difficult to repair than when the boiler is supported on a pair of lugs near each end. When two pairs of lugs are used, in addition to affording better distribution of weight on the walls there is also the advantage of imposing less transverse strain on the boiler, from the weight of the boiler and its contents, as equal transverse strains would occur only if the lugs were placed at the ends.

Babbitt Metal

State a formula for making hard babbitt metal for engine bearings. What is 12, 1 and 1 babbitt metal and how is it made?

J. H. A.

A good, hard babbitt metal is made in the proportions of 88.9 per cent. tin, 3.7 per cent. copper and 7.4 per cent. antimony. Melt 12 parts of copper and add 36 parts of tin; reduce the temperature as soon as the copper is melted. Add 24 parts of antimony and 36 parts of tin, having the surface of the molten metal covered with pulverized charcoal to prevent oxidation of the tin and antimony. The alloy thus formed is to be remelted as required and used in proportions of 1 part of the alloy with addition of 2 parts of tin. "Twelve, one and one" babbitt is a tough white metal used for lining bearings, crankpin bushes, etc., and is composed of 12 parts of tin, 1 of copper and 1 of antimony. It is usually made by melting 6 parts of tin and 1 part of copper and separately melting 6 parts of tin and 1 part of antimony and mixing the alloys, the object of forming separate alloys being to expose only one-half of the tin to the high temperature of fusion required for first introduction of the copper and thereby subject the more highly oxidizable metals to as low temperatures as possible.

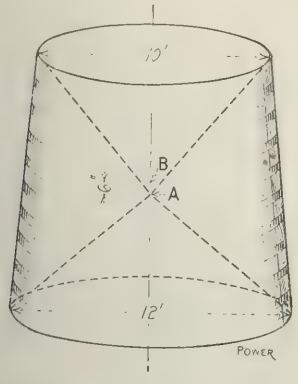
Study Questions

This Week's Questions
Last Week's Answers

(96) What is the available horsepower from an engine of 90 per cent. mechanical efficiency when the indicated horsepower is 85.6?

(97) It is found that six ells and seven tees, all for the same size pipe, have a combined weight of 33 lb., while seven ells and six tees weigh 32 lb. What is the weight of each ell and each tee?

(98) In a roof tank of the form shown in the diagram, a point *A* is found in the vertical axis, 6 in. below the mid-



height *B*, which is equally distant from all parts of the two circles formed by the intersection of the sides with the top and bottom (as along any of the dotted lines shown). The diameter of the top is 10 ft. and of the bottom 12 ft. How many gallons will the tank hold?

(99) The power factor of a 500-volt, three-phase induction motor is 0.85 when taking 25 hp. from the line. What will be the current in each phase?

(100) A grade 1000 ft. long has a rise of 50 ft. What horsepower is required to draw a load of 200,000 lb. up the grade in 1 min., assuming the friction to be 2 per cent.?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

---

(91) The absolute pressure corresponding to 175 lb. gage would be

$$175 + 14.7 = 189.7 \text{ } lb.$$

Perfect vacuum would be 0 lb. or 30 in. of mercury. The vacuum is not perfect by

$$30 - 26 = 4 \text{ } in.$$

and the absolute pressure corresponding to 4 in. is

$$0.49 \times 4 = 1.96 \text{ } lb.$$

Hence the total difference in pressure is

$$189.7 - 1.96 = 187.74 \text{ } lb.$$

(92) *A* riding twice as fast as *B* walks, reaches the end of the first mile when *B* has only gone half a mile and is still half a mile ahead of *B* when *B* reaches the bicycle, so that both reach the end of the second mile together. As the same is true of any even number of miles, *A* and *B* reach their destination at the same time. Each rides half the distance, 6 miles and walks 6. The riding speed being 8 miles an hour and the walking 4—the time taken is

$$\frac{6}{8} + \frac{6}{4} = 2\tfrac{1}{4} \text{ } hr.$$

(93) If the pipe is 9 in. thick its internal diameter is twice 9 in. or 1½ ft. less than the external, or

$$3\tfrac{1}{2} - 1\tfrac{1}{2} = 2 \text{ } ft.$$

If the pipe were solid its volume would be its length times the cross-sectional area. The area of a circle being

$$\frac{\pi D^2}{4} = 0.7854\, D^2$$

where D = diameter, the volume is

$$0.7854 \times 3.5 \times 3.5 \times 16 = 153.938 \text{ } cu.ft.$$

Similarly the volume of the bore is

$$0.7854 \times 2 \times 2 \times 16 = 50.266 \text{ } cu.ft.$$

Then the actual volume is

$$153.938 - 50.266 = 103.672 \text{ } cu.ft.$$

and the weight

$$35 \times 103.672 = 3628.52 \text{ } lb.$$

(94) In an 8-pole generator there are four cycles per revolution. To obtain 60 cycles per second the generator must make $\frac{60}{4} = 15$ revolutions per second, or

$$15 \times 60 = 900 \text{ } r.p.m.$$

(95) By trigonometry the sine of the acute angle made with the base is equal to the altitude divided by the hypothenuse. Since the hypothenuse is equal to the square root of the sum of the squares of the other two sides it equals

$$\sqrt{28^2 + 35^2} = 44.82$$

Then the sine equals

$$\frac{28}{44.82} = 0.62472$$

From tables of natural trigonometrical functions the corresponding angle is found to be

$$38 \text{ } deg., 39 \text{ } min., 42 \text{ } sec.$$

As the sum of the acute angles of a right-angled triangle is 90 deg., the other acute angle is

$$90 \text{ } deg. - 38 \text{ } deg., 39 \text{ } min., 42 \text{ } sec. = 51 \text{ } deg., 20 \text{ } min., 8 \text{ } sec.$$

Erratum: An error in the location of the decimal point appears in the answer to Study Question No. 79 in the Sept. 3 issue. The volume of a pound of water is 0.016 cu.ft. instead of as given and the answer should have been 8.375 lb.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

A contemporary calls New York City's 55-story building a "sky-scratcher." This probably accounts for so many people itching to climb to the top of it.

It costs 56c. to stop a 1500-ton train running 15 miles an hour, says a railroad journal. Costs more than that before an exploding boiler is stopped.

DORAN—Who is ut put th' 'con' in economy, Terry

HORAN—An' who is ut t'rew th' fish in efficiency, Jawn?

Time! No hitting in the clinches!

Your scribe could find nothing of interest last week at Knuckel & Funnibone's fertilizer plant except that Guy Lines had two feet taken off and Derrick had a stiff leg.

A Western paper records a somewhat painful and embarrassing accident to Will Brown, a South End boy, by saying that "he was burned in the South End by a live wire." Undoubtedly, friend William will stand around for some time. The rest of the scholars may resume their seats.

F. Tennyson Webster has worked the poetic muse and made her give up the following:

TEN LITTLE ENGINEERS

Ten little engineers running a turbine;
One lost his vacuum; then there were but nine.
Nine little engineers figuring water rate;
One fell in the test tank; then there were but eight.
Eight little engineers worked till half-past 'leven.
One dropped in the flywheel; then there were but seven.
Seven little engineers in an awful fix!
One blocked the safety valve; then there were but six.
Six little engineers in a grimy hive;
One asked for 60 bucks; then there were but five.
Five little engineers strutting 'round the floor;
One broke the governor; then there were but four.
Four little engineers taking down a tee—
Took off all the nuts at once—then there were but three.
Three little engineers cleaning out a flue;
Along came a back draft; then there were but two.
Two little engineers having lots of fun,
One leaned on a live switch; then there was but one.
One little engineer stayed there all alone,
Till the owner bought his power—now there are none!

Keeping Track of Things

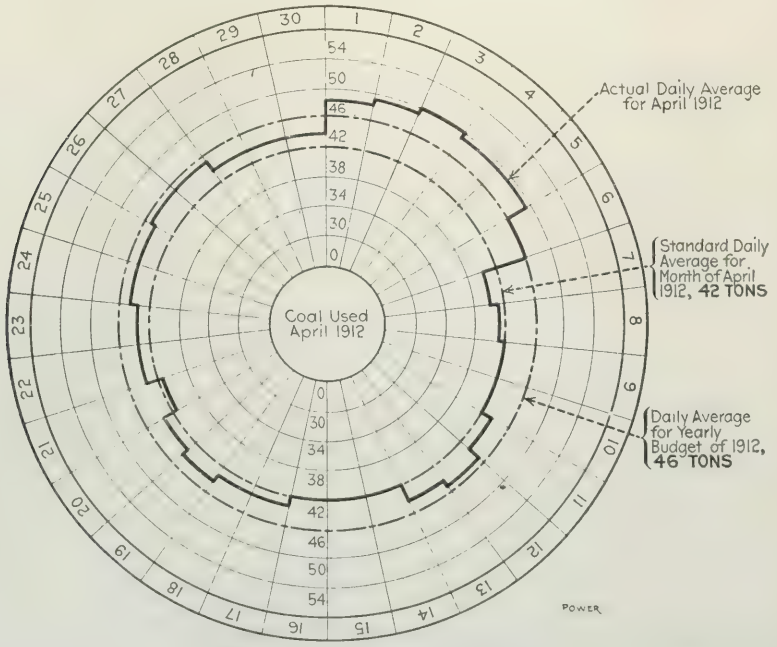
J. C. Jurgensen, president of the Institute of Operating Engineers, has made his success by keeping and improving records. In the last issue of the *Bulletin* of the institute is described how an engineer with no apparatus can chart his course, tell whether he is drifting into extravagance and be put on guard against extraordinary losses indicated by abnormal consumption by means of a circular chart system which Mr. Jurgensen uses.

In order to explain the method of use and the application of this chart, there is shown on the accompanying chart a coal-consumption curve for the month of April with its table of figures. The table has four columns, one for the date, one for the tons used each day, one for the total tons used to date and one for the daily average to date. There are three significant curves on the chart, the daily average as figured from the budget, the standard daily average and the actual daily average to date for the month.

The budget is the amount of coal that is allowed for use in a certain period. By dividing the total tons thus given by the number of days in the period, "the daily average shown by budget" is found. If the employers do not specify a certain number of tons for a certain period, the chief engineer can fix his own budget. It is the duty of every operating man to see how much he can save on the budget and to do this there must be fixed a certain standard, adherence to which means such a saving. The operating engineer after studying the conditions should fix a standard for his men to work by and to try and better. This standard will vary from month to month and should be fixed after a careful study of the records of the previous months of the period. It is on the bettering of this standard that the payment of a bonus is often based. Now two standards have been fixed for everyday use, the standard as figured from the budget and the standard as figured from previous records in conjunction with the budget. The first is fixed once for each period, year or half year as the case may be, and the second is fixed once for each month.

As soon as the chart is drawn up, the standards are put in with red ink, and the amount of the standards written beside them, thus placing at hand all necessary data so that the chart tells at a glance just how the daily consumption stands as compared with previous records. The broken curve is for the accumulative daily average. It is plotted day by day from the figures in the table. Coal used for any day is entered opposite the date in the second column and added to the total in the third column, then the third-column figure is divided by the date of the month and the result is the accumulative daily average to date for the month. After entering the figures in the table and drawing the daily line on the chart one can see at a glance how the day's run was made. If the consumption was too high, if the daily average is creeping up or if there is a large variance from running conditions it becomes apparent and one can investigate and remedy it immediately. The curve may be kept in any convenient unit, for instance, if there is no scale upon which to weigh the coal or ash the record can be kept in the number of wheelbarrows of coal or in the number of cans of ashes.

Date	Tons of Coal Used	Tons Used to Date for Month	Daily Average for Month	Date	Tons of Coal Used	Tons Used to Date for Month	Daily Average for Month
1	48	48	48	16	50	673	42
2	50	98	49	17	55	728	43
3	52	150	50	18	48	776	43
4	45	195	49	19	52	828	44
5	49	244	49	20	35	863	43
6	30	274	46	21	15	878	41
7	10	284	40	22	60	938	43
8	45	325	41	23	58	996	43
9	48	373	42	24	55	1051	44
10	51	424	42	25	58	1109	44
11	52	476	43	26	60	1169	45
12	60	536	45	27	40	1209	45
13	35	571	44	28	20	1229	44
14	12	583	42	29	45	1274	44
15	40	623	42	30	50	1324	44


Actual Daily Average for April 1912
Standard Daily Average for Month of April 1912, 42 TONS
Daily Average for Yearly Budget of 1912, 46 TONS
Coal Used April 1912
POWER

CHARTING THE COAL CONSUMPTION

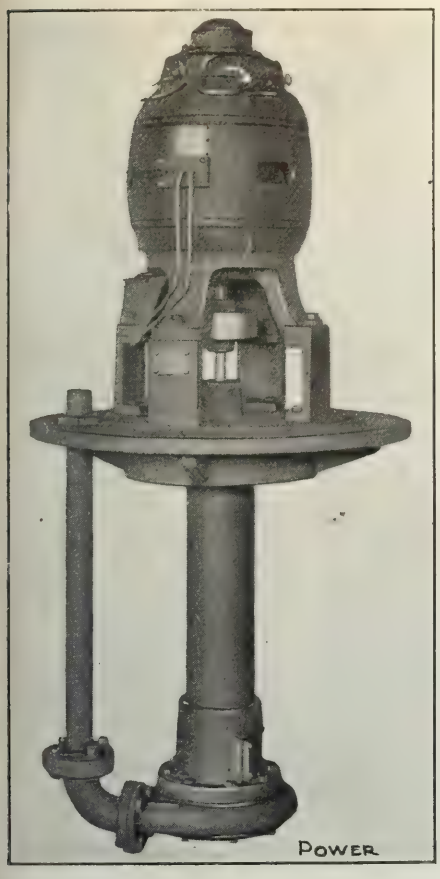
There is another chart which lends itself very nicely to power-plant work and that is the chart upon which are plotted the tons of coal used per day and the tons of ash removed that day. By direct comparison one can see from this chart how the firemen handle the furnaces each day and whether the coal is of a better grade or not, and by measuring the distance between the two curves can be found directly the number of pounds of combustible that were used. Where there are three watches in each day's run, the space for days on the chart may be divided into three spaces and the records of each watch plotted in these spaces and then the efficiency of each watch becomes apparent. This chart can be applied to cost of repairs, cost of salaries, material used, hours machines were in operation, ash removed, coal used, oil used, average loads, water used and in many other ways. Once one learns to keep and study his records, from the chart as described, he will find many ideas for its use springing up and will be surprised how, by a small amount of watching and studying, the costs will be kept lower and better results obtained without going to the expense of purchasing elaborate instruments and hiring men to tend them.

A Motor Driven Centrifugal Sump Pump

The outfit, shown herewith, consists of a vertical motor directly connected to a single side-suction pump, that is connected to a cast-iron pit cover by a rigid pipe column. This cover also acts as a support for the motor pedestals. The shaft is inclosed by the column which connects the pump to the cover and is therefore protected from the liquid in the pit. A removable plate on the flange which connects the pump and sup-

porting column permits inspection of the stuffing-box.

The electric motor is controlled by a starting device, actuated by a float in the sump. When the liquid rises to a certain level and the float is lifted enough to close the switch the motor is started. It continues to operate until the sump



MOTOR-DRIVEN CENTRIFUGAL PUMP

is nearly empty, when the float drops back to the switch-operating position.

For service where electric current is not available, a similar pump has been developed, equipped with a pulley for belt drive by any available power. This pump is manufactured by the Goulds Manufacturing Co., Seneca Falls, N. Y.

Mueller-Cash Reducing and Regulating Valves

The H. Mueller Manufacturing Co., Decatur, Ill., makes a line of reducing and regulating valves for all kinds and conditions of service. Those illustrated in the accompanying figures represent a part of that line.

Figs. 1 and 2 show the single-seat self-contained valve for service in connection with water, steam, air, gas, oil, etc. It is especially adapted for use with water where considerable reduction and close regulation are required and for steam-heating systems down to any pressure as low as 5 lb. absolute.

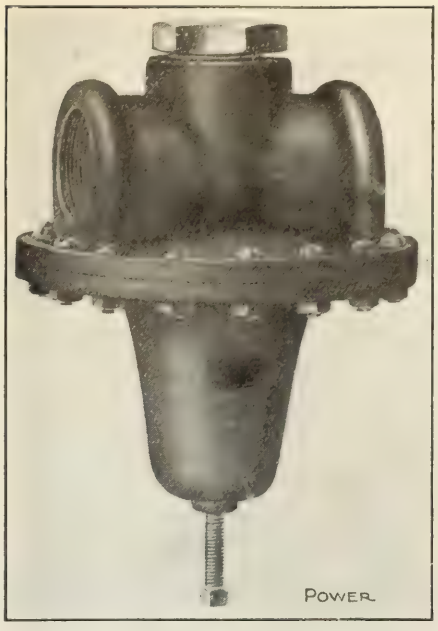


FIG. 1. THE MUELLER-CASH SINGLE-SEAT VALVE

The principle upon which this valve operates is as follows: The pressure on the discharge side of the valve acting upon the diaphragm *A*, Fig. 2, is opposed by the spring *B*, the tension of which may be adjusted by turning the screw *C*. As the pressure on the discharge side falls the spring *B* extends and through the pressure plate and pusher posts *D*, raises the disk off its seat. The small spring *E* furnishes the pressure to seat the disk against the difference in pressure between the admission and discharge sides of the valve. Where necessary a pop safety valve may be installed on the cap *F* to prevent excessive pressure on the

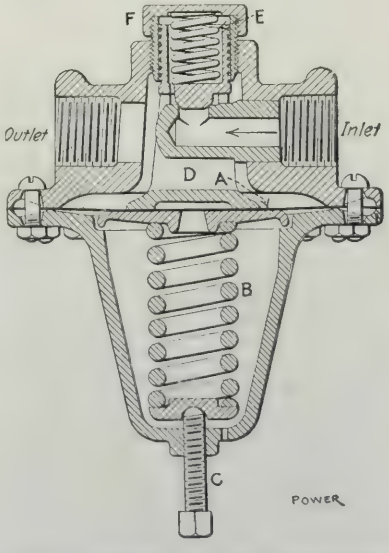


FIG. 2. SECTION OF THE SINGLE-SEAT VALVE

discharge side caused by any obstruction lodging on the seat.

The valve shown in Figs. 3 and 4 may be used as a pump governor, for general power purposes, on blower engines and as a reducing and regulating valve on water, steam, air, gas, oil and similar services with low, intermediate or high initial pressures reducing to low or intermediate delivery pressures.

This valve has a double outside spring which acts against the pressure in the

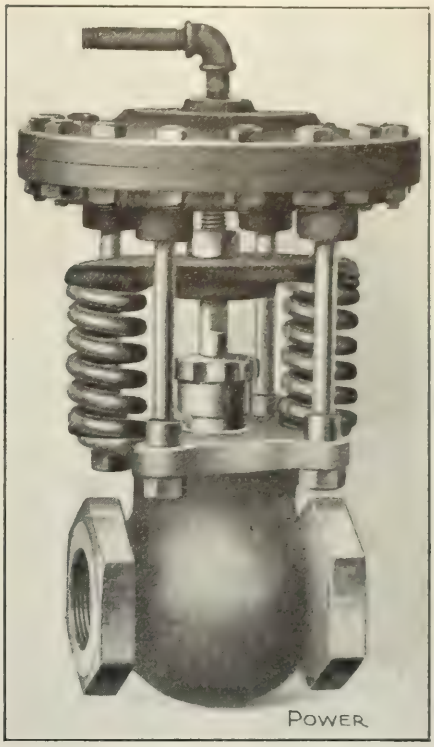


FIG. 3. DOUBLE OUTSIDE-SPRING VALVE

diaphragm chamber *G*, Fig. 4, which connects with the discharge side of the valve. Stem *H* and nut I adjust the tension.

Figs. 5 and 6 show the reducing and regulating and balanced relief valve

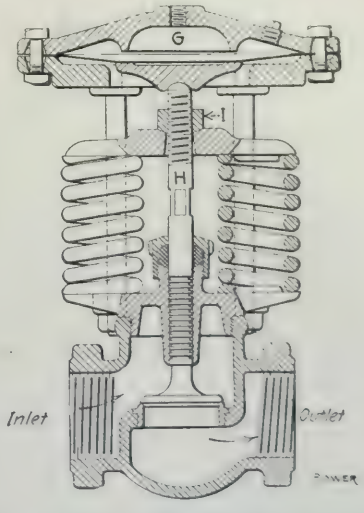


FIG. 4. SECTION OF THE DOUBLE OUTSIDE-SPRING VALVE

which is made for use on very exacting service. This is identical in construction with that shown in Figs. 3 and 4, except that it is controlled by a small sensitive

pilot or auxiliary valve which operates the main valve by admitting the full initial pressure to the diaphragm chamber, causing it to close or open with a very slight change in delivery pressure,

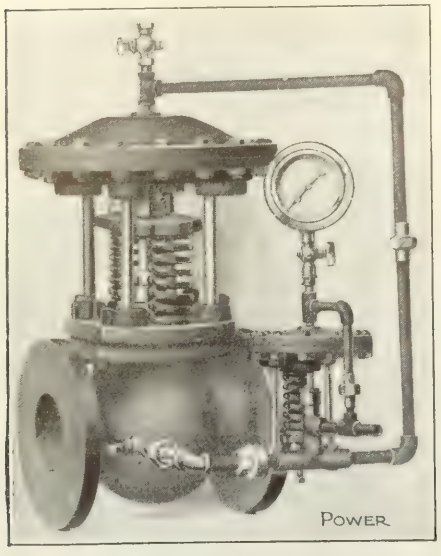


FIG. 5. REDUCING VALVE WITH AUXILIARY PILOT VALVE

regardless of variations in volume of delivery.

These valves are now used as relief valves between new installations of high-pressure boilers and old low-pressure

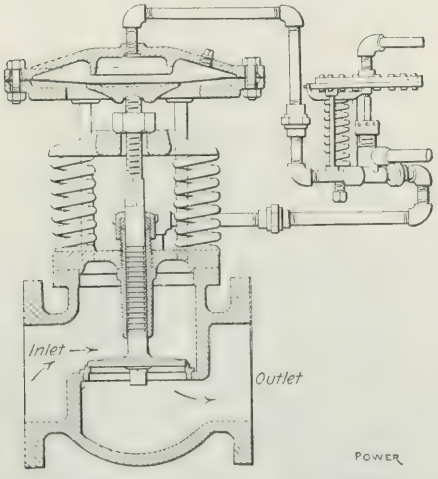


FIG. 6. SECTION OF THE PILOT VALVE TYPE OF REDUCING VALVE

boilers, in such manner that any surplus of steam from the high-pressure system will discharge into the low-pressure system, thereby helping to supply any deficiency in the low-pressure boilers. Valves for the last named purpose are of the same construction as those for reducing purposes except for a slight difference in the construction of the auxiliary valve.

The Pennsylvania Railroad System paid $180,577.73 to employees in July. Since the establishment of its pension fund $33,704,578.25 have been paid out.

Smokeless Combustion of Coal

BY W. B. WILKINSON

To insure as near perfect combustion of bituminous coal as possible, 4 lb. of oxygen must be supplied to the burning fuel to successfully burn the escaping carbon. With the ordinary method of supplying air from the atmosphere, at the temperature of the boiler room, through the ashpit doors under the grate, only about 1 lb. of oxygen is supplied to the burning fuel after passing up through the body on the grates; consequently only one-fourth of the carbon is ignited and burned; the other three-fourths escape in the form of free carbon or black smoke with the gases escaping to the chimney. The problem is therefore to supply the necessary amount of oxygen to the coal. If an additional amount of air is blown in the ashpit by a fan, it is not evenly distributed to all parts of the burning fuel, especially in case of grates having air spaces for burning hard coal; consequently holes are burned through the fuel bed in spots, and a larger part of the oxygen escapes without coming in contact with the carbon. There is also the waste of energy expended in producing the forced draft.

The velocity by which the air is forced into the furnace also prevents it from being heated or raised to a temperature to free the oxygen from the nitrogen. It is, therefore, apparent that to get the best results the required excess of oxygen must be supplied in some way to the combustion chamber over the burning coal, and at a temperature so high that on its injection into the furnace the oxygen will be dissociated from the nitrogen. This requires that the air should be raised to a temperature of about 400 to 500 deg. F., so that upon entering the furnace where the temperature is much higher, the oxygen will be free to ignite the free carbon escaping from the coal.

The problem is where and how to introduce this heated air so as to mingle with the burning coal and escaping carbon, which, being of a higher specific gravity, is rapidly drawn away from the fire bed by the chimney draft and must be retarded and made to mix with the oxygen.

One method is to use a fan to supply air under an automatically regulated pressure, both under and over the fire, through conduits of suitable size, with discharging ports at regular intervals under the grate, and also distributing air over the fire or into the furnace at a point to meet the escaping carbon and gases before they pass to the combustion chamber. The air supply can be heated by contact with the steam used in operating the fan and also by passage through conduits which are heated by contact with or reflected heat from the incandescent coal and burning gases.

With a given boiler or a power plant where the boilers are sufficiently large to do the work required, good draft, good quality of coal, with shaking grates of the right size to burn the coal properly, so as to keep the fire bed clean and moderately low, and where intelligent firemen are employed, the foregoing method of supplying air to the furnace will meet all requirements necessary to burn bituminous coal without objectionable smoke, and will show great economy over ordinary or natural conditions.

Many prefer to burn soft coal with the primary advantage of obtaining more power from the boiler or increased evaporation capacity when the boilers are overloaded or when the draft is defective. In such cases another element is needed to change the chemical condition of the furnace to obtain a higher temperature of the gases passing through or around the boiler tubes and so insure a higher rate of evaporation. This element can best be supplied by hydrogen gas formed from superheated steam injected into the furnace under regulated pressure, operating automatically so as to avoid waste when not required.

A certain amount of steam supplied by the fan engine mingles with the air, but it is not enough for all requirements; therefore a steam superheater retort can be placed in the brick wall over the fire-door arch. This retort could be perforated at intervals with holes of proper size, and the steam jets ejected into the fire chamber so inclined as to give the best results in accord with the conditions of draft, coal used and the requirements of the boiler. Steam itself does not enter the fire chamber to the detriment of the brick walls, grates and boiler shells, but is superheated to a degree necessary to separate the hydrogen when ejected into the furnace. The hydrogen, as a new and additional element, naturally aids the evaporation capacity of the flue gases by adding additional heat units, the thermal equivalent of hydrogen gas being double that of coal gases.

Thus the conditions required to completely prevent any objection from the use of soft coal is provided, but as no two boilers ever operate or steam alike, so no two adaptations of the apparatus can be made alike. Each equipment must be made to meet the individual requirements, necessitating careful study of the plant to be equipped.

The amount of economy, prevention, etc., depends not on the equipment altogether, but also on the ability of the men in charge to operate it properly after it is turned over to them. If they have faith in it and will follow instructions and profit by experience, they will get better results day by day. If equipments are made for smoke prevention, a few

trials will prove the results by observation. Increased draft, increased power and economy of coal can be determined by evaporation tests or by comparison with previous records.

Barr Unit Compound Compressor

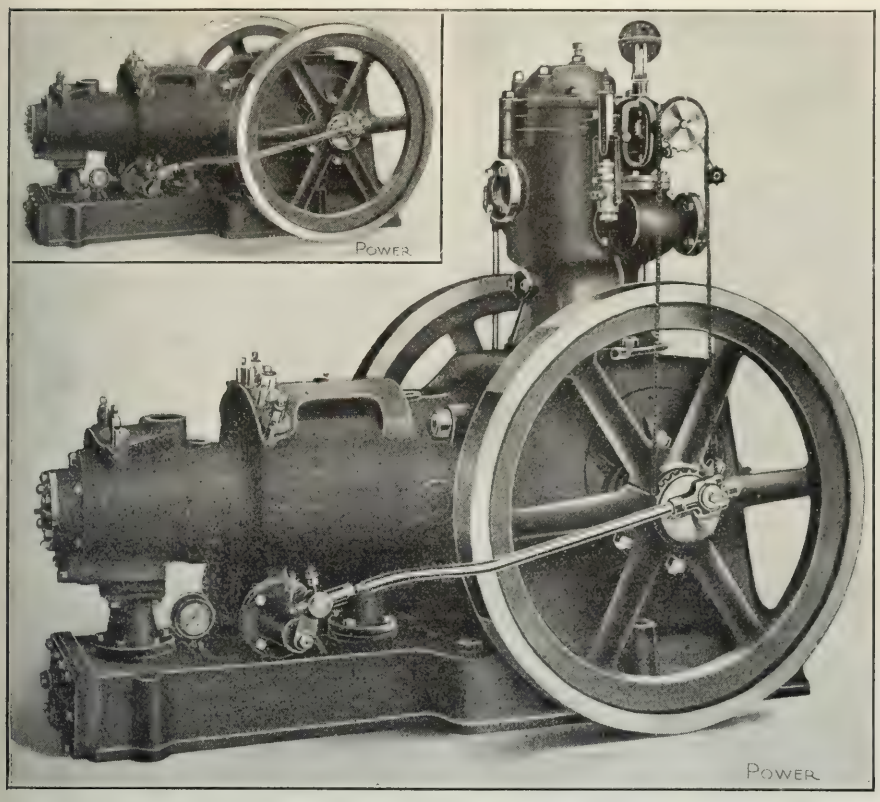
Four types of air compressors, built by the Pennsylvania Pneumatic Co., Erie, Penn., are shown herewith. They are made steam, belt and motor driven, with single and twin cylinders.

In Figs. 1 and 2 are shown single-cylinder, belt and steam driven compressors, the latter being of the angle type width the steam cylinder set at right angles with the air cylinder. Figs. 3 and 4 show a twin, belt-driven, and a single motor-driven compressor, respectively.

The cylinders are supported by a heavy inclosed bed, also the crankshaft bearings. The bed is provided with a cover plate. Both steam and air cylinders are accessible independently and are made single acting and fitted with a one-piece trunk piston. The air cylinder combines a high- and low-pressure bore and compresses in two stages.

The intercooler is of the sectional type with hollow water-cooled units of cast iron. It is located in the bed extension.

A balanced, double-ported, quick-opening steam valve is located in the cylinder head, which is steam jacketed. The steam and air valve-gears are outside connected. The discharge valves are of the one-piece cushioned type.



FIGS. 1 AND 2. SINGLE STEAM AND BELT DRIVEN COMPRESSOR

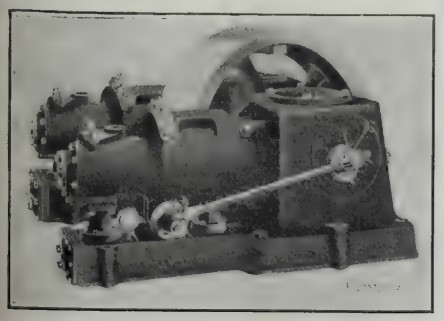


FIG. 3. TWIN BELT DRIVEN COMPRESSOR

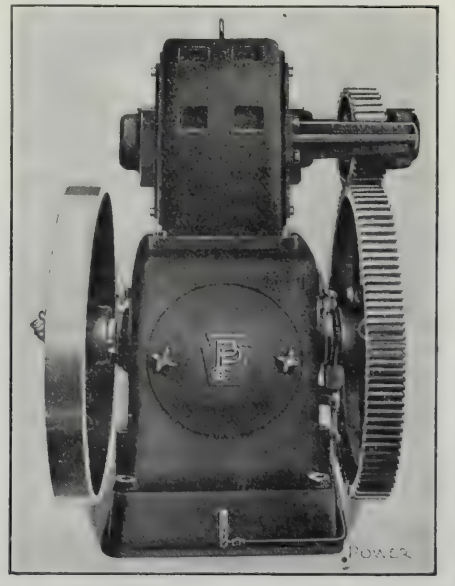


FIG. 4. SINGLE MOTOR DRIVEN COMPRESSOR

Teaching Engineering in Turkey

Under the direction of Prof. John R. Allen, Robert College, in Constantinople, has added an engineering department. It will chiefly be developed along mechanical and electrical lines, including water-power developments and irrigation, says the Newark *Evening News*.

The first wing of the new building under construction will be 56x115 ft. and four stories high. The forge shop will have down-draft forges blown by an electric fan, and a cutoff shear and punch. The foundry will be on the second floor. The cupola will be in the court and have a spout entering the foundry room. This room will be arranged for both floor and bench molding and equipped for making brass castings as well as iron. On the third floor will be the machine shop, with an engine lathe, a gap lathe, speed lathe, drill press, large planer shaper, milling machine, including all necessary tools and attachments, each group-driven electrically.

Power will be obtained from a central power house. The boiler house, now under construction, will contain two 150-hp. boilers, with room for a third to be installed later. These boilers will furnish steam for the electric lighting engines and for the heating.

The east wing of the engineering building will be the same size as the west or shop wing, and contain the power plant and experimental steam, electric and hydraulic laboratories. The first floor will be used for the power plant. Machinery will be available for instructing students in the operation and testing of a power house. On this same floor will be the larger gas engines and the hydraulic laboratory.

In the basement will be the feed pumps and condensing apparatus. The second floor forms a gallery for the first floor. The smaller gas and gasoline engines, the experimental electric motors and generators will be placed on this gallery. The third floor will be devoted to the calibration of instruments, computing rooms, special laboratories and offices. On the top floor will be drawing rooms, lecture rooms, museum, library and recitation rooms. This building will be built as soon as the growth of the department makes it necessary.

In addition to the strictly engineering work, it is proposed to establish classes in manual training for the younger boys. This will stimulate their interest in mechanics, besides giving them some knowledge of the manufacturing processes. The woodworking and pattern department on the fourth floor will contain a planer, a surfacer, a rip and crosscut saw, band saw, two speed lathes, a universal trimmer and a molding and mortising machine. Each machine will be driven by its own motor.

The First Electric Light Plant

Sept. 4 was the thirtieth anniversary of the beginning of commercial incandescent lighting. In 1882, Thomas A. Edison placed in operation in New York the world's pioneer central station.

From a single generator of 125 hp. in an old brick building in lower Pearl St., current was sent through a system of underground cables to some 400 lamps in a territory about a mile square. With only two interruptions of less than 12 hours the service has continued until it now extends from the Battery to Yonkers, includes about 1300 miles of underground transmission lines and feeders, supplies 129,000 customers and has the equivalent of 11,000,000 lamps.

Instead of the old brick building that housed the generating apparatus there are now two structures covering two city blocks, while 31 substations are required to convert the high-tension transmission voltage to the low voltage for distribution; and the historic "jumbo" of 125 hp. has yielded ground to a 30,000-hp. steam turbine.

The Edison company of today is the successor of the company that under the personal direction of Mr. Edison established the world's first permanent central station. Two years of preliminary work saw the establishment of a steam-generating station, the extension of a distribution system of some 15 miles and the installation of about 400 incandescent lamps. This system now has 1300 miles of cable, 360 of which are of the high-tension transmission system, while from 400 incandescent lamps there are about 5,250,000, the connected load equaling 714,000 hp.

Steam boilers were installed in the basement of the reconstructed building at 257 Pearl St. and the six generating units, each driven by a 125-hp. engine, were set up on the second floor. These were the now historic "jumbos," the first type of commercial generating machines in the world.

During the summer of the same year the underground distribution system was laid out and the wiring installed in the buildings of prospective customers. The wiring, laying the street mains and installing the generating apparatus were all done under Mr. Edison's personal supervision.

At 3 p.m., Sept. 4, the current was turned into the mains and to the buildings of the "first customers." A New York paper said at the time: "The Edison central station was yesterday one of the busiest places downtown and Mr. Edison was by far the busiest man in the station. According to Mr. Edison, they will go on forever, unless stopped by an earthquake."

On Jan. 2, 1890, fire destroyed the building. Only one dynamo, No. 6, was saved. That fire put the lighting system out of business for less than half a day, because the Liberty St. annex, opened in 1887, was able to take up the load, and by placing certain restriction on the use of current, carry the burden until new machinery could be installed at Pearl St. In 1890 another annex to the Pearl St. station was opened in the Produce Exchange Building.

In the meantime the new Duane-Pearl St. station, the fourth built to meet the increasing demand, had been completed, and in 1895, the old station was dismantled and sold. Old No. 6 (Fig. 2) is now treasured by the New York Edison Co. as a relic of the early days of the lighting industry.

The Waterside stations, built in 1900 and 1905, are the largest generating sta-

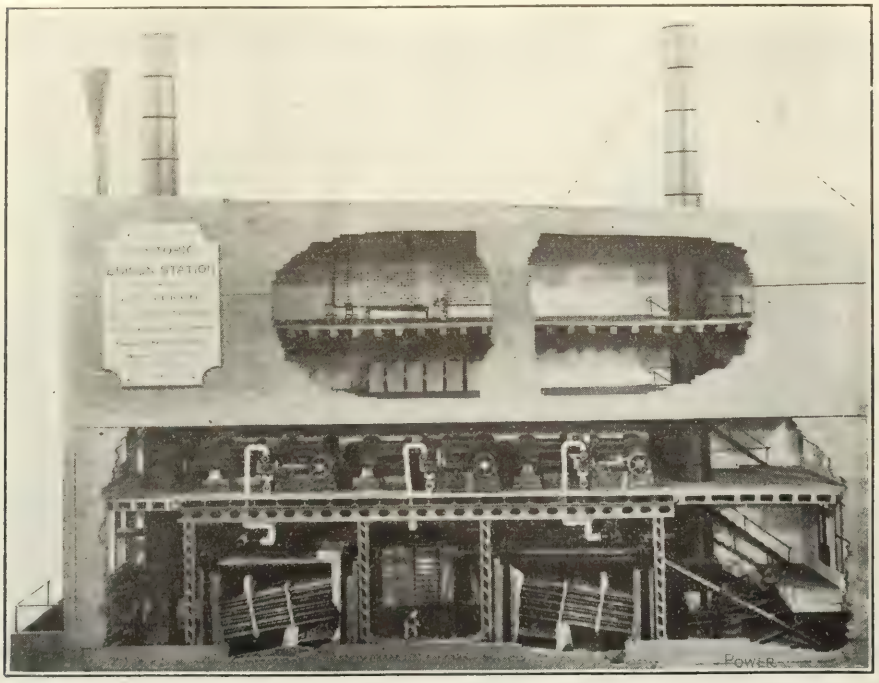


FIG. 1. MODEL OF THE FIRST LIGHTING PLANT

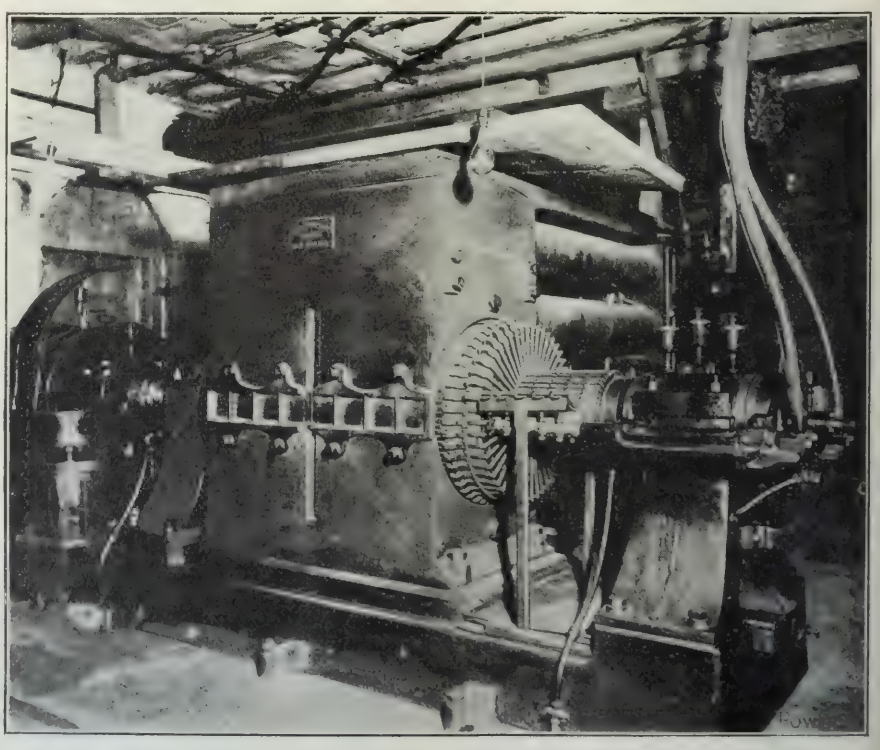


FIG. 2. ONE OF THE OLD-TIME "JUMBOS"

tions in the world, occupying two city blocks on the East River front and having a capacity of approximately 700,000 hp. They light 5,245,000 incandescent lamps, 40,400 arc lights on the system and supply 337,200 hp. in motors, while 159,000 meters are required to measure the current.

Although the Edison was the first organized company to do commercial electric lighting, there were earlier instances of incandescent lighting, all based on the inventions of Mr. Edison. At his home in Menlo Park he had an underground system supplying over 400 lamps. In 1897 a lighting system had been installed on the steamship "Columbia," while less than a month prior to the opening of the Pearl St. station a small generator of a different type was placed in operation at Appleton, Wis., where a waterfall supplied the power.

Non Bypass Valve for Large Steam Lines

By F. E. Sanborn

Steam engineers and others having to do with large steam lines will be interested in a valve which is being used in England. In this country it has been customary to use a large, heavy and consequently expensive valve of the size of the pipe in which it is placed. On such a valve is a bypass with its valve.

The valve now used in England is deserving of consideration. Its flanges are for the size of pipe used, but the run of the valve is shaped something like a Venturi meter. The valve gate is placed in the constricted part and is shown in its closed position in Fig. 1.

When the gate is raised by the handwheel a ring is brought up from below, filling the space in the metal and forming a smooth passage for the steam, as shown in Fig. 2. For pipe sizes up to and including 12 in., no bypass is needed to equalize the pressure in order to open the valve. For sizes above 12 in. a bypass is used.

The advantages of this valve include less cost, the expense being from 50 to 65 per cent., including duty, of that for the American bypass valve, outside stem and yoke; less leakage, as the perimeter of the valve disk and seat is about one-half as long; protection of seats when steam is flowing; smooth passage for steam and so fewer eddies in the current; less work in opening and closing the valve; and less weight.

The valve body may be had in cast iron or cast steel. The valve seats and disks are of a special metal as hard as nickel-steel, and are renewable. These are suitable for use with superheated steam.

Inquiries of several users of the valve in England developed the fact that it was giving satisfaction. It seems to be little, if at all, used in this country and yet is a valve with which an uptodate steam user or engineer should be familiar.

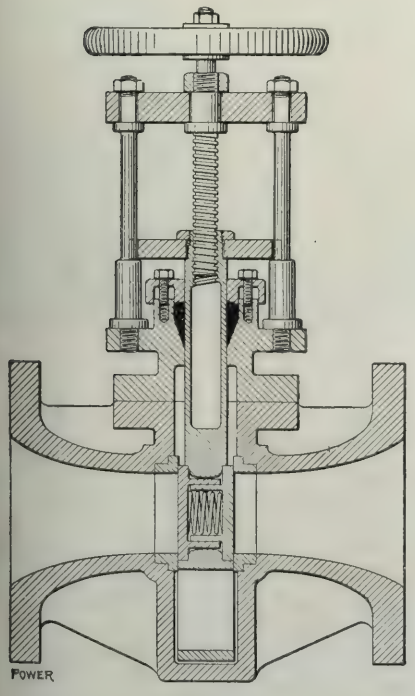


Fig. 1

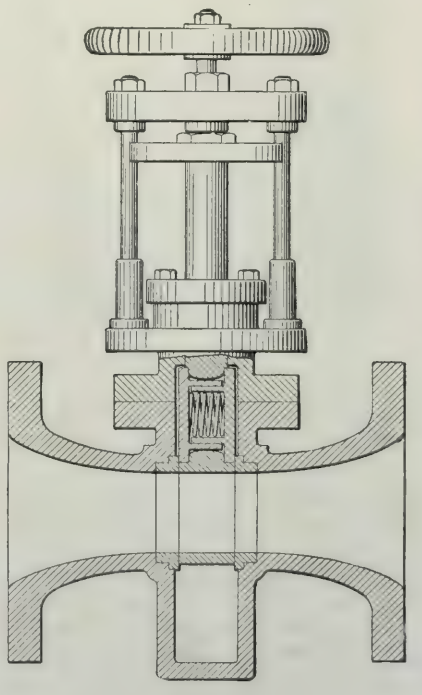


Fig. 2

Non Bypass Valve for Large Steam Lines

The National Association of Stationary Engineers Convention

The thirtieth annual convention of the National Association of Stationary Engineers was held at Kansas City during the week commencing Sept. 8. Notwithstanding the distance of the convention city from the geographical center of the membership and from the numerous Eastern supply houses, the occasion brought together 370 delegates from all over the United States and some 150 establishments took part in the exposition, which has become a feature of these conventions. Their representatives with the families and guests of the delegates brought the total attendance to upward of 1000.

The sessions were held in the New Casino, a hall of generous proportions almost contiguous to the headquarters hotel. The spacious hall, built for the Democratic National Convention of 1900 and but a few blocks removed furnished an ideal place for the exposition.

On Monday evening an informal reception in the parlors of the Coates House gave an opportunity for the renewal of old and the making of new acquaintanceships. A concert by Hiner's Band was an enjoyable feature of this and other evenings.

The opening session of the convention was called to order at 10 o'clock on Tuesday morning by W. F. Groom. chairman of the local committee, and after an invocation by the Rev. E. B. Olmstead, Mayor Henry L. Jost, of Kansas City, welcomed the delegates, and National President Edward H. Kearney responded. Addresses of welcome were also made by ex-Senator J. K. Cubbison and John H. Atwood with responses by National Vice-President John F. McGrath and Past National President Carl S. Pearse.

At the close of the opening exercises the audience proceeded to Convention Hall where the exposition was formally opened by Vice-President McGrath.

President's Address

The address of President Kearney dealt at length with the educational work of the association. The previous convention had authorized him to appoint a committee to report at this convention a "Plan of Education." He could not brook a year's delay and appointed the committee consisting of Professors Edward F. Miller and Joseph C. Riley, Massachusetts Institute of Technology, and John W Lane, editor of the official organ of the association, and authorized the immediate establishment of a course of instruction under their direction. He recommended establishing a permanent educational board to develop a system which may be amplified and extended from year to year. He suggested also

an information bureau to which members might turn when in need of information "such as is sold at prohibitive prices by consulting engineers."

He believed that energetic and concerted action by individuals and associations will do more for the passage of license laws than the lavish expenditure of money.

State associations are here to stay and must be recognized as an all important factor in the work which the National Association is carrying out. "Let us hope," said he, that from year to year state associations will continue to gain in membership and importance and that their task will not be considered as complete until a staff of lecturers has been provided to meet the requirements of subordinate associations and a license and boiler-inspection law has been enacted in every state where there is none in existence."

He recommended compiling and publishing a history of the association.

Reports

National Deputy John F. McGrath reported that during the year 2085 members had been added to the association, 1035 members had been lost, leaving a net gain of 1050 and making the present membership 19,054.

C. H. Wirmell, chairman of the license committee, told of the efforts made in New York, Delaware, Kentucky, Maryland and Michigan to secure license laws with defeat all along the line. He attended the meeting of the Commission on Uniform State Laws in Milwaukee, and, with a representative from the American Boiler Manufacturers' Association, presented a brief covering the essential reasons for uniform laws relating to boiler inspection and the licensing of engineers, which was referred to a committee to report on at the next conference.

Prof. Riley, reporting for the Educational Advisory Board, said that in December a series of articles was printed in the official organ of the association calling attention to the desirability of studying operating costs in power plants. Subordinate associations were invited to arrange their programs so that each month a different type of plant could be discussed and questions appearing in the *National Engineer* be answered by the members. Cash prizes were offered dealing with operating costs in power plants of any type. Other prizes were offered to associations for similar papers embodying their joint opinion.

No response was received from any association to this invitation, but a number of individual papers, some of a high order of merit, were received.

The first prize, $125, was won by George F. Wheaton, of Rhode Island No. 2, for a paper dealing with power costs in textile mills.

The second prize, $75, went to E. S. Clark, of Ohio No. 1, for a paper dealing with the cost of producing power in a paper mill.

The third prize, $50, was awarded to R. L. Ellis, of Alabama No. 1, for an essay upon the general subject of power-plant costs.

The fourth prize, $25, was won by Norman S. Campbell, of Michigan No. 1, for a paper on the cost of hydraulic and electric-elevator systems.

C. L. Finch, of Massachusetts No. 14, received honorable mention.

Slide-rule classes were advocated and several were formed. Prizes in the form of slide rules to the amount of $137.50 were awarded.

Prizes of $50, $25, $15 and $10 were awarded respectively to the Illinois, Wisconsin, Colorado and Iowa state associations for the best convention programs from the educational standpoint.

The committee believes that the proper place for the educational work to be done is in the local associations and recommends that the national association assist these associations by publishing in the society's organ, lists of questions for discussion, furnishing lantern slides for illustrated lectures, etc.

Prof. Riley also delivered a lecture illustrated by lantern slides on the "Action of Gas in the Gas Engine," and C. R. Foster, of the Buckeye Electric Co., of Cleveland, one on "Modern Factory Illumination."

The convention passed with enthusiasm a resolution presented by Thomas W. Roberts, Cleveland, pledging the moral and financial assistance of the association to engineers, who were opposing unjust encroachments and discriminations by the central station, whether for light, power or heating.

The Life and Accident Department has a membership of 3272. Amendments requiring a medical examination were passed. J. D. Taylor was reëlected secretary-treasurer.

The mileage of the delegates to Kansas City cost the association $17,134.10. The convention authorized the formation of district associations and in view of the growing importance of state associations and the increasing cost of the national convention appointed a committee to consider and report upon the advisability of making the conventions of the national association biennial.

Results of Elections

The officers elected were: John W McGrath, Chicago, president; James R Coe, New York, vice-president; Frec Raven, Chicago, secretary; Samuel B Forse, Pittsburgh, treasurer; William J Reynolds, Hoboken, trustee; George A Camblin, Denver, conductor; F. W. Fisher Knoxville, doorkeeper.

The National Exhibitors' Associatio

The Exhibits at the N. A. S. E. Kansas City Convention

elected the following officers for the en suing year:

M. B. Skinner, president; Homer Whelp ley, vice-president; Henry A. Pastore secretary; J. M. Chapman, treasurer John H. Foote, Harry D. Raymond Charles A. Wilhoft, Paul Payne an Charles Cullen, members of the executiv committee.

The Ladies Auxiliary elected the fol lowing officers:

Mrs. Alice Heckler, president; Mrs Sophia Moran, vice-president; Mrs. Mar garet Flinn, secretary; Mrs. Emma M Stone, treasurer; Mrs. Emily A. Dill chaplain; Mrs. Fred Felderman, con ductor Mrs. Gertrude Hunt, doorkeeper

The program for the entertainment o the convention comprised a reception b the Ladies Auxiliary at the Coates Hous with an entertainment under the direc tion of Laura Valworth Hull Tuesda

evening; a picnic at Fairmount Park Wednesday afternoon and evening; an entertainment by the National Exhibitors' Association Thursday evening, and a ball Friday evening. Special entertainment, including automobile rides, was provided for the ladies. The usual presentations were made to the retiring presidents of the N. A. S. E., the Exhibitors Association and the Ladies Auxiliary and to the chairman of the local committee, and the usual votes of thanks were passed, in this case exceptionally well merited for the attentions received and hospitality enjoyed.

The next annual convention will be held at Springfield, Mass., the second week in September, 1913.

[We regret that failure to receive the photograph prevented our having the usual view of delegates and visitors.—EDITOR.]

EXHIBITORS

Following is a list of those who had space in Exposition Hall:

Albany Lubricating Co., American Steam Gauge & Valve Manufacturing Co., The Anchor Packing Co., V. D. Anderson Co., Armstrong Cork Co., Arrow Boiler Compound Co., The Ashton Valve Co., Builders Iron Foundry Co., Bishop, Babcock, Becker Co., Bowers Rubber Works, S. F. Bowser Co., Inc., Buffalo Forge Co., Central Coal & Coke Co., A. W. Chesterton Co., The Chapman Valve Manufacturing Co., Clement Restein Co., Cling Surface Co., Consolidated Engineering Co., Crandall Packing Co., Dearborn Drug & Chemical Works, Diamond Power Specialty Co., Joseph Dixon Crucible Co., Durabla Manufacturing Co., Economy Fuse & Manufacturing Co., Elliott Co., The English Tool & Supply Co., The Fairbanks Co., Falls Machine Co., Fisher Governor Co., Flexitallic Gasket Co., France Packing Co., Garlock Packing Co., Greene, Tweed & Co., Gustin Bacon Manufacturing Co., Harrison Safety Boiler Works, Hawkes Boiler Co., Hawkeye Compound Co., Heine Safety Boiler Co., Hill Pump Valve Co., Hills McCanna Co., Homestead Valve Manufacturing Co., Howe Rubber Co., Illinois Malleable Iron Co., The Imperial Brass Manufacturing Co., International Correspondence Schools, Jenkins Bros., H. W. Johns-Manville Co., Kansas City Engineering Co., Kansas City Rubber & Belting Co., Kent Lubricating Co., Keystone Lubricating Co., Lagonda Manufacturing Co., A Lescher Sons, Liberty Manufacturing Co., Lunkenheimer Co., Mason Regulator Co., The Mayer Coal Co., McCord Manufacturing Co., James McCrea & Co., McLeod & Henry Co., W. B. McVicker Co., *Mining & Engineering World,* Missouri Fidelity & Casualty Co., Morehead Manufacturing Co., Murray Iron Works, *National Engineer,* The New York Belting & Packing Co., Nicol Smokeless Grate Co., Ohio Blower Co., The Ohio Brass Co., Ohio Grease Lubricating Co., Ohio Injector Co., Otis Elevator Co., Palmer Forced Draft Slack Burner Co., Peerless Rubber Manufacturing Co., Penberthy Injector Co., The Perolin Co. of America, Peterson Engineering Co., Philadelphia Grease Manufacturing Co., POWER, *Practical Engineer,* Quaker City Rubber Co., Reeves & Skinner Machine Co., Revere Rubber Co., Richardson-Phenix Co., Ridgway Dynamo & Engine Co., John A. Roebling's Sons Co., Rogers Metal Works, Rosedale Foundry & Machine Co., Schaeffer & Budenberg Manufacturing Co., *Southern Engineer,* C. E. Squires Co., Standard Oil Co., Standard Regulator Co., Strong, Carlisle & Hammond Co., Under-Feed Stoker Co. of America, United States Graphite Co., Western Boiler Compound & Chemical Co., Western Tire & Rubber Co., Westinghouse Air Brake Co., Westinghouse Machine Co., White Star Refining Co., The Williams Gauge Co., Yale & Towne Manufacturing Co.

Joint Meeting of Engineers in Germany, 1913

Elaborate preparations are in progress for the entertainment of the American Society of Mechanical Engineers on its visit to Leipzig, Germany, to meet with the Verein deutscher Ingenieure in June of next year. While it is early to announce the features of the program, it may be said that the attractions include a reception at Hamburg by the senate, another by the Hamburg-American Line, and a visit to the shipyards of Blohm & Vose, the largest shipbuilders on the continent, where an opportunity will be offered to inspect the "Super-Imperator," the largest steamship yet under construction, and to see Diesel engines of 6000 hp. each, 2000 hp. being developed in one cylinder.

The party will then be taken by special train to Leipzig, where it will be received by the King of Saxony and the first of the professional sessions will be held. A paper is promised by Prof. Lamprecht, of the University of Leipzig, the famous historian, one by Dr. C. Matschoss, probably on the history of engineering, and another by Prof. Busley on aërial navigation. It is hoped that Count Zeppelin will be present with one of his dirigible balloons. There will also be a concert by the famous Gewandhaus orchestra under Nikisch, the most famous living conductor in Europe. While the party is in Leipzig, the dedication of the monument erected in commemoration of the battle of Leipzig 100 years ago will take place. The next stop will be at Dresden, where the castles, galleries and technical high schools will be visited. From here the party will proceed to Berlin, where the program will comprise an excursion through the castle of the Kaiser at Pottsdam, a great lake festival at Wannsee, and a reception at the Zoological Gardens, as well as visits to the Allgemeine Electricitäts Gesellschaft, Borsig, Ludwig-Lowe, Siemens-Schuckert and others. A special train will then be taken to Essen, where the party will be received by Krupp von Bohlen, the president of the Krupp works, and exceptional facilities will be given to see the latest processes in steel manufacture, etc. A visit is also planned to the Krupp works at Rheinhausen, which are the finest steel works in Germany, and an opportunity will be afforded to inspect the entire process from the raw material to the finished product. At Düsseldorf the party will be received by the iron and steel institutes, and at Cologne by the city authorities, the reception in the latter city being held in the famous Old Guild Hall. From Cologne visits will be made to the establishment of Deutz, Humboldt. From Cologne the party will go up the Rhein by steamer to Rüdeschim, where a festival will be held on the banks of the river. At Heidelberg an illumination of the castle is promised. At Mannheim, visits will be made through the shops of Sulzer Bros., Lanz, and Brown Boveri. The party will then proceed to Munich with a possible stop at Nürnberg. At Munich the professional sessions will be continued in the famous industrial museum, and there will be receptions by the city and by the Prince of Bavaria.

Discontinuing the Power Plant

One of the present-day results of electric transmission, especially in large towns and cities, is the offer of electric-power people that tempts the machine woodworker, along with other users of power, to discontinue the power plant and buy current. There is perhaps more hesitancy on the part of woodworkers than anyone else, as they make a certain amount of shavings and dust and thus have fuel for making their own steam power. Yet many mills and factories are cutting out their engines and buying current.

Whether or not anyone should, depends materially on the cost, and the matter of having fuel should be looked at from both sides. Some fuel will be needed anyway, for drying and heating purposes. This, in some plants, will take care of the dust and shavings, and much of the other scrap may be sold for kindling. Some woodworking institutions that have never installed engines of their own, have a boiler for the drykiln and for heating the plant during the winter, and for other uses calling for steam. The dust and shavings are piped to it, a moderate steam pressure is maintained with but little firing trouble, and they get along very nicely. Others have cut out the use of their engines, but retain a little fire under their boilers in the same way. It is compara-

tively easy to do this and try the effect, and it is especially tempting when the engine runs down, gets too small for the work, or is too heavy a tax on the boilers.

Taking it altogether, the way the wind blows now, it looks like the discontinuation of lots of woodworking power plants. Some of large size have already swung over, even if they do have shavings and dust. They use it for the drykiln and heating, and perhaps figure on cleaning up their waste closer than they once did, and selling the balance for kindling.—*The Woodworker.*

To Succeed Long Acre Co.

It has been announced that the Intercity Power Co., of New York City, which was recently incorporated under the laws of Delaware, will take over the entire franchise, property and assets of the Long Acre Electric Light & Power Co. It is also reported that the new company proposes to furnish lighting and motor service throughout Greater New York and will enter into an agreement with the Hudson & Manhattan R.R. Co. to receive all of the latter's surplus electrical energy. No details have yet been announced.

Three Gas Tanks Explode

The explosion of three gas tanks at the A. O. Smith Co. plant, at Milwaukee, Wis., on Aug. 26, killed one employee and injured six others. The tanks stored acetylene gas for welding. The pressure at the time of the explosion was 40 lb. It is believed the gas was ignited by a flashback from a welding torch.

Power House Collapses

Early in the morning of Sept. 8, the old power house of the No. 4 mill of the Lonsdale Co., Lonsdale, R. I., slid into the mill trench, dragging with it a portion of the five-story brick building with which it was connected and causing damage estimated at about $75,000.

The accident came without warning and was evidently caused by the undermining of the foundations. When the power house dropped, a large section of the adjoining building was wrecked. Had the accident occurred a few hours later, when the mill would have begun work, 75 of the power-house employees might have been crushed in the débris. About 600 employees will be out of work until the necessary repairs are made.

The collapse is believed to have been due to the undermining of the big waterwheel, which has furnished part of the mill power during the last two generations. The walls of the wide trench are made of stone, and beneath the power house the waterwheel has rested on these. It is thought that they gave way, releasing the supports at the end of the building.

BOOKS RECEIVED

MACHINE DESIGN. By H. D. Hess. J. B. Lippincott Co., Philadelphia, Pa. Cloth; 364 pages, 5¼x9 in.; 318 illustrations; tables; indexed. Price $5.

SHOP ARITHMETIC. By E. B. Norris and K. G. Smith. McGraw-Hill Book Co., New York. Cloth; 187 pages, 6x9 in.; 86 illustrations; tables; indexed. Price $1.50.

TEXTBOOK OF THEORETICAL NAVAL ARCHITECTURE. By Edward L. Attwood. Longmans, Green, and Co., New York. Cloth; 518 pages, 4¼x7¼ in.; plates; tables; 145 illustrations; indexed. Price $2.50.

NEW PUBLICATIONS

HOUSE WIRING. By Thomas W. Poppe. Published by The Norman W. Henley Publishing Co., New York, 1912. Size 4x6 in.; 100 pages, illustrated. Price 50c.

A little book devoted exclusively to describing and illustrating the methods of installing electric-light wiring and fixtures in modern office buildings and dwellings. It is thoroughly practical in character and, being intended primarily to aid the electrician and helper, no attempt has been made to cover the field of the illuminating engineer; only the simplest problems being given in laying out a lighting system.

THE STEAM TURBINE; Revised fourth edition. By Robert M. Neilson. Published by Longmans, Green & Co., London. Cloth, 651 pages; 9½x6 in.; illustrated. Price, $5, net.

It has been stated that the design and construction of the steam turbine is practically standardized, yet the appearance of new and enlarged editions of the well known manuals of steam-turbine practice has not ceased.

The fourth edition of Robert M. Neilson's "The Steam Turbine," although published as lately as 1908, has now been reissued with additions, increasing its value to a considerable extent. The new matter consists of an extension of the list of patents to include all British patents up to the end of 1909 and an appendix of 35 pages in which the author summarizes the "most important developments" in the last four years.

Mr. Neilson gives considerable space to the combined impulse and reaction type which so many turbine builders have adopted as their standard practice, but naturally puts emphasis on the later developments of the marine turbine, the geared turbine and the "recipro-turbine" in marine work. On page 630 he gives a most interesting set of curves on steam consumptions of modern turbo-generators, which he says are "unlikely to be surpassed to any great extent in the future."

The power-station reproductions are well chosen and valuable.

PERSONAL

F. G. Clark, of New York, has accepted a position as chief engineer of the Toronto Electric Light Co.

J. Albert M. Robinson, formerly engineer of the Brown Portable Elevator Co., is now hydraulic engineer for the American Well Works, First National Bank Building, Chicago.

B. A. Brennan has resigned as sales manager of the power department of the Bethlehem Steel Co. to accept the office of manager of the corporation department of the Mercantile Trust Co., of St. Louis. He began his new duties Sept. 10.

Francis F. Longley has been admitted to partnership in the firm of Hazen & Whipple, consulting civil engineers, New York. Mr. Whipple, who last year became professor of sanitary engineering in Harvard University, retains his membership in the firm and is still open to professional engagements.

Dr. Conrad Matschoss, the European writer of notes on industrial history, historian of the Royal Polytechnic High School of Berlin, is visiting the United States as the official representative of the Verein Deutscher Ingenieure, with special reference to the arrangement for the visit of the American Society of Mechanical Engineers to Germany in June of next year.

Robert F. Pack, formerly general manager of the Toronto Electric Co., Toronto, has been appointed general manager of the Minneapolis General Electric Co. Since the resignation of A. W. Leonard, early in August, the position has been temporarily filled by Samuel Kahn, who will resume his duties in the operating department of H. M. Byllesby & Co. Gen. George H. Harries, vice-president of the Consumers Power Co., will continue to act in a supervisory capacity over all the properties of this corporation. Mr. Pack is president of the Canadian Electric Light Association and a member of the American Institute of Electrical Engineers.

SOCIETY NOTE

On Oct. 25, the National Association of Stationary Engineers will be 30 years old. The subordinate associations of Chicago are planning to hold an informal stag dinner on the evening of that date to celebrate the event and to tender a reception to John F. McGrath, the newly elected national president of the association, who is a member of Illinois No. 28. The following committee is in charge of the arrangements: Messrs. Raven (chairman), Lightly, Moore, Illinois, No. 1; Warren, Noward, Kleynstuber, Illinois, No. 2; Haydin, Barnhardt, Whyte, Illinois, No. 28; Tavernier, Bishoff and Anderson, Illinois, No. 38.

OBITUARY

David M. Forker

On Aug. 29, David M. Forker, vice-president of the D. T. Williams Valve Co., Cincinnati, Ohio, died at his home in that city at the age of 51.

POWER

Vol. 36 NEW YORK, OCTOBER 1, 1912 No. 14

By Their Works Ye Shall Know Them

"PRETTY good advertisement of your outfit and the way you use it," remarked the man who, attracted by the sign, stooped to see how much was ash and how much coal.

The fireman, hearty and good-natured, brawny and begrimed, saw the point and invited the stranger in. He beamed with justifiable pride on the whitewashed walls, the well-kept boiler fronts with their polished bright work, the cleanly swept floor, and the pile of "cinders," which was nothing but fine ash and clinker.

Then he showed his visitor the color of the fire through several peep holes in the side of the setting, explained understandingly the operation of the mechanical stoker which was feeding the fuel in with persistent regularity, and discussed intelligently the effect of different rates of feed of both coal and water, of different thicknesses of firebed, sizes of coal, different intensities of draft, etc. He did not refer to "calories" nor "B.t.u's," nor dally with chemical terms, but he knew what percentages of ash he got from different kinds of coal that he had used, and the efficiency of his plant on a basis of the steam made and the combustible fired.

There were no diplomas or certificates on the wall. The fireman did not boast membership in any union or lodge or society, nor wear any badges to indicate it, but that pile of ashes, his steam gage, and the top of his stack told conclusively that if a fireman is one who understands combustion then he has an unquestioned right to be known as such.

Power Plant of the Stanley Works

By Warren O. Rogers

Account of the development of the steam end of the works, from a 6-hp. engine to compound engines and a low-pressure turbine. The company was the first to install many of the modern steam plant appliances used in the city of New Britain, Conn.

The development of the power plant of the Stanley Works, New Britain, Conn., was largely pioneer work, as many parts of the equipment were the first of their kinds installed and operated in that city.

The original Stanley bolt works was built in 1832 by F. T. Stanley, who ran the first steam engine in that section of the country.

It was a small Burdon engine with a 6-in. cylinder and 24-in. stroke, and developed between 6 and 8 hp. This engine was operated until 15 years ago and is now set up and connected with steam in a room adjoining the main engine room, where it is preserved as a relic, Fig. 1. This was followed by a larger double, slide-valve Burdon engine.

Forty years later, or in 1872, a 125-hp. engine was installed and belted to the factory line shaft. The engine ran noncondensing until 1891, when a jet condenser was put in, the first used in the city.

The load increased and a 600-hp. cross-compound Corliss engine was installed, which was also the first in New Britain and is still in use. In 1905 a 24 and 44x42 in. compound engine was purchased, rated at 1200 hp., running at 120 r.p.m. It is directly connected to a 400-kw. direct-current 250-volt three-wire machine. It is also belt connected to a line of shafting.

In 1910 the first low-pressure steam turbine in the city was installed to run in connection with the large compound engine. The back pressure on the low-pressure cylinder is about 1½ lb. The turbine is connected to a Lablanc condenser which maintains a 28½-in. vacuum. The turbine is of 750 kw. capacity and runs at 1800 r.p.m. with a steam pressure of 150 lb. per square inch. The turbine is set on a concrete foundation, built separate from the building, as shown in Fig. 2. Its condenser is the only one of its type in the city, and, under a full load test run maintained 29 in. of vacuum.

The rated capacity of all generators is 1500 kw. or approximately 2000 hp., a gain in 40 years of 1994 hp.

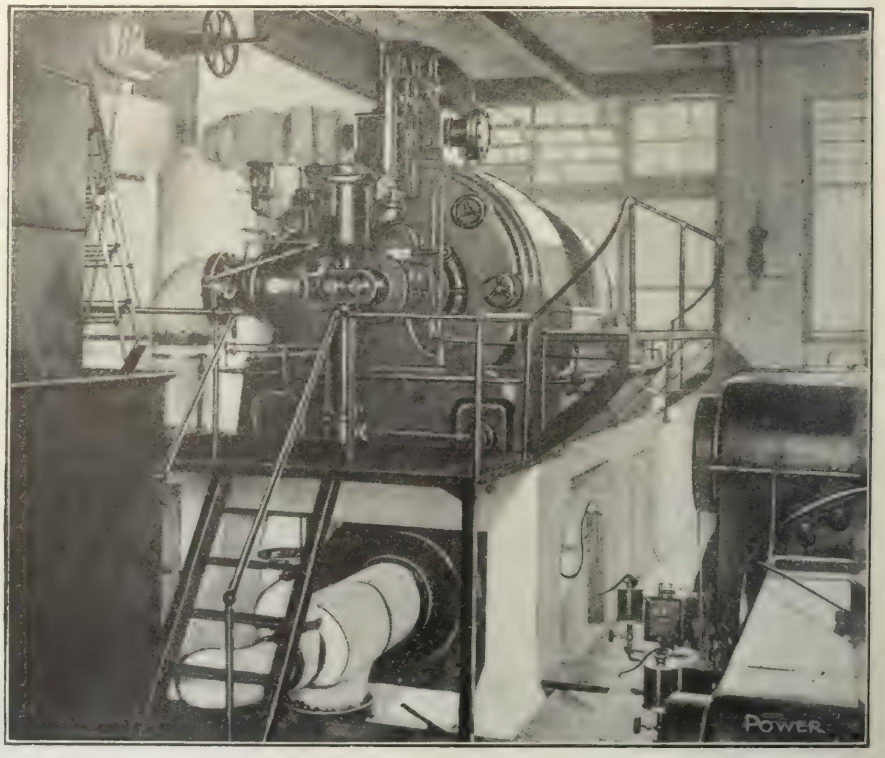


FIG. 2. EXHAUST STEAM TURBINE AND CONCRETE FOUNDATION

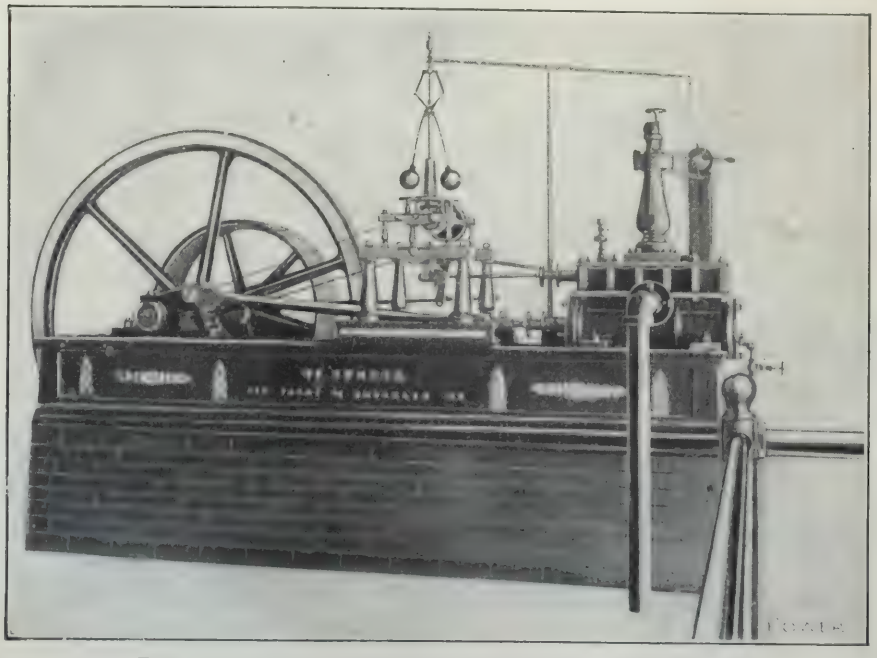


FIG. 1. ORIGINAL POWER ENGINE OF THE STANLEY WORKS

BOILER PLANT

The original plant consisted of Lowe boilers which were soon replaced by horizontal return-tubular boilers but in 1898 the first two water tube boilers used in New Britain were installed and two years later two more boilers of the same type, each having a rating of 338 hp., were put in.

The latter boilers had superheaters, again the first used in the city, and among the first made and used in this country. They are still in operation at the Stanley Works.

In the main boiler house four water-tube boilers are now installed, having a total capacity of 1352 hp.

Steam is carried from the boiler plant through a 600-ft. tunnel to the engine room. In this tunnel are all the steam and water pipes, and also the electric wires. It is said that the loss of steam due to condensation is less than 1 per cent.

The boiler furnaces have the first automatic stokers used in the city. A view of the boiler room is given in Fig. 3.

Back of the boilers are the two fuel economizers. The new boiler house is on Curtis St., but two smaller boilers, one 250-hp. water-tube and one 125-hp. return-tubular boiler are in the small boiler house next to the engine room. This makes a total of over 1700 b.hp. capacity.

The only complete coal conveying system in New Britain is at the new boiler plant. Coal is delivered to an elevated siding so graded that the cars can be easily pushed along by hand and unloaded into a hopper under the track. The empty car is then run to the switchback and returned to a siding, as shown in Fig. 4.

The dumped coal falls into a crusher from which it is delivered to a continuous belt conveyor and carried from the hopper pit to the scale house where an automatic scale registers the amount of coal passing through it. The coal is then carried by a belt conveyor to the boiler-house coal bunkers. The fuel is fed through metal chutes to the hoppers supplying the stokers. Each coal chute is made in five sections, two pieces in the spout, three in the flare or mouth, one on top and two on the bottom. This permits removing any worn section without discarding the entire chute. The boiler plant is laid out for future duplicate units, outlets from the coal bin for which are provided as shown in Fig. 3.

The coal bin has a capacity of 420 tons. A reserve supply of 4000 tons can be stored in the yard. When storing coal, it is discharged from the first section of the conveyor onto the ground. It is then removed to the yard by a grab bucket, operated by a steam hoist.

When taken from the yard, the coal is picked up by the bucket and dumped into a hopper over the second section of the conveyor and carried to the coal bunkers above the boilers.

Ashes from the grate drop into a hopper below the boiler from which they

FIG. 4. COAL CONVEYORS AND RUNBACK TRACKS

are loaded into a small iron car, and hauled out and dumped on the ash pile. This ash car is drawn by a clutch-controlled winch, driven by the stoker engine and a bevel gear attached to the end of the stoker shaft. A second shaft with a winch and clutch attached to one end, is secured to the side of the boiler setting and a gear on the opposite end meshes with the stoker shaft gear. When ash is to be handled the clutch is thrown in and the ash car with its load is pulled up an incline to a 12-ft. level, from which the car runs by gravity to the end of the trestle. If the operator neglects to shut off the power, the link on the end of the cable will be automatically disengaged preventing damage to the driving gear or car.

Fig. 5 shows an appliance for grinding on the caps on the water-tube boilers. It was devised several years ago, by C. E. Hart, then in charge of the boiler plant and has since been patented by him. It bolts on the ends of two adjoining caps and can be put on either horizontally or vertically. It is a labor saver in grinding caps, and especially where they have been leaking and the seats have become somewhat pitted.

A universal joint just above the cap being ground gives the cap a proper fit on its seat, a perfectly ground joint resulting. A slight amount of powdered emery is used between the cap and the seat to facilitate the grinding.

FIG. 5. TUBE PLATE REGRINDING TOOL IN PLACE

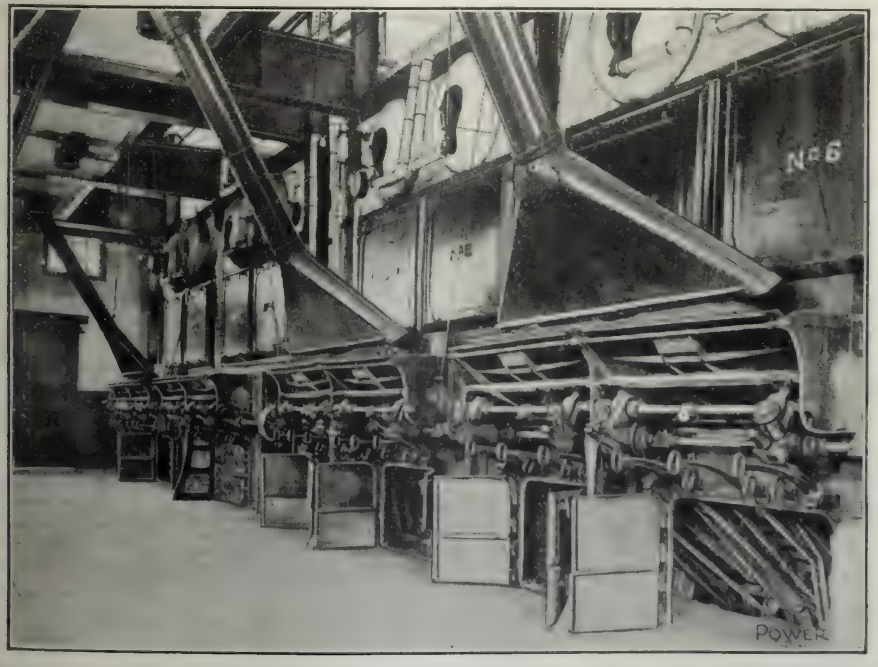


FIG. 3. BOILER ROOM OF THE STANLEY WORKS

SPRAY COOLING SYSTEM

The condensing water for the condensers is cooled by a number of jets, Fig. 6, which give the water a swirling motion, so that it falls to the surface of the pond in a fine spray. This reduces it to practically the same temperature as the atmosphere. It is the only spray-cooling system in New Britain.

FACTORY HEATING

In 1896 the Stanley Works discarded the old methods of steam factory heating and put in the first hot-water heating system in which the water was circulated by pumps. The water is heated by exhaust steam from the small engines, pumps and air compressors, and is handled by either one of two steam-turbine-driven

pumps installed in the boiler room, one of 30 and one of 60 hp. In extremely cold weather the exhaust from one of the reciprocating engines can be used to heat the circulating water in the heating system.

FIG. 6. SPRAY WATER COOLING POND

ELECTRICITY FOR SHOP USE

Up to 1870 electricity was not used at all in any of the city factories, but in this year an 18-arc-light machine was installed and was belted from the main shaft of the factory. In 1891-2 electric power transmission was installed at the works, which was not only the first electrical power transmission for factory use in the city, but was the first in Connecticut. Today practically every department is driven by electric motors.

In 1898 the Stanley company installed the first electric crane to be used in New Britain; there are now six at the works. In January, 1901, an electric locomotive was put in use on the industrial railway in the yard, and in 1903, electricity was still further used to operate electric elevators, also the first in the city.

From the foregoing, it is evident that, although the power plant as a whole is not as compact as might be desired, it comprises the most uptodate steam apparatus that can be purchased.

Figures regarding the operating cost are not available, but it is very probable that the power-plant cost is well below the average.

The author is indebted to E. W. Pelton, chief engineer, for data and photographs.

"Manners, like oil in the bearings of a machine, prevent friction and aid efficiency."

POWER PLANT EQUIPMENT OF THE STANLEY WORKS

No	Equipment	Make	Use	kw.	volts	R.p.m.	Size	Pressure Lt.	Hp.	Manufacturers
1	Simple engine	Burdon	Original unit						6	Unknown
1	Cross compound engine	Frick	Main unit			75	6x14"	150	600	Frick Co.
1	Tandem compound engine	Rice & Sargent	Main unit			120	24x42x42	150	1200	Providence Engineering Co.
1	Simple engine	McEwin	Main unit			200	18x18	150	200	Ridgway Dynamo & Engine Co.
1	Steam turbine	Curtis	Main unit			1800		150	1000	General Electric Co.
1	D. C. Generator	Ridgway	Main unit	200	250	200				Ridgway Dynamo & Electric Co.
1	A. C. Generator	General Electric	Main unit	750	250	1800				General Electric Co.
1	D. C. Generator	Westinghouse	Main unit belt driven	400	250	120				Westinghouse Machine Co.
1	Condenser	Le-blanc	Turbine							Westinghouse Machine Co.
1	Condenser	Blake	Reciprocating engine							Blake Pump & Condenser Co.
23	Spray nozzles	Schutte & Koerting	Cooling water				3"			Schutte & Koerting
1	Steam pump	Worthington	Boiler feed				10x6x10			Henry R. Worthington
1	Steam pump	Worthington	Boiler feed				9x 6x10			Henry R. Worthington
4	Economizers	Green	Boilers and furnaces							Green Fuel Economizer Co.
1	Regulator	XXth Century	Damper							Atwood & Morrill
1	Co^2 recorder	Uehling	Flue gases							Uehling Instrument Co.
1	Meter	Trident	Water				6"			Neptune Meter Co.
1	Filter		Oil							Ross Valve Mfg. Co.
1	Meter	Arrow	Water							Henry R. Worthington
2	Regulators	Crosby	Pumps							Crosby Steam Gage & Valve Co.
4	Boilers	Babcock & Wilcox	Steam generators					155	338	Babcock & Wilcox Co.
1	Boiler	Babcock & Wilcox	Steam generator						250	Babcock & Wilcox Co.
1	Boiler	Return tubular	Steam generator						125	

Pumping Coal

Following the transportation of oil in barrels and tank cars came the more efficient distribution by pipe lines. It is possible that coal may not only be taken out of the mines but transported to considerable distances in the same efficient way.

A movement is on foot in this direction. J. H. Hoadley and W. H. Knight have developed a system whereby the coal instead of being broken or blasted out of the seams in lumps is scraped off from the face of the workings in a granular state suitable for coking. Coal in this finely divided state and mixed with water can be pumped through pipe lines at a cost considerably less than that at which it can be transported by rail. Allowed to settle out of the water but still in a finely divided state, it can be blown into a furnace and burned under conditions of control which approach those of gaseous or liquid fuel.

The coal-mining machine consists of a rotary cutter adapted in diameter to the width of the seam. This is revolved by a direct-connected electric motor and is forced against the face of the seam hydraulically, being given at the same time a sidewise swing through a small angle so that the proper width may be cut. While the cutting is going on a stream of water is directed against the face of the seam, preventing the flying of the dust and keeping the tools in a cool condition.

The machines at present made have a capacity for cutting 100 to 250 tons per day. These will be followed by larger machines which are expected to raise the tonnage to 500. The mixed coal and water are driven by centrifugal pumps with one or more stages according to the lift required. Trials have been made of the machine at the Lewisburg mine in Alabama and it is said to have fulfilled the anticipations of its projectors.

Centrifugal Pumps for Condensers

By Richard L. Strobridge*

Characteristics and operating requirements of circulating, hotwell and tail pumps including a consideration of priming and sealing.

*Mechanical Engineer for Wheeler Condenser & Engineering Co., Carteret, N. J.

The design of a condensing equipment is in each case an individual engineering problem. This applies to the circulating pump, to the hotwell and tail pump, as well as to the condenser itself. The two most important elements to be considered in the design of the centrifugal pump are the capacity in gallons per minute and the dynamic head. The capacity is determined primarily by the size and design of the condenser, the vacuum and the temperature of water supply and the service the pump is to perform—that is, circulating pump, tail pump, or hotwell pump. For hotwell and tail pumps the head corresponds to 28 or 29 in. of vacuum plus any static head and friction head. For circulating pumps, the head is from 10 to 50 ft. and more (where a cooling tower is used), including the friction head lost in the condenser, which depends upon the length and number of passes, size of tubes and, as will be pointed out, the quantity of water being pumped.

Circulating Pumps for Surface Condensers

In a surface-condenser installation the circulating water is drawn from an intake well, canal, etc., through the suction pipe to the impeller of the centrifugal pump, whence it is discharged through the several banks of tubes and then through the discharge pipe to the sewer, discharge canal, or to the cooling tower. Where it is possible to seal the circulating-water discharge pipe, the system operates as a siphon, and the static head is the difference in elevation of intake and discharge canals. Where the discharge cannot be sealed, because of local conditions, the static head is equal to the difference of elevation of intake water and the top pass in the condenser. Where a cooling tower is used the static head is the difference between the elevation of the distributors at the top of the tower, and the suction of the pump. To this static head, in any case, must be added the head lost in friction, to obtain the total or dynamic head.

The head required to overcome friction varies as the square of the velocity of the water, and since for a given condenser the velocity of the water determines the quantity pumped, the head to overcome friction increases also as the square of that quantity. This can be best explained by reference to the curves of Figs. 1 and 2. The static head is 10 ft. and is constant regardless of the quantity pumped. For the rated capacity of the pump, 2500 gal. per min., the friction head with a certain condenser might be, say 15 ft., so that the total dynamic head is 25 ft. and the water horsepower 16. If it is desired to increase the quantity of circulating water to 4000 gal. per min. (with exactly the same condenser), the friction head becomes 43 ft., owing to the increased velocity of the water, the dynamic head 53, and the power 52.5, or over three times the power at rated load. Whether this expenditure of power would be profitable depends upon the design of the condenser, the influence of the increased velocity of the water on its efficiency and capacity and also on the type of prime mover, cost of coal and temperature of the water.

The characteristic of a typical Wheeler circulating pump driven at rated speed is shown in Fig. 3. The sharpness with which the head falls off above rated capacity insures a drooping power curve and hence protects the driving machine from injurious overloads. Regulation of the discharge is obtained by adjusting a valve in the suction of the pump, where the velocity is low, and hence the head lost is slight.

This way of adjusting is usually adopted because provisions for speed regulation on the part of driving motors and turbines add to their complication, and require more expert attention from the operating force. With steam-turbine drive the exhaust is used in the feed heater, so that the loss, which in any case is small, is practically all regained and returned to the boiler as heat in the feed water. With engine or direct-current motor drive, the discharge is regulated by speed adjustment. The characteristics of a centrifugal circulating pump at different speeds is shown by Fig. 2.

Hotwell Pumps

Hotwell pumps must deliver water against the head corresponding to the vacuum, plus the friction head and the static head. The pump cannot create a suction sufficiently greater than the vacuum in the condenser to draw water into the impeller by its own action. Thus a static or gravity head is required to produce the entrance velocity and overcome the frictional and entry resistances to the flow of water from the bottom of the condenser into the impeller. Once the water is in the impeller it can be discharged against the pressure of the atmosphere and any static head imposed upon it, but it is essential that proper

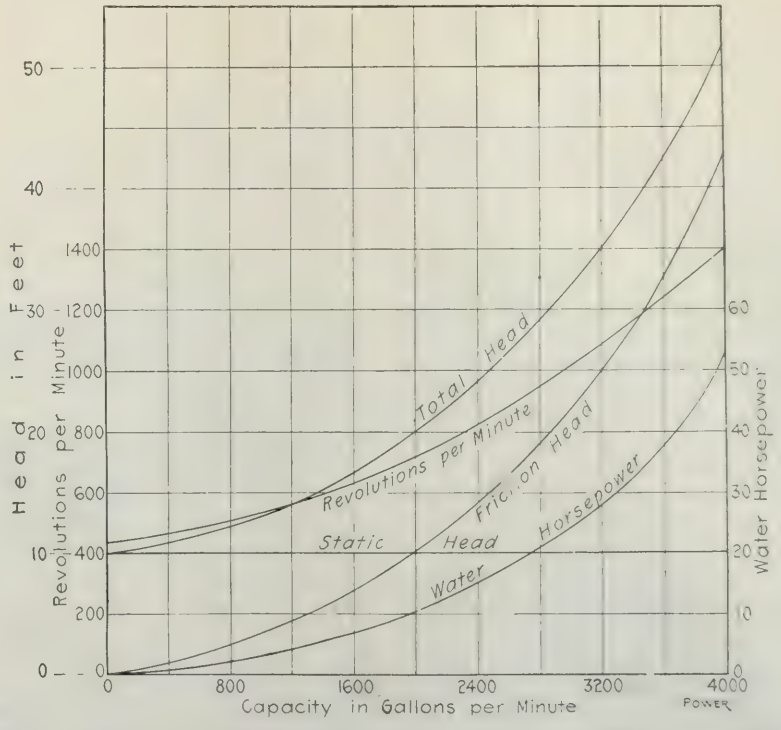


Fig. 1. Effect of Increasing Velocity of Circulating Water on Power Consumption of Pump

provision be made for getting the water into the pump. With hotwell pumps and also with tail pumps, reliable service is insured by providing a suitable gravity head.

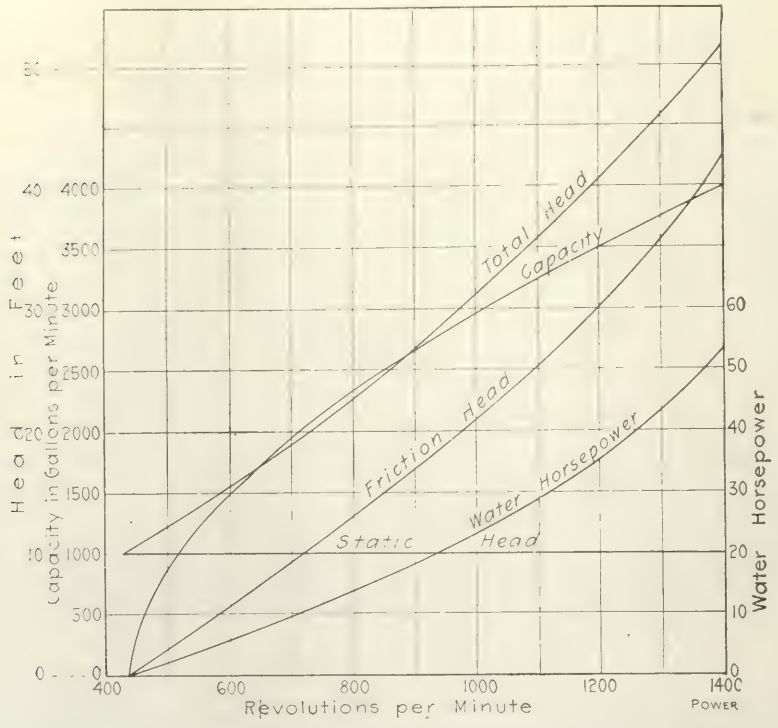


FIG. 2. RELATIONS BETWEEN R.P.M., CAPACITY, HEAD AND WATER HORSEPOWER

Hotwell pumps are built in single-stage and two-stage types. Typical designs are shown in Figs. 7 and 8. Two-stage pumps are sometimes used where the available gravity head is close to the minimum. Under these conditions the first-stage impeller helps to overcome entry losses and to increase the reliability of operation.

Fig. 4 is a head-capacity curve of a two-stage turbo hotwell pump of a maxi-

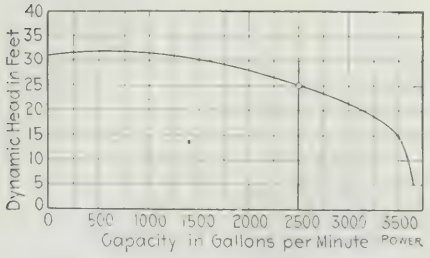


FIG. 3. HEAD CAPACITY CHARACTERISTIC OF CIRCULATING PUMP

mum capacity of 250,000 lb. of steam per hour. Hotwell pumps are rated at maximum capacity. A moment's consideration will show why this is the best method of rating. On heavy overloads the maximum output of the complete condensing equipment is not expected to be sufficient to maintain the rated vacuum. The vacuum is expected to fall off. The hotwell pump, however, is called upon to deliver the maximum quantity of water. Accordingly, a specific guarantee of the maximum capacity of the hotwell pump, and not the normal rating as with pumps for other services, should be demanded.

TAIL PUMPS FOR JET CONDENSER SERVICE

Fig. 5 gives the characteristic curve of a high-speed centrifugal tail pump for removing the condensed steam and cool-

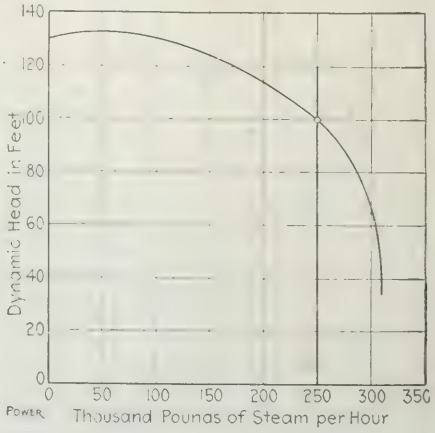


FIG. 4. HEAD CAPACITY CHARACTERISTIC OF HOTWELL PUMP

ing water from a jet condenser. This type of pump is used for rectangular counter-current jet condensers and under type jet condensers, taking the place of a barometric column. The discussion regarding the necessity of sufficient inlet gravity head for hotwell pumps applies with equal force to tail pumps. Reliable service can only be insured by providing ample gravity head.

The cooling water used by the jet condenser is controlled by throttling on the injection line to the condenser. The tail pump is usually run at constant speed, as is a hotwell pump, and delivers the combined cooling water and condensate against the pressure of the atmosphere and the static head.

For cooling tower work tail pumps have characteristics similar to that in Fig. 5, except that the head generated is higher throughout. Besides the head corresponding to the vacuum there is an additional static head depending on the type and design of the tower and its elevation.

Where a barometric jet condenser is used, the tail water is removed by the barometric column, but a pump must be employed to overcome the frictional head and the unbalanced static head against which the water supplied to the barometric cone must be pumped. Pumps for this service have a characteristic similar to that of Fig. 5, designed to deliver, when starting, a moderate quantity of water against full head, corresponding to the

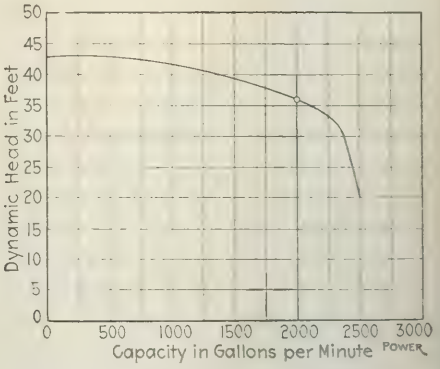


FIG. 5. HEAD CAPACITY CHARACTERISTIC OF TAIL PUMP

elevation of the barometric cone, and on full condenser load, greater capacities against a somewhat smaller head, since the vacuum in the condenser helps to lift the water.

PRIMING AND SEALING

Centrifugal circulating pumps for condenser service may be primed as are pumps for other services, that is, by a steam ejector creating a partial vacuum in the pump, and drawing up the water through the suction pipe, or, if a foot valve is used, by supplying water to the pump, either from a convenient water main or by an ejector. A circulating pump may be primed also by using the condenser and air pump to furnish the vacuum required to lift the water up into the pump. In surface condensers this system of priming necessitates a small bypass pipe from the water box into the condensing chamber proper, and after a vacuum is created in the condenser by the air pump, the valve in this small bypass is opened, thus transmitting the vacuum to the circulating water circuit, and to the centrifugal pump, so that

priming water is drawn up through the suction pipe into the impeller.

If the stuffing-boxes are in bad condition and air leaks into the pump, priming into the impeller is likely to cause the pump to lose its water and to give trouble, due to air pockets.

Furthermore, with surface condensers,

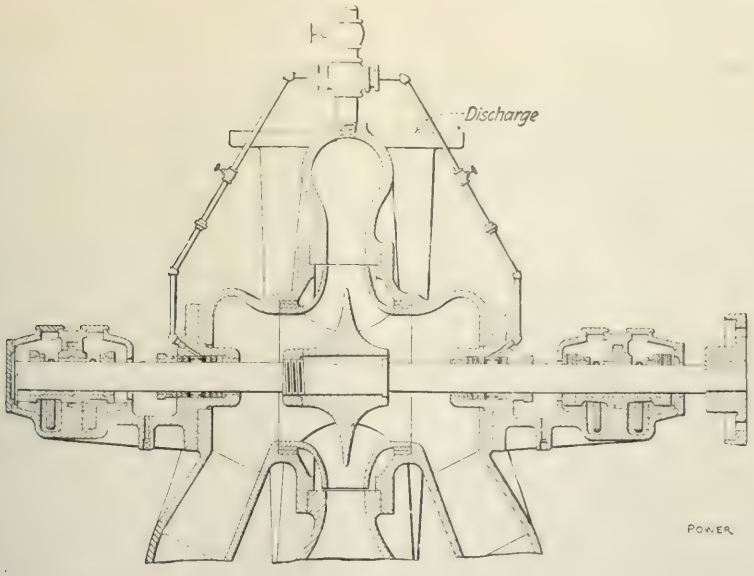


FIG. 6. SECTION OF CIRCULATING PUMP, SHOWING BEARINGS AND METHOD OF SEALING

ing by this means or by the ejector creating a partial vacuum, will be very difficult. Sealing against the inleakage of

the system is often operated as a siphon, and if air leaks into the pump, it will collect in a pocket at the top of the siphon

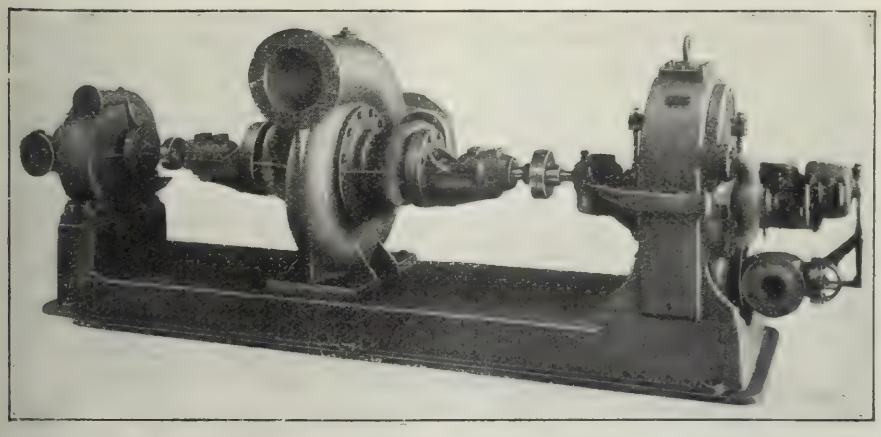


FIG. 7. CIRCULATING AND HOTWELL PUMPS ON COMMON BASE

air along the shaft of a centrifugal pump is also necessary because, if the pump is operating with a suction lift, air leak-

and eventually result in the pump losing its water. Where a centrifugal tail pump is used with a jet condenser, the stuffing-

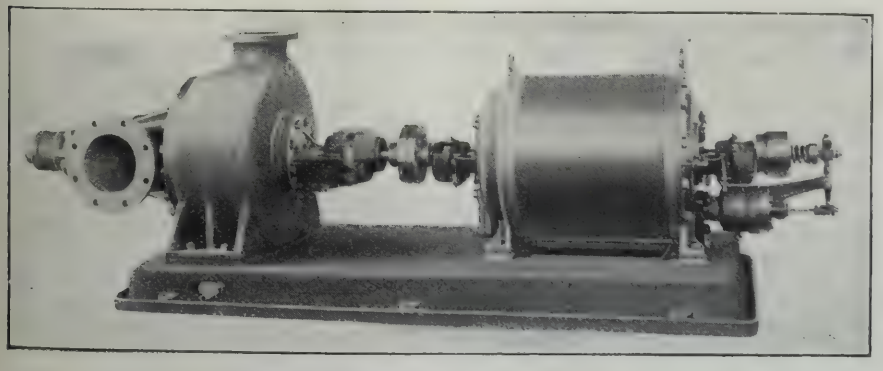


FIG. 8. TWO-STAGE HOTWELL PUMP

boxes must be tight, or the air leaking in will form a pocket in the pump and prevent the tail water getting into the impeller, which it must do by gravity. This also applies to hotwell pumps in connection with surface condensers.

The design of stuffing-boxes used with Wheeler pumps is shown in Fig. 6. The stuffing-boxes are extra deep and take a large number of turns of packing, thus necessitating less pressure and eliminating binding of the shaft. Midway of the stuffing-box, that is, between the two packings, is a spacer or lantern, consisting of a bronze ring which receives the sealing water by the piping shown. No air can be drawn into the pump through the inner packing, since any leakage will only result in the introduction of sealing water continually supplied to the lantern, and the outer packing serves to prevent the leakage of this sealing water out along the shaft.

The Brotherhood of Power Workers

The Brotherhood of Power Workers is a commendable growing organization in Massachusetts. The following facts concerning it are from John A. Levy, of Greenfield, Mass:

The Brotherhood was organized in September, 1910, and those engaged in the generation and transmission of power regardless of race or color are eligible to membership. Its purpose is primarily to protect the plant worker by educating him to greater competency in his field. It does not interfere in labor disputes, except when a member asks for assistance in adjusting trouble. A committee is appointed for this purpose. It provides that no member shall fill a place left vacant by any organized worker through labor disputes.

The Brotherhood takes the stand that a cheap man in the plant is far worse than cheap material and supplies. Through its employment bureau, it aims to offer high-grade service, combined with coöperation with employers to obliterate friction between employer and employee. An examining board questions and coaches an applicant desirous of obtaining a license. A sick and accident benefit of $5 per week is allowed, and a death benefit of $1 from every member, granted to beneficiaries of a deceased member. It is proposed to establish an insurance department to assist in maintaining and providing the death and sick benefits. The initiation fee to the Brotherhood is $3, one of which goes into the death-benefit fund. The dues are 50c. a month. Additional information may be obtained from the corresponding secretary, C. C. Harris, 89 Hall St., Springfield, Mass.

We extend our best wishes to the Brotherhood for a long and prosperous life and the realization of its aims.

Some Common Thermometer Errors

By J. C. Peebles

Mercury thermometers are subject to errors which often are overlooked. This article indicates where these are likely to be found and how they may be avoided or corrected.

While the ordinary glass thermometer filled with mercury is the instrument in most general use for the measurement of temperature, the experience of the writer in the use of many of them has been that they are oftentimes subject to very serious errors which the average user fails to detect or to take into consideration. The very simplicity of the instrument makes it easy for these errors to pass unnoticed, and it is the purpose here to point out to the average user just where these errors come in, and to indicate how they may be avoided or reduced in magnitude.

Error Due to Distillation of Mercury

When the ordinary thermometer is used above a temperature of 200 deg. F., a more or less rapid distillation of the mercury takes place. If the upper part of the stem is cool, this mercury vapor condenses in the upper part of the capillary tube and remains there. After a prolonged period of heating at a temperature of 200 to 300 deg. F., it is no uncommon experience to find a thread of mercury several degrees in length in the upper part of the capillary tube. Obviously this introduces an error into the indication of the instrument. For example, if a thread of mercury 5 deg. in length has collected in the upper part of the tube, the thermometer indications will be 5 deg. low.

Up to about 400 deg. this distillation and subsequent condensation of the mercury can usually be avoided by keeping the top of the mercury column cool or the whole stem warm. The former prevents distillation from the surface of the mercury and the latter the condensation of the mercury vapor in the top of the tube. It is believed that most users will find it easier to keep the top of the mercury column cool by allowing the stem to project far enough out of the hot body so that the top of the mercury is at a comparatively low temperature, say 200 deg. or less. If the nature of the work makes this impossible, the whole thermometer must be kept warm.

Above 400 deg., these precautions will be found inadequate. In the manufacture of the better grade of thermometers intended for high-temperature work, the capillary tube above the mercury is filled with an inert gas, usually nitrogen. As the mercury rises in the tube the nitrogen is compressed into smaller volume and the increase in pressure prevents the boiling of the mercury. When purchasing a thermometer to be used above 400 deg., where reliable results are essential, a nitrogen-filled instrument should be specified.

Error Due to Shrinkage of Glass

After a thermometer has been made, filled with mercury, and the scale etched on, it may undergo changes which very seriously affect its accuracy. These changes are caused by a gradual adjustment of the internal stresses in the glass which were produced when the stem was made. As these stresses are relieved by the aging of the glass, contraction of both stem and bulb takes place, thus rendering all the readings high. This slow contraction may continue for months, and even for years, before all the glass stresses are relieved. Thus, if it has been made from "green" or unseasoned glass it will show an increasing error as time passes.

The only remedy for such a condition is to season or age the thermometer before the scale is put on by a process of annealing. This is done by subjecting it for a period of four to seven days to a temperature somewhat above the highest at which it is to be used, followed by slow cooling for two to three days. The artificial aging removes all the internal stresses in the glass and leaves it in a condition of equilibrium where no further contraction of stem or bulb can occur.

Experiments conducted by the writer on unannealed instruments showed that an error of 40 deg. in 600 deg. F. was not unusual. Errors of 60 deg. in 750 deg. were found in several cases, while errors of 25 deg. in the range from 400 to 700 deg. were common. As many thermometers in the market have not been properly annealed, it must be kept in mind that their indications, particularly in the foregoing range, are capable of very serious error and should at all times be regarded with suspicion. When reliable results are required, only a well annealed instrument, from a reliable maker, should be used.

Error Due to Partial Immersion

When a thermometer has been properly annealed in the making, and has been filled with nitrogen to prevent boiling the mercury, its indications may still be subject to errors depending upon the conditions under which it is used. Thermometers are calibrated under a condition of "total immersion"; that is, the entire instrument is brought to the desired temperature when the fixed points of the scale are located. Therefore, the indications are correct only under a condition of total immersion. But in its ordinary use a condition of total immersion is very seldom realized.

The usual condition is to have the bulb and a short length of the stem immersed in the hot medium while the remainder projects into the air at a much lower temperature. A considerable length of mercury in the capillary tube is thus at a much lower temperature than that in the bulb; this causes the thermometer to read lower than if the entire instrument were at the same temperature. This "stem correction," as it is called, will depend upon the difference in temperature between the immersed and emergent parts, the length of the emergent part and the glass of which the thermometer is made. In many cases the magnitude of this stem correction is such as to interest every user, the practical as well as the purely scientific man.

The determination of the flash or burning point of an oil is a test which any operating engineer may wish to perform, and the experience of the writer may be of interest in this connection. Some heavy cylinder oil was placed in a copper cup and heated electrically. The thermometer used had a scale running from 150 to 800 deg. F., and was immersed to the 200 deg. point in the oil. The burning point was ascertained to be 625 deg., as shown by the thermometer. A calibrated platinum resistance thermometer, which is compensated for depth of immersion, read 652 deg. F. when placed side by side with the glass instrument, which was therefore 27 deg. low at 652 deg., the latter being the correct burning point of the oil. As a further precaution to insure reliable results, the two thermometers were placed side by side in an oven, which gave a condition of total immersion for the glass one. At 650 deg. their readings were substantially the same.

Formula for Correcting Error Due to Partial Immersion

This stem correction may be determined from the equation

$$S = K n (T_1 - T_2)$$

where

S = Stem correction in degrees;
n = Number of degrees that mercury emerges from hot body
T_1 = Temperature of hot body;
T_2 = Average temperature of emergent stem;

K = A constant depending on the glass of the thermometer.

In the above experiment

$S = 27$ deg.;

$n = (625 - 200) = 425$ deg.;

$T_1 = 625$ deg.;

$T_2 = 150$ deg.

whence

$$K = \frac{27}{425\ (625 - 150)} = \frac{27}{425 \times 475} = 0.0001337$$

For this thermometer, then, the equation

$$S = 0.0001337 \times n\ (T_1 - T_2)$$

will give the stem correction for any condition of partial immersion.

Summary

When working in the range from 300 to 700 deg. F. the average thermometer user should keep in mind the following points: (1) Do not allow the mercury to distill and condense later in the top of the tube; better still, use a nitrogen-filled thermometer. (2) For careful work use only one which is known to have been well annealed in the making. (3) Ascertain the stem correction, for it may frequently introduce an error as large as 5 per cent. If a standard thermometer is not available for this determination, any reliable testing bureau will do the work for a very nominal charge.

The 1912 U. S. Standard

The *Journal* of the American Society of Mechanical Engineers comments upon the flange situation as follows:

The various makes of flanges and fittings now on the market, figured on the basis of the bolt stress, show a factor of safety in many cases as low as two and in a few cases even less. This compared with the necessary factor of safety of the pipe itself which is approximately sixteen, shows at a glance the necessity for the changes made in the 1912 U. S. Standard, of bringing up the factor of safety in the bolts, flanges and fittings, especially in view of the enormous strains encountered due to changes of temperature.

The other important feature in the 1912 U. S. Standard is the uniformity of dimensions providing for interchangeability of fittings. Until very recently, each manufacturer had his own standard and chaos has resulted, so that the engineer must know what particular make is to be purchased before he can proceed to design or erect the piping work. Hereafter all fittings of all manufacturers who comply with the new schedule will be uniform in the principal dimensions, and this will be an important factor in obtaining quickly spare parts throughout the country and enable one readily to make repairs and alterations. The new standard does not attempt to fix the quality of the metal or the thickness of the shell of fittings, leaving this to the manufacturers to regulate in accordance with their guarantee, depending on the pressure the fittings are to carry.

Standard Weight Flanges

In comparing the 1912 U. S. Schedule of Standard Weight Flanges with the alternative standard submitted by the subcommittee of manufacturers with respect to diameter of bolt circle, number of bolts and diameter of bolt-holes, we find they are identical, with the one exception that the U. S. Standard gives ⅞ in. instead of ¾ in. for diameter of bolt-hole for 4-in. pipe.

The standard of the Engineering Standards Committee of Great Britain and the standard of the Verein deutscher Ingenieure do not have sizes exactly corresponding to the standard weight and extra-heavy weight of the 1912 U. S. Standard, but have the most important dimensions, the same for all pressures, making their piping on low pressures unnecessarily heavy and expensive, but without securing absolute safety for high pressures. Comparing the proposed Manufacturers' Standard with British standard pipe flanges for working steam pressures of 225 lb. per sq.in., we find the British standards give higher values, especially as far as number of bolts is concerned.

Extra-heavy Flanges

For extra-heavy flanges up to 9 in. inclusive, the 1912 U. S. Standard and the standard of most manufacturers are identical. Above 9 in. the U. S. Standard is somewhat larger than the present standards, as has been stated. With the British standard, the diameter of bolt circle and number of bolts are the same for pipes corresponding to the 1912 U. S. Standard or proposed Manufacturers' Standard weight and extra-heavy weight, but from 10 in. on the diameter of the bolt circle is from ¼ to ½ in. less than the proposed Manufacturers' Standard and ¾ in. less than the U. S. Standard. The number of bolts in most cases for pipes from 10 in. on in the U. S. 1912 and Manufacturers' is from two to four bolts greater than in the British standard. As far as size of bolts is concerned, the British standard makes a distinction between pipes for pressures up to 225 lb. and up to 325 lb. The bolts for pipes up to 225 lb. are below the Manufacturers' Standard ⅛ in. in nearly every case and below the U. S. Standard from ⅛ to ⅜ in. In pipes for pressure of 325 lb. the British standard is the same as the proposed Manufacturers' schedule in all cases except 14 in. and 18 in. where the British standard is ⅛ in. larger than the Manufacturers' standard, but from ⅛ to ¼ in. smaller than the U. S. standard.

It is thus seen that wherever there is a distinction between the 1912 U. S. and proposed Manufacturers' Standard, the 1912 U. S. Standard is invariably on the side of greater safety and strength and has the added feature of interchangeability where the proposed Manufacturers' schedule has different face to face dimensions. As to the comparison with the British and German standards it must be remembered that they cover by one standard both weights of the American specifications, and as would be naturally expected with compromises must have higher values for standard weight pipes and lower values for extra-heavy weight pipes. However, in Germany there is reported in the public press dissatisfaction with the existing standards as not being safe. The Alsace Association of Owners of Steam Machinery prints in its annual report for 1911 as follows:

"It is nearly impossible to estimate with any degree of precision the bending stresses to which pipe flanges are submitted when, to obtain a good joint, the bolts are screwed tightly, especially if the flange has to withstand at the same time stresses due to expansion. As a result, many of the joints in pipes and similar apparatus are working with very low factors of safety, as has been shown lately by the fact that accidents from piping have been far more numerous than from boilers, and with as fatal consequences."

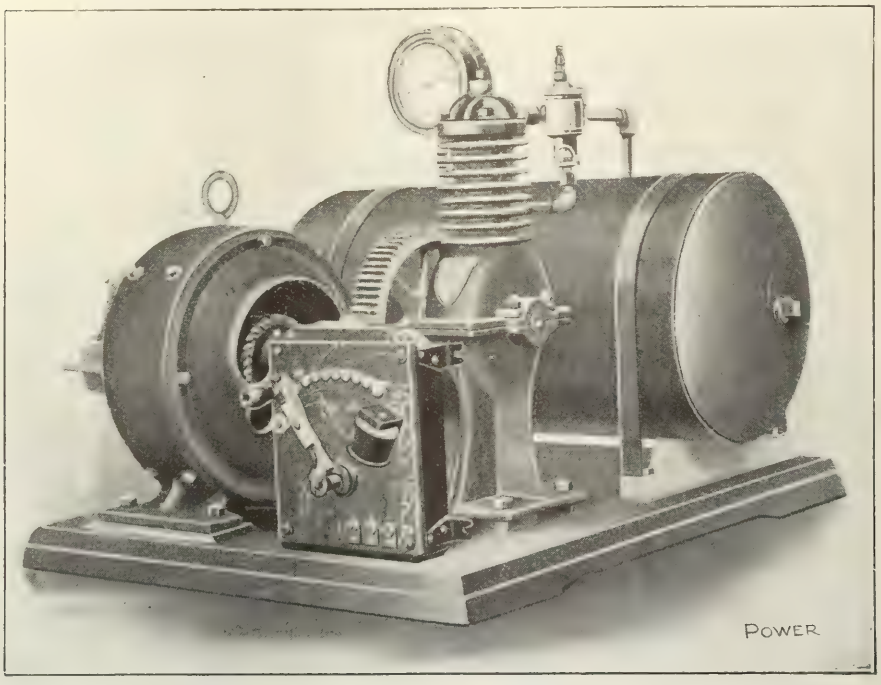
The 1912 U. S. Standard is the result of the efforts of the aforementioned committees covering a period of over a year during which time every detail was considered. The committee had the active coöperation of engineers representing large power plants, manufacturers and the U. S. Navy Department. Prominent engineers and architects throughout the country were communicated with and their opinions and advice received and considered, and the consummation is the 1912 U. S. Standard.

The first impulse turbines had blading in which the inlet and discharge angles were equal. Now almost all builders use blading of which the discharge side of the blade makes a sharper angle with the axis than the inlet side. This does not necessarily mean that the discharge area of the blades is smaller than the inlet area, for the blade is usually lengthened radially on the discharge side. Thus both inlet and discharge areas are made equal. The sharp angle of discharge reduces slightly the relative velocity of exhaust from the moving blade. In turbines of the Curtis type there is usually only a small difference between the inlet and outlet angles of the first row of moving blades, but on the second row of moving blades in the stage the entrance and exit angles often differ by as much as 15 deg.—*The Engineer.*

The 12-gun battleship "Wyoming" has a cruising speed of over 22 knots an hour.

Goulds Air Compressor Outfit

The Goulds Manufacturing Co., Seneca Falls, N. Y., has placed on the market a complete air-compressor outfit, consisting of an air-compressor pump mounted on an iron base with the driving motor, starter, and air-storage tank. The motor has an output of 1 hp. and the pump has a capacity of 1.27 cu.ft. of free air per minute when operated at 150 r.p.m. The outfit is compact and self-contained. With a tank 28x14 in. in diameter, with a capacity of 3¼ cu.ft. of air at 150 lb. pressure, the overall dimensions of the base are 30x30 in. Tanks of larger or smaller capacity are furnished as desired.



GOULDS MOTOR-DRIVEN AIR-COMPRESSING OUTFIT

Stop Valves in Boiler Piping

The location of stop valves frequently turns an otherwise excellent piece of boiler piping into a rank failure, and often produces a menace to life and limb. Fig. 1 shows the usual construction adopted for connecting a boiler to a steam header. Such a method is excellent with the exception of the location of the stop valve, which is the one most frequently chosen, apparently because it can be most easily reached from the top of the boiler, when so located.

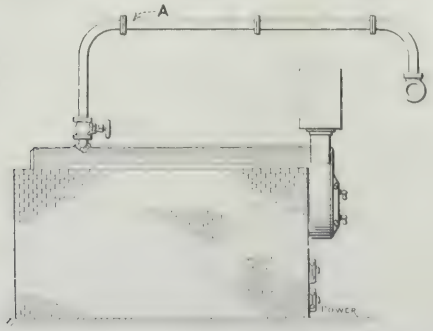


FIG. 1. BOILER AND HEADER CONNECTION

With the valve placed as shown, a large quantity of water can collect above it when it is closed; this is a source of great danger when cutting a boiler in on the line. Of course, a bleeder can be connected above the valve, but sooner or later someone in a hurry, or forgetful, will open the main valve without draining the pipe, with probably disastrous results. If the pipe pitches from the main header to the boiler, water will lie along the horizontal section when the valve is closed, thus producing trouble at the joints by leakage.

The correct location of a valve on this type of connection is at *A*, with the pipe pitching from *A* toward the steam main.

To have a stop valve accessible is a very important consideration. This should be accomplished by providing chains so that the valve may be handled from the boiler-room floor. A stop valve should always be located where it will prevent the accumulation of water, accessibility being a secondary consideration.

In Fig. 2 is shown an arrangement of piping and valves that appears excellent, but it developed a serious objection when put under steam pressure. The connections from the main header to the individual boilers, which were about 40 ft. long, were made of 6-in. pipes, using long-radius bends where connections were made to the boilers. With the valves located as shown, there was no chance for water pockets as the pipes drained back to the boilers. However, when one of the boilers was idle and its stop valve was closed, the connecting pipe would cool off to the temperature of the atmosphere, which at 60 deg. F. made it about $\frac{15}{16}$ in. shorter than the other two at a temperature corresponding to the running pressure of 125 lb.

Owing to the rigidity of the 12-in. header, this change in pipe length was practically all accommodated by the giving of the pipe at the end where it was attached to the boiler. Joint trouble resulted almost immediately, and after about three years' use the pipes began breaking at the flanges where connected to the boilers due to the excessive strain and the vibration of the pipe.

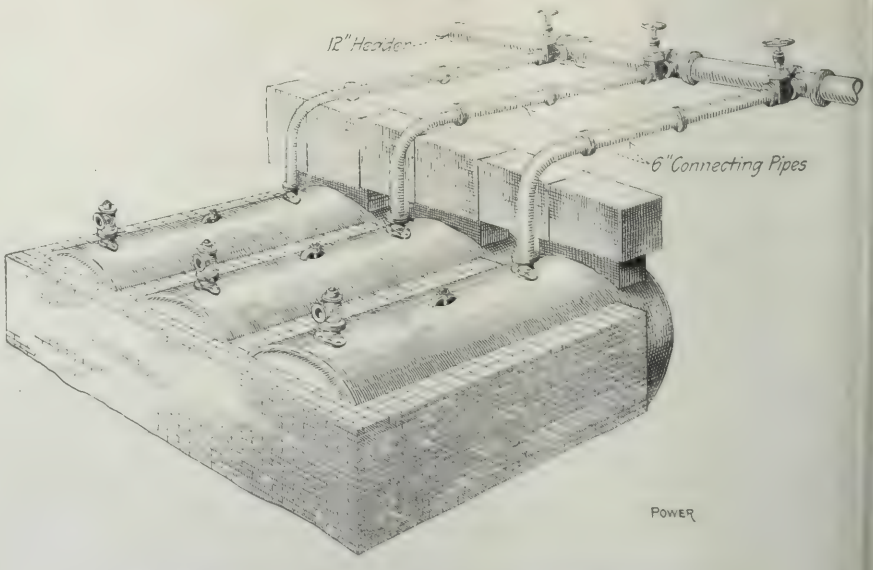


FIG. 2. LOCATION OF STOP VALVE IN HEADER CONNECTION

The proper location for the stop valve was, of course, near the connection to the boilers; then all of the piping beyond them would have always remained at a uniform temperature.

Threshing Machine Boiler Explosion

By E. H. Kirchner

At 8:30 a.m., Aug. 28, a 20-hp. threshing-engine boiler exploded, killing the engineer in charge; a 16-year old boy was badly scalded, but there are hopes of his recovery.

The outfit was in service a short time the day before. In the morning the boiler was fired up by the owner until 120 lb. of steam was registered with two gages of water showing. The engineer was on the foot-board just starting the engine when the explosion took place. The boy was blown about 200 ft. The firebox and rear truck were thrown back about 15 ft., pinning the engineer under the heavy mass, and killing him instantly. Figs. 1 and 2 show the result of the explosion.

The front truck was thrown about 10 ft. forward and 5 ft. to the right. The boiler, as indications show, ruptured at the seam where it was riveted to the firebox.

The sheet was nearly straightened out. It was patched about two years ago and was bagged. This piece practically contained the whole boiler sheet which sheared off 2 ft. from the front end and at the back where it was riveted to the firebox. It weighed about 450 lb. and was blown into the air about 100 ft., dropping through the roof of a barn.

The boiler tubes were bent and twisted and driven into the ground. The shell was not crystallized and did not show low-water indications. The shell was ¼ in. and the head ⅜ in. thick, but in places the shell was pitted badly, leaving only about ⅛ in. thickness.

The flues looked good and if there was any scale on them it was blown off when the explosion occurred. The rivets in the bottom of the girth seam pulled through the rivet holes nearly half around the boiler; the balance of the shell was torn and ripped open lengthwise. The boiler was about 12 years old, but had been rebuilt.

Test of Line Shaft Bearings

By W. C. DuComb, Jr.

The following results were obtained from comparative friction tests of line-shaft bearings recently made under the direction of S. A. Ellicson, president of the Chicago Pulley & Shafting Co. The object of the tests was to determine just how much less friction is found in correctly designed roller bearings operating under normal conditions of line-shaft service than in ring-oiling babbitt bearings operated under similar conditions.

A testing device specially designed for the purpose was set up with means provided whereby a known load could be applied to the bearings under test, and the friction load accurately measured. The friction was electrically measured by the testing device.

The tests with stock bearings were made as follows: The motor was first run light to find the power required to turn it over, the ammeter indicating 0.775 amp. The belt was then placed on the motor pulley and on the pulley on the shaft supported by two ring-oiled babbitt bearings. A load equivalent to 1200 lb. was applied by the lever and weight, thus imposing a load of 600 lb. on each shaft or 300 lb. on each bearing tested.

The motor was started by moving the lever on the starting box over slowly, and after the first high reading, due to starting the shaft, the amperage gradually dropped until 1.4 amp. was indicated. The difference between 1.4 and 0.775 amp. or 0.625 amp. required to run the motor is taken as the power loss in the bearings.

The belt was then moved to the shaft pulley supported by two roller bearings and the same load applied. The pointer settled at 1.15 amp. The difference between 1.15 amp. and 0.775 is 0.375 amp., which is taken as the power loss in the two roller bearings. This makes a difference of 0.25 amp. in favor of the roller bearings or ½ hp.

A high-grade machine oil was used for the ring bearings, and a nonfluid oil was used on the roller bearings. Oil was also tried in the roller bearings and practically the same results obtained. The only difference observed being that slightly less effort was required to start the shaft when oil was used.

Swain Metallic Packing

In the line of metallic packing, manufactured by the Swain Lubricator Co., 328 West Lake St., Chicago, Ill., is an improved piston-rod packing for heavy service. The wearing rings are made with angles of 30 and 60 deg. to make efficient contact joints. The rings come in two sections each and are faced with bronze; one section overlaps the other and each is doweled to keep it in position and form a tight joint.

Another design is called the double or compound packing. The rings retain any condensation that may escape past the primary wearing rings and thus keep the external engine parts dry. The wearing rings are made from a special babbitt encased in bronze, and suited to the conditions under which they are to be used.

There is also a rotary valve-stem packing for use with Corliss engines. The outer parts rotate with the valve stem, the advantage being the elimination of wear on the stems. The operating joints come next on the valve stem. These joints are lubricated with oil supplied through a feed pipe. The tension of a spring is just sufficient to prevent leakage past the valve stem. The packing is used with superheated steam.

Figs. 1 and 2. Remains of the Exploded Boiler

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

The Single Phase Commutator Motor

By John A. Randolph

The extensive application following the recent development of the single-phase commutator motor, has led many engineers to study the design and operation of this type. As a result, many modifications have been produced, each with its own particular merits. However, after the test of service, many of these have been compelled, on account of commutation difficulties and low efficiency, to give way to a few styles which have proved practical.

The investigation in detail of alternating-current phenomena is more or less complex in that it requires the extensive use of mathematical equations and vector diagrams together with exhaustive experimentation; hence the task of those who have brought forth successful single-phase series motors has been by no means light. To enter a detailed mathematical discussion of the various successful types is not within the scope of this article, but rather it is the purpose to set forth the principles involved and to give a general explanation of the methods of design most commonly employed.

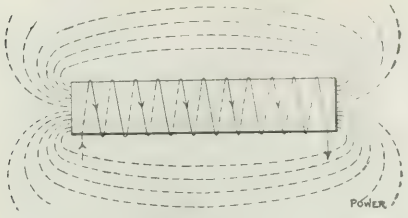


Fig. 1

Fig. 2

It is well known that if both the armature and field connections of a direct-current motor are reversed, the direction of rotation is not changed. It therefore follows that the same result will ensue when an alternating current is applied to these terminals. However, while a direct-current motor can be made to turn on an alternating current, many difficulties are encountered which render it wholly unsuited for such work unless especially designed for operation on an alternating-current circuit.

One of the chief difficulties is the heavy loss caused by the eddy currents produced in the field cores by the alternating flux, a loss which must be considered very carefully in the design of all alternating-current apparatus employing iron cores in the magnetic circuit. The means employed for its reduction consists in laminating the core, the field frames and poles being laminated in the alternating current motor instead of solid iron, as in the direct-current motor.

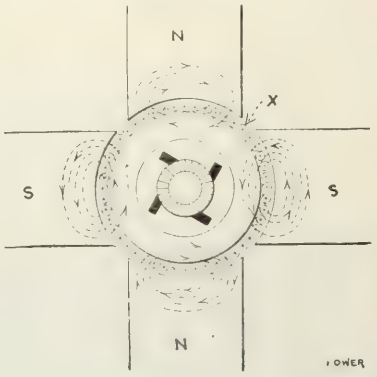


Fig. 3

Another obstacle is the short-circuit current flowing in the coils undergoing commutation, an action analogous to short-circuiting the secondary of a transformer. In the motor, the field winding acts as the transformer primary and the armature coil, undergoing commutation, as the short-circuited secondary. With a strong flux and a high frequency, this current will be comparatively large, causing a local I^2R or heat loss of considerable magnitude and heavy sparking as the coil passes from under the short-circuiting brush. To obviate this, several means are employed. One consists in greatly reducing the flux by increasing the number of ampere-turns on the armature and decreasing those on the field. The density is further reduced by spreading the flux more broadly over the armature, the number of field poles being increased to accomplish this purpose. This accounts in part for the greater number of field poles in the alternating-current motor and also for the larger armature than is customary for direct-current motors of the same output. This feature, combined with the laminated construction of the field frame, makes the weight of the alternating-current motor, in general, from 30 to 50 per cent. greater.

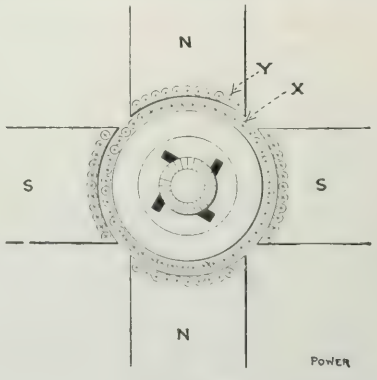


Fig. 4

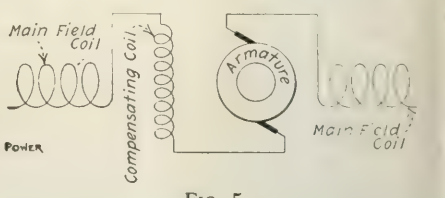


Fig. 5

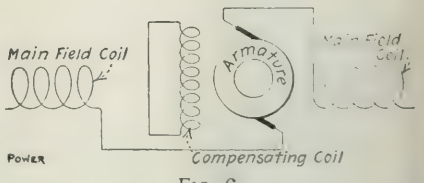


Fig. 6

A feature always to be considered in connection with alternating-current work is the electromotive force of self-induction which exists in a coil through which the current is rapidly alternating. In the motor, this force causes the current to lag behind the impressed voltage thus lowering the power factor and consequently the efficiency of the motor. It is chiefly for this reason that the shunt motor is not adapted for alternating-current work, owing to the great number of its field ampere-turns and the consequent high inductance. In the series motor, the current in the field is always in phase with that in the armature and the inductance of its field is much less than that

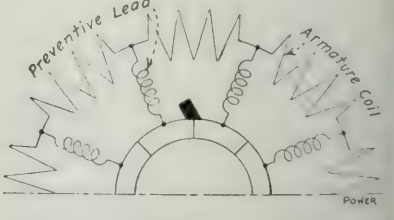


Fig. 7

of the shunt type; therefore this is generally used for a single-phase commutator type. Nevertheless, in the series motor, considerable self-induction still exists which must be overcome. Inasmuch as this force opposes the flow of current set up by the impressed voltage, part of the latter must be expended in

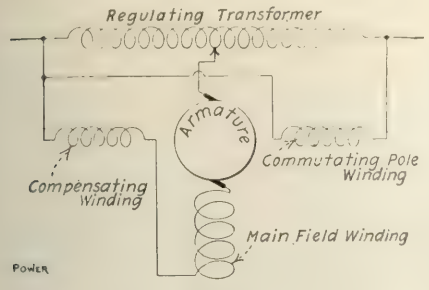


FIG. 8

overcoming it; hence the applied voltage must be greater than that for a similar direct-current machine. Another difficulty which may occur as a result of the self-induction is the accidental short-circuiting of one or more field coils. When this occurs, the coils at once act as the short-circuited secondary of a transformer, thus causing a heavy flow of current which is liable to burn out the insulation of the affected coils unless detected in time. Great care is therefore necessary in the design of the coils.

The electromotive force of self-induction is directly proportional to the frequency of the supply voltage, to the field

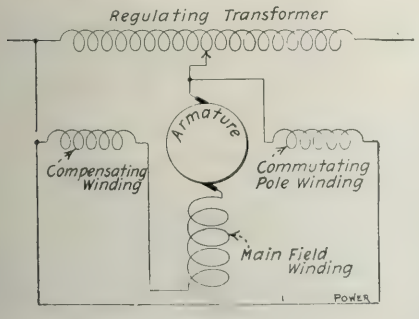


FIG. 9

flux and to the square of the number of its turns; therefore, in the design of an alternating-current series motor, these points should all be considered in order that the power factor may be as high as possible. The frequency can be made as low as commercial practice will permit, but it cannot well be lower without special generating apparatus. By increasing the number of armature coils and conductors and decreasing the field strength, the inductance of the field is greatly reduced, but the cross-field of the armature is increased.

The cross-field should be carefully considered in the design of all commutating machines whether built for direct or alternating current. Its tendency to distort the main field flux and thus to greatly increase the commutation difficulties is well known. In alternating-current series motors it is counteracted by compensating coils, in explanation of which consider a simple coil of wire wound upon an iron core, as in Fig. 1. If a current is passed through this coil, a magnetic flux will be induced as indicated, its intensity being greatly increased by the presence of the iron core. Now let a second coil containing the same number of turns, be run parallel to the first coil, as in Fig. 2.

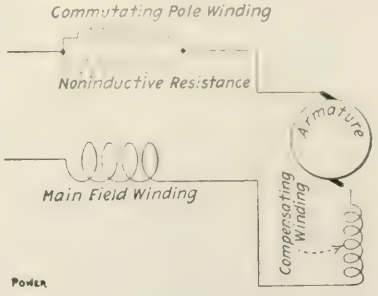


FIG. 10

If a current is passed through the second coil of the same strength as that in the first coil, but in the opposite direction, the magnetic induction of the first coil will be counteracted so that practically no flux will flow through the iron core.

Now modify Fig. 1 as in Fig. 3 and

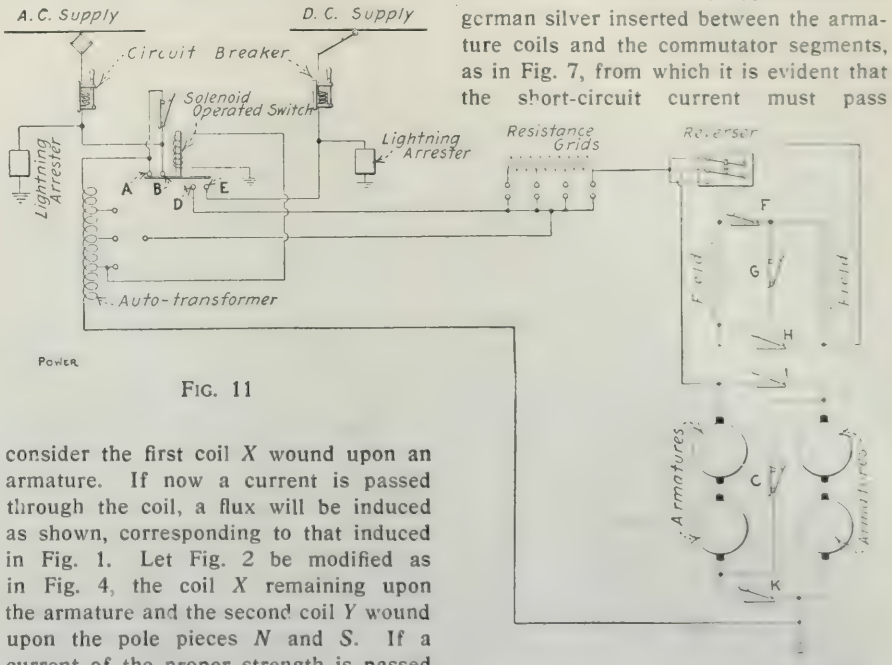


FIG. 11

consider the first coil X wound upon an armature. If now a current is passed through the coil, a flux will be induced as shown, corresponding to that induced in Fig. 1. Let Fig. 2 be modified as in Fig. 4, the coil X remaining upon the armature and the second coil Y wound upon the pole pieces N and S. If a current of the proper strength is passed through coil Y in a direction opposite to that in coil X, the magnetomotive force setting up the flux shown in Fig. 3 is neutralized, and the flux disappears. This flux thus compensated is the element which would cause field distortion if allowed to exist.

This principle is utilized in designing a single-phase commutator motor. The armature is surrounded by a fixed winding carried in slots cut in the pole pieces, this winding being so designed that its current will be equal in magnetomotive force and opposite in phase to the current in the armature, thus neutralizing the distorting effect of the cross-field set up by the latter. A common method of supplying current to the compensating winding consists in connecting the coil in series with both field and armature as in Fig. 5. This is known as the forced compensation or conductive method.

Another method consists in short-circuiting the coil upon itself, as in Fig. 6. In this the current is induced in the compensating winding, acting as the short-circuited secondary of a transformer of which the armature is the primary.

A serious obstacle encountered in alternating-current commutator motors is the current generated in the coils undergoing commutation. When the coils are short-circuited at the commutating point by the brushes bridging over the commutator segments, a large current will flow through the coil, owing to the action of the alternating flux. Unless effective means are employed to reduce this current, it will seriously decrease the efficiency of the motor and shorten its life, owing to its heating effect and its tendency to produce heavy sparking at the brushes. To overcome this difficulty several means are employed. One of the most common is the use of preventive leads. These consist of resistances of german silver inserted between the armature coils and the commutator segments, as in Fig. 7, from which it is evident that the short-circuit current must pass through two of the resistances in series, thus being greatly reduced. In considering this feature, the question may arise as to whether this means offers material advantage as the useful current must also pass through these leads before it reaches the coil. It is true that a small loss occurs on this account, but as the effective current, after entering the first coil, passes through a number of other coils before it finally flows through an-

other resistance to reach the next brush, the loss thus occasioned is small compared to the total loss that would take place in the short-circuited coils were the preventive leads not used.

As a further means of reducing the short-circuit current, the brushes are made with a high resistance which, being in series with the preventive leads, also reduces this current.

Another common means of meeting this difficulty is the use of commutating poles. These are placed at the points of commutation and are so designed that they set up a magnetomotive force which counteracts that of the short-circuited coils. A wide variety of methods are employed in the design and connection of these poles, each with its own particular merit. The design depends principally upon the conditions of load, speed and service under which the motor is to operate. Fig. 8 shows a method used when it is desired either to maintain a constant torque with varying speed or a constant speed with a varying load. In this arrangement, the speed is varied by the regulating transformer; while the voltage applied to the motor varies, that of the commutating-pole winding remains constant.

Another scheme of connections is shown in Fig. 9. In this method, the voltage of the commutating pole varies with that applied to the motor.

An arrangement which has been found to give good results is shown in Fig. 10. In this the commutating-pole winding is placed in series with the armature, but a noninductive resistance is placed in parallel with the winding. The main current passes partly through the commutating-pole winding and partly through the resistance.

Sometimes it is desired to operate series motors on a mixed service, that is, either on a direct- or alternating-current supply. This can be easily done by properly designing the control apparatus. Fig. 11 shows a general diagram of connections used for accomplishing this purpose with railway motors. The solenoid-operated switch closes the alternating-current supply circuit across the contacts *A* and *B* as soon as the current is admitted to the autotransformer. The switch maintains this position as long as current continues to flow through the solenoid. However, as soon as the alternating-current supply is broken, the switch opens the bridge across the contacts *A* and *B* and at the same time closes the direct-current supply circuit across the contacts *D* and *E*, thus changing the motors from alternating- to direct-current control. It will be observed that the motors are connected in pairs, each pair being connected permanently in series. The two pairs can be operated in series by opening the switches *H*, *F*, *K* and *I* and closing switches *C* and *G*. For parallel operation switches *C* and *G* are opened.

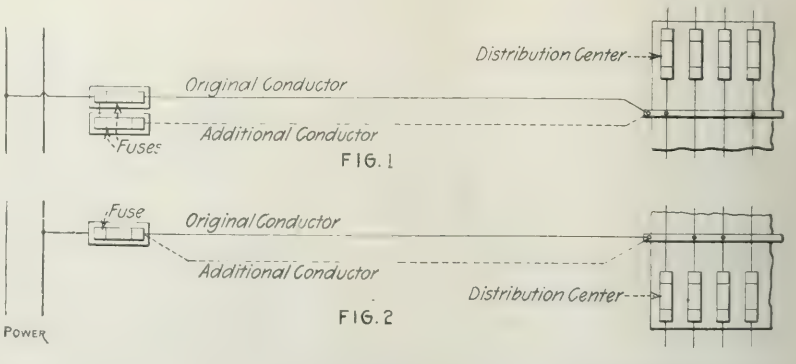
For traction work the alternating-current voltage generally employed is 11,000. The motors are usually designed to operate at 250 volts or less, the motors being connected in series groups of two. switches *H*, *F*, *K* and *I* are closed while The line voltage is stepped down to the requisite pressure for operation by means of the autotransformers.

As compared with other types of alternating-current motors the single-phase commutator motor is superior for purposes requiring a heavy starting torque and a ready adaptability of speed to varying conditions of load. The single or polyphase synchronous motors are unsuited for purposes requiring a heavy starting torque as are also the single-phase induction motors. The alternating-current series motor also possesses an advantage over the direct-current series motor in that it can be operated efficiently on a mixed service.

CORRESPONDENCE

Protecting Reinforced Conductors

Where it is necessary to increase the carrying capacity of a circuit by adding a second conductor, it may be done by one of the methods illustrated. If the wires are small and so located as to be easily broken, the reinforcing wire should be protected with its own fuse, as in Fig. 1. Where the wires are large and not likely to break, both the reinforcing and the reinforced wire can be connected in parallel and protected by one fuse, as in Fig. 2. If the second method were used for the conditions in the first case, one of the wires might break and the remaining wire would be protected by a fuse too heavy for it; hence it might become overheated and cause a fire. However, where at all feasible, the second method should be used, because with the first the current is likely to divide unequally between the two conductors, due to differences in contact resistance at the terminals.



MEANS OF PROTECTING REINFORCING CONDUCTORS WITH FUSES

CLARE V. SHEAR.

Wilkinsburg, Penn.

Reversed Polarity

In the case cited by Mr. Mitchell in the July 2 issue, it would appear that the machine was reversed in polarity by putting in the "main switch" after the machine switch was closed, while the dynamo was standing idle. If so, it would surely have caused the polarity to be reversed, owing to the difference due to the voltage of the No. 1 machine and no voltage on No. 2 machine. The usual way of restoring the polarity to its normal direction is to pass an electric current from another machine around the pole pieces, that is, through the field coils, long enough to change the polarity to the right direction. This may be done by taking a separate set of wires from another machine or by using the switch, circuit-breaker and the busbar connection to the machine.

In small plants with small machines it is customary to raise the brushes from the commutator as no current is needed in the armature. But in large stations with large dynamos where there are a hundred or more brushes it is not easy to raise them all; therefore, the cable connection from the brush-holder busbar, attached to the yoke of the machine, is disconnected and may be wrapped with tape and bent back to keep it clear from contact. The current is thrown on the machine by closing the circuit-breaker and then the switch for 3 to 5 sec., after which the circuit-breaker and the switches are pulled out in the order named. The cable is then connected to the brush-holder busbar and the machine brought up to speed and up to voltage. If the voltmeter does not show the voltage to be building up, a portable voltmeter should be applied across the terminals of the switches to be sure the voltmeter on the switchboard is not out of order. If both voltmeters do not show any voltage or sign of building up, then the operation must be repeated and the current left on the machine field coils for a longer time. I have tried this on many machines and they have all come back to the proper polarity.

R. A. CULTRA.

Cambridge, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

First Diesel Liner to Cross the Atlantic

New York City had its first glimpse of a Diesel-engine driven liner when the new Hamburg-American ship "Christian X" arrived at this port on Sept. 19. A feature which attracted much popular attention, as the vessel came up the bay, was the absence of funnels, the exhaust being through one of the masts.

FIG. 1. DIESEL SHIP "CHRISTIAN X"

In many respects the "Christian X" is like the "Selandia" of the same line which was placed in commission last spring (see POWER, Apr. 23, 1912). The latter, however, was designed for service in the Far East while the newer ship is to run regularly between New Orleans and Hamburg. Not only is she the first Diesel liner to enter the Atlantic service, but the first to cross the Atlantic.

On the maiden trip from Hamburg to Havana, thence to New Orleans, the distance of 4627 miles was covered in 17 days at an average speed of 11 miles per hour and at a fuel consumption of 290 tons of oil. This is an average of 16 tons of oil per 24 hr. when developing 2300 hp. or 0.58 lb. of oil per hp.-hr.

During 24 hr. of this trip, however, a special trial run was made on which the average performance was greatly exceeded. The total fuel consumption for this run was 9385 kg., or a little over 10 tons per hour during which the average indicated horsepower was 2383.

The ship was built at the yards of Burmeister & Wain, of Copenhagen, and is 370 ft. long, 53 ft. beam and has a displacement of 9800 tons when loaded. While designed primarily for carrying freight, she also has passenger accommodations for 22 persons.

Power is supplied by two eight-cylinder Diesel engines of the four-stroke-cycle type, the diameter of the cylinders being 20⅞ in. and the stroke 28¾ in. When driving the propellers at 140 r.p.m. each engine develops 1250 hp. A view of the engine room is shown in Fig. 2.

Two 250-hp., four-cylinder, two-stroke-cycle Diesel engines each drive a dynamo and an air compressor. The former supplies electricity for lighting and motors and the latter delivers air at 300 lb. to a tank for starting the engines. A compressor driven direct from each main engine takes air from this tank and compresses it to 900 lb. for fuel injection.

Reversing is effected by first throwing the starting lever into the neutral position; this shuts off the air and fuel supply. Another lever throws into action the reversing mechanism, which is actuated by air at 300 lb., and which shifts the camshaft so as to bring another set of cams under the push-rod rollers. The starting lever is again brought into action and admits air at 300 lb. for over half the stroke, starting the engine in the reverse direction. It might be mentioned that the reversing mechanism cannot be thrown into action until the starting lever has been put in neutral position. It is claimed that the engines can be reversed from full speed ahead to full speed astern in 8 sec. All auxiliary apparatus is in duplicate and all the pumps are electrically driven except the ballast and fuel-oil pumps which are air driven. A donkey boiler supplies a steam-driven four-stage air compressor for supplying air to the starting tanks when the small Diesel engines are not running and the air pressure has fallen.

The oil-storage tanks are in the double bottom of the ship and have a capacity of 1000 tons. From these the oil is pumped to two working tanks, each of 12 tons capacity—more than sufficient for a day's run.

The space taken up by the engine room is 43 ft. of the ship's length and 10 men comprise the engine-room force.

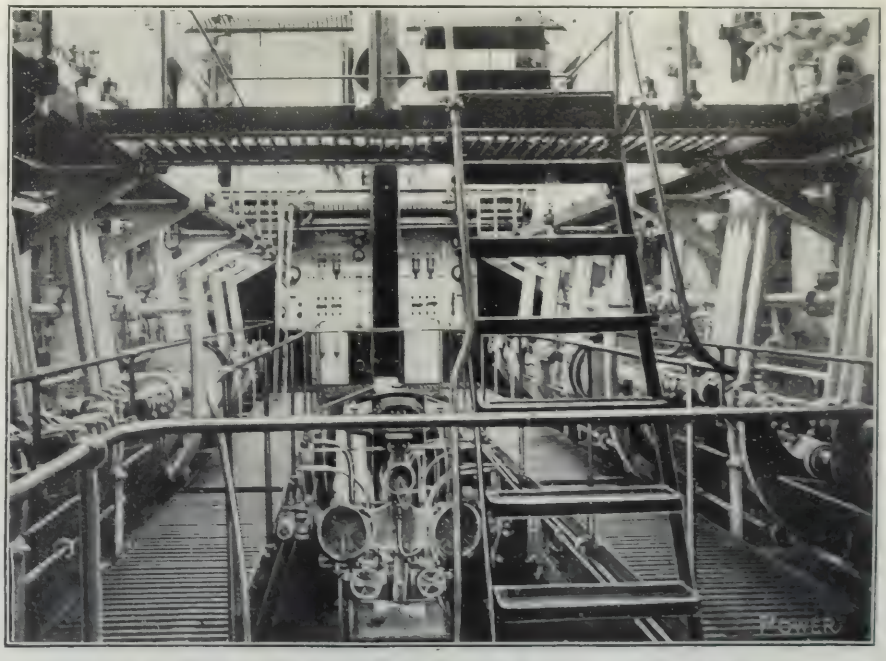


FIG. 2. ENGINE ROOM SHOWING DIESEL ENGINES ON EITHER SIDE

The Bessemer Oil Engine

The Bessemer Gas Engine Co., Grove City, Penn., has added to its product an oil engine, the design of which closely follows that of the gas engine it has been building many years. Fig. 1 shows the general appearance of the new engine. Like the Bessemer gas engine, the new oil engine works on the two-stroke cycle, and the cylinder and piston design and construction are practically the same, as Fig. 2 will indicate to anyone familiar with the gas engine. The front end of the cylinder constitutes a pump chamber into which the air for combustion is drawn by the engine piston and from which it is forced into the power end of the cylinder through the port *A*, Fig. 2. The pump chamber is closed by a head through which the piston rod works in a packing gland.

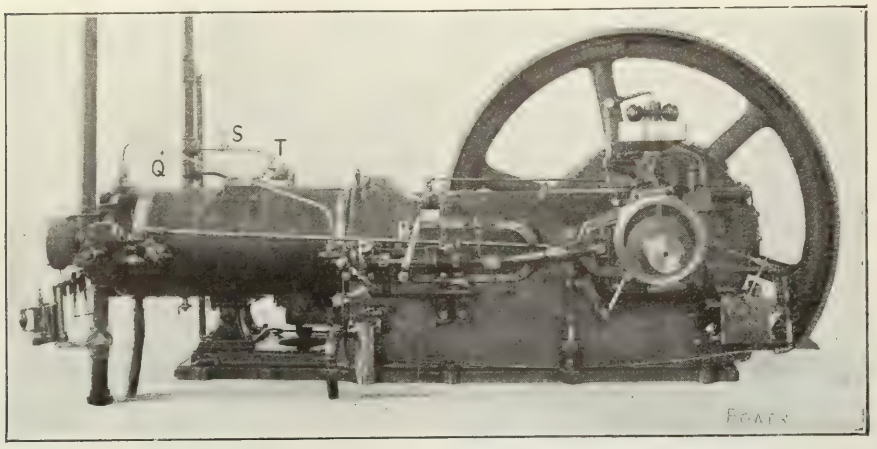


FIG. 1. BESSEMER OIL ENGINE WITH ONE FLYWHEEL REMOVED

The fuel oil is fed to the cylinder by a pump (*P*, Fig. 1), operated by an eccentric on the crankshaft, and delivers the fuel through the small pipe *Q* to the injection nozzle. The latter is shown in section in Fig. 2 at *N*. It consists of a simple plug in which is mounted a check valve of the mushroom form, normally held on its seat by a spring. When the pump delivers a charge of oil, the pressure opens the valve *N* and the oil is discharged on the lip *L* of a vaporizer *V*. This is of the hot-bulb type and the oil vapor is carried into it by the motion of the air in the cylinder, which is being compressed into the vaporizer when the oil is delivered on the lip *L*. Delivery occurs just ahead of the completion of the compression stroke by the main piston.

A special feature of the engine is the admission of water into the cylinder along with the air for combustion. This is accomplished by means of the arrangement shown at the top of the engine cylinder in Fig. 2. Water is supplied from the water-jacket inlet pipe to a little constant-level reservoir *R* whence it passes by a regulating needle valve *S* to the cup *T*. When the piston travels from the position here shown toward the cylinder head, on the compression stroke, the suction in the front end of the cylinder and the annular chamber *C* opens the check valve *U*, and the water in the cup *T* is drawn into the chamber *C*, where it is picked up by the air simultaneously drawn in through the inlet valve *I*; when this charge of air is compressed and finally forced into the power cylinder by the return (forward) travel of the piston, it carries with it the small "dose" of water which was sucked in as described.

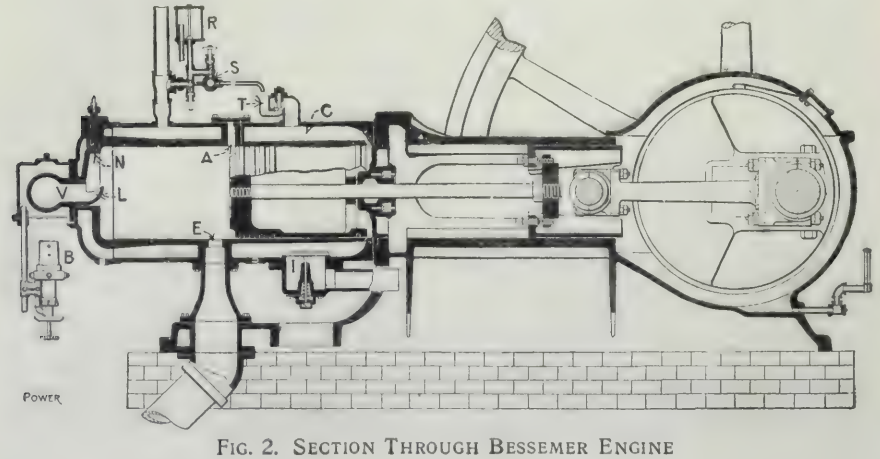


FIG. 2. SECTION THROUGH BESSEMER ENGINE

The object of admitting water with the air is to keep down the explosion pressure, which the water effects by absorbing heat from the burning fuel. The steam thus formed assists in forcing the piston forward, so that not all of the heat absorbed by the water charge is wasted. The question of heat economy, however, is subordinate. The oil engine has intrinsically such high thermal efficiency that it can well afford to sacrifice a little to smooth out the peak of the indicator diagram and relieve the maximum stress at the instant of explosion.

For starting, a small "torch" or bunsen burner *B* is provided for heating the vaporizer bulb, and a compressed-air admission valve is provided at the side of the cylinder head, the engine being turned over by air until it picks up. For stopping the engine, a handle *H*, Fig. 1, is provided, which throws the fuel pump out of operation.

Regulation is effected by a linkage which enables the governor to control the position of a pivoted dog or blade mounted on a slide which is moved to and fro by the pump eccentric. This dog engages with a stepped block fastened on the end of the pump-piston rod, the particular step engaged (and therefore the distance of pump travel) being determined by the position of the dog. The governor is a modified Jahns type.

The engine is supplied with lubricating oil by a force-feed machine operated by a link from the fuel-pump eccentric, as shown in Fig. 1.

Utilization of Waste Heat

Stahl und Eisen for July 11 prints an extract from a paper on the utilization of the waste heat from different prime movers presented by Chief Engineer K. Kutzback, of Nuremberg, at a meeting of a local section of the Verein deutscher Ingenieure. The author says the real reason why the reclamation of waste heat from internal-combustion engines has hitherto played only a small part in comparison with that from steam engines is clearly evident when the quantities of heat available are compared as in Fig. 1.

With the steam engine the available heat is present in the convenient form of steam. Furthermore, a demand such as for heating purposes can be supplied from the receiver. With the internal-combustion engine the waste heat is about equally divided between the cooling water and the exhaust gases.

With the larger gas engines the temperature of the cooling water should hardly exceed 100 deg. F. because of the danger of preignition; hence it is only in the exceptional cases that this portion of the heat is reclaimable. Cooling the jacket water by evaporation is a method frequently applied to small gas engines, but for the larger ones, it is impracticable. With oil engines the conditions are more

favorable as, through absence of the hazard of preignition, the jacket water can be raised to 200 deg. F.

While the use of the moderately warm water from the jacket is easy there are many difficulties in the way of utilizing the heat of the waste gases. These are of large volume, relatively small heat content, and poor heat-imparting capacity (about 200 times poorer than steam). In addition they are fouled by oil, sulphurous acid or soot and their direct application can seldom be considered. On the other hand, they leave the engine at a high temperature, 750 to 1100 deg. F., so that water of 212 deg. F. or with special boilers, steam of 350 deg. can easily be supplied. If the boiler efficiency be taken no higher than 60 per cent., seeing that the risk of corrosion must not be carried to the point where the cooling of the gases would condense their moisture, a recovery of about 2000 B.t.u. per effective horsepower-hour can be had from the exhaust gases of the gas engine and about 1200 B.t.u. from the Diesel engine (see Fig. 2). Although this is only about one-tenth the waste heat recoverable from steam engines, calculation has shown the great advantage of its utilization once there exists a general demand for hot water or for steam, and especially when the installation can be used at a high-service factor so that interest and amortization may not attain a disproportionate relation to the profit.

Fig. 3 shows a form of regenerator made by the Augsburg-Nürnberg Machine Works for the larger gas-engine plants. These are made either horizontal or vertical and are easily freed from scale and sediment on the water side and also permit cleaning on the gas side, though this is seldom necessary. They have the particular advantage that the cleaning can be done without taking down the exhaust piping. The illustration shows the comparatively light outer jacket removed for this purpose, it being necessary to disconnect only one flange. In general the vertical form is used for hot-water supply and the horizontal, by reason of its more favorable evaporating surfaces, for the generation of steam. Both types are built to withstand pressures up to 170 lb. gage and to resist stresses due to explosions in the exhaust line.

The following calculation gives an insight into the economy of such installations: For a 2000-hp. gas engine the regenerator and its accessories will cost from $1600 to $2100. If the annual charges for interest, amortization and maintenance be taken as 15 to 20 per cent. this will represent for one year an outlay for these accounts of from $300 to $400. With such an appliance nearly 2000 B.t.u. can be recovered per effective horsepower-hour; meaning for a 2000-hp. engine an hourly recovery of

$$2000 \times 2000 = 4{,}000{,}000 \text{ B.t.u.}$$

in the form of hot water or of steam. Assuming a cost of $0.0069 for, say, 40,000 B.t.u. generated in the form of hot water or of steam in a coal-fired boiler with coal at $3.50 per ton and a boiler efficiency of 70 per cent., the gain through this waste heat utilization will be represented by an hourly return of

$$\frac{4{,}000{,}000 \times .0069}{40{,}000} = \$0.69$$

With a load factor of 70 per cent. for the engine this means

$$8760 \times 0.70 \times 0.69 = \$4231$$

per annum gained by the utilization of otherwise wasted heat for a yearly outlay of from $300 to $400 so that at least approximately $3800 can be saved and the boiler pay for itself in six months.

In addition it must be remembered that this form of saving operates without attendance. Another favorable point is that any increased heat consumption by the engine through faulty erection or valve-gear or through abnormal ignition is largely recoverable, due to the temperature increment of the exhaust gases and is thus never wholly lost where a regenerator is in use.

If the 2000 B.t.u. per effective horsepower-hour recoverable through suitable construction of the regenerator in the form of highly superheated high-pressure steam be used in a modern steam turbine with an hourly rating of from 12,000 to 14,000 B.t.u. additional power can be developed. This gain of about 15 per cent. in power may be employed for additional scavenging or otherwise used.

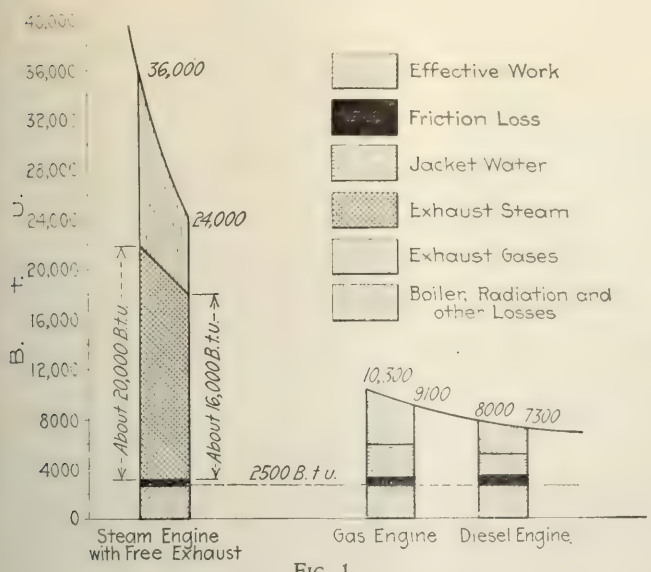


FIG. 1

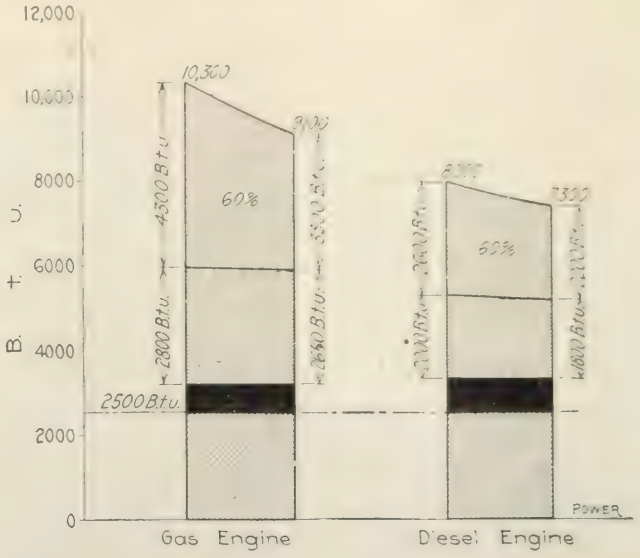


FIG. 2

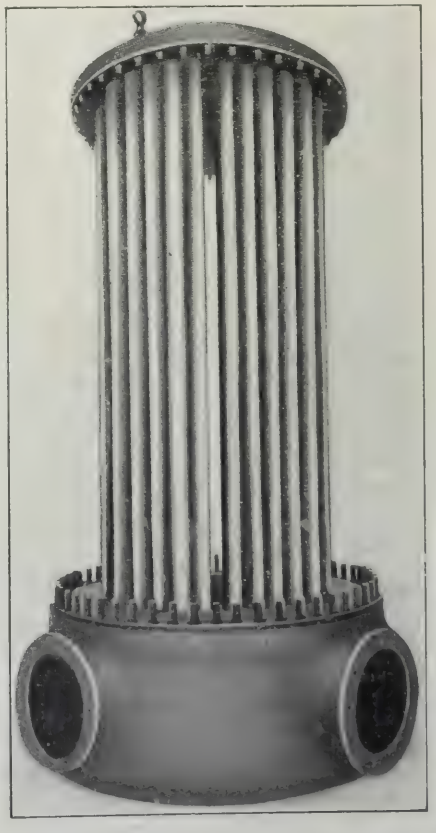


FIG. 3. HEATER WITH CASING REMOVED

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Office Practice in Estimating Heating and Ventilation*

By John D. Small

Before the meeting the author had sent out a number of sets of questions and received several replies. Some of the questions and a summary of the answers are as follows:

What rule do you use for estimating radiation?

Most favored a formula based on the heat units lost through various cooling surfaces and materials and the heat units required to compensate for air change due to leakage and exposure. Coefficient tables for this are found in several handbooks and the losses due to air change represent what the engineer's judgment dictates, except where fixed by law. Carpenter's and Mill's rules are largely used also.

How many air changes per hour do you allow in the following classes of buildings, residences, hotels, hospitals, office buildings, store buildings, theaters and factories (except where exhausters are used)?

This factor seems to be indefinite where the amount of air displaced is not fixed by law or otherwise. In buildings having no mechanical ventilation, the rate of air change would be affected by several causes, a principal one being natural leakage, which varies with the kind of construction, the exposure, the wind velocity and the building height.

Under the head of construction, the kind of window frames used has a great deal to do with the rate of infiltration of air. The sides of the building exposed to prevailing winds will show a marked increase in air displacement over the protected sides. Air currents between high buildings, due to deflection from one to the other, will often affect the surface which otherwise would be protected. Tests made by W. H. Whitten have demonstrated that with wind velocities below six miles per hour infiltration is reduced to a minimum; while with velocities as high as 30 miles per hour a very substantial effect is produced upon the rate of air change of the interior of the exposed portion of the building. Again, the leakage is relatively greater as the building increases in height, due to increased wind pressure. From the foregoing observations it is important to use considerable judgment in arriving at the maximum allowance to compensate for losses due to this element, and in the absence of a fixed rule, the following schedule, in the author's opinion, would be a safe basis for calculating the amount of heat required under maximum conditions of air change in addition to that required to offset losses through the cooling surfaces.

*Abstract of a paper read before the American Society of Heating and Ventilating Engineers at Detroit.

NUMBER OF AIR CHANGES TO BE USED IN HEATING CALCULATIONS

Office buildings—Portions above grade—1 air change per hour.
Basement, general—4 air changes per hour.
Mechanical Plant—10 air changes per hour.

Factory buildings, which have no mechanical or natural ventilation, one change per hour. For factories where large doors from the outside are frequently opened, about four air changes per hour.

Residences—having loose windows, two changes per hour.

Churches—Four changes per hour except small rooms, which should have five or six changes per hour. These data for churches contemplate mechanical ventilation.

The majority of public buildings and many of the factories require ventilation or the fan system of heating. The usual specifications of air supplies per person are as follows:

Hospitals, ordinary—35 to 40 cu.ft. per min. Hospitals, epidemic—80 cu.ft. per min.

Hospitals—Tuberculosis	Air Change min.
Detention room	6
Toilet rooms	6
Bath and duty rooms	8
Kitchen	3
Serving	10
Fumigating	10

Workshops—25 cu.ft. per min.

Prisons—30 cu.ft. per min.

Theaters—20 to 30 cu.ft. per min.

Meeting halls—20 cu.ft. per min.

Schools—30 cu.ft. per min. per child and 40 cu.ft. per min. per adult.

Hotels—Following air changes are usual:

Room	Air Change min.
Engine	6
Kitchen	1½
Restaurant	6
Base toilet	5
Billard	10
Barber shop	8
Dining room	15
Palm room	12
Buffet	8
Cafe	8
Lobby under balcony	8
Main lobby	20
Banquet hall	15
Retiring room	10
Kitchens	8
Toilets	6
All others	15

Libraries

Room	Air Change min.
Corridors	15
Basement rooms	15
Reading rooms	12
Inside rooms	8
Corner rooms	7
Toilet rooms	5

Laundries should have an air change every 4 to 6 min. Radiation on sides of buildings subjected to prevailing and cold winds would be increased 10 per cent. up to the 10th floor and 15 per cent. above.

In your opinion is it more practical to heat and ventilate with hot air only or to ventilate with tempered air and provide direct radiation for heat losses through cooling surfaces?

While opinions differed on this, providing direct radiation for use when fans are shut down seems desirable. It is argued that omitting direct radiation makes heating without ventilating impossible, as the fan must be run to heat. Direct radiation in addition to the fan system, one to offset the cooling effect of walls and glass and the other for ventilation only, makes a flexible system and admits of uniform regulation of temperature for various exposures perhaps better than the fan system only.

The relation of supply and exhaust opening in a given room sometimes results in short-circuiting and defeats thorough ventilation as well as requiring direct radiation to care for parts not warmed on this account. Apparently, therefore, where possible, combining the direct with the blast system would be not only more practical but more satisfactory results could be guaranteed.

Do you consider it good practice to install radiation in factories only sufficient for normal winter temperatures and increase pressure to compensate for deficiency when maximum winter temperatures prevail?

The consensus of opinion is decidedly against this practice, especially where exhaust steam is used for heating, as the engines would be subject to back pressure and general efficiency is reduced. The money saved on first cost of the heating system would be spent in operation later, thus resulting in poor economy in the long run. Therefore, this evidently would not be considered good practice, although owners are often influenced to cut down the first cost in this way, not fully realizing the net result.

Do you advocate using mains as heating surface or covering them throughout?

The conditions and class of buildings govern largely. Under the head conditions, would be considered the cost of covering, the location of mains and the length of run together with the length of risers in connection with the mains. For low buildings and not excessively long runs, the mains are very often left uncovered, and without bad effects. In high buildings and in central heating systems the mains must be covered to reduce the steam pressure and therefore the temperature as little as possible at the terminals. In some cases the steam has condensed to such an extent, due to surrounding temperature, that covering the mains became necessary. Most favor covering the mains, as a rule, for best results in heat distribution.

In large open rooms, such as in stores, factories, etc., would you figure the radiation the same on all sides and then place a larger proportion on the most exposed or windward side? If so, what proportion? Or would you figure it the same on all sides and add radiation on the most exposed sides? If so, how much?

The majority answered the first question affirmatively, the idea being that the heat will equalize, due to wind pressure, and eventually find its way to the opposite side of the building, whereas, if the same proportion were placed all around, the temperature on the windward side would be too low, and on the opposite side too high. To figure the radiation the same all around and then add to that on the most exposed or windward side would require more radiation than otherwise, and would not be so effective and economical as the first method.

If windows are provided with weather strips, would you reduce the amount of radiation? If so, how much?

Ordinarily the sash and frames are assumed to be wood, without weather strips. If concealed weather strips are used the radiation may safely be reduced from 10 to 15 per cent. While if metal sash and frames are used the radiation should be increased.

Open Windows with Mechanical Ventilation*

By Ralph C. Taggart

Ventilation by open windows has been discussed of late and certain misstatements have gained considerable publicity. Its advocates usually maintain that ventilation should be by open windows alone, while the partisans of ventilating systems often claim that no opening of windows should be allowed. They argue that opening windows will upset the balance of the ventilating system and make it ineffective. This closed-window rule has been enforced to such an extent that students in some schools have been suspended or expelled for opening a window.

In the author's opinion no ventilating system should be installed in which the balance of the system is appreciably affected by opening windows. Where the outdoor air is reasonably good and fairly free from dust, etc., a mechanical ventilating system will give as good or better results when used in conjunction with open windows than in any other way, and, further, the engineer who insists upon closed windows is making almost certain the unsatisfactory operation of his apparatus.

To the question what is meant by good ventilation, some will answer pure air. A layman or perhaps a doctor may say "I can tell pure air the minute I enter a room," and this is the way the air is judged by most people. In other words, pure air is what appeals to the senses as pure or refreshing. Good ventilation does not then for most people mean chemically good air. It means air which makes one feel good.

The ventilating engineer, while he has changed some of the old standards by which air is judged, still holds to a chemical test of the air. To the ordinary man, however, air which makes him feel good is air which meets certain physiological requirements. These requirements are not the same for two different individuals, and they are not the same for the same individual at different times of the day or on different days. For many individuals considerable variation may be allowed in most of the qualitative conditions of the air without much discomfort, while sensitive persons can stand only very small variations in the air conditions which surround them without feelings of discomfort.

An able surgeon, two hours after being well satisfied with the air conditions in one of his hospital wards, was very much dissatisfied with them, but a careful examination showed that the conditions were identical. In the meantime he had exercised and eaten a hearty meal, and undoubtedly needed a lower air temperature or perhaps a lower humidity to make him feel as he did at first.

Ventilation involves, therefore, a question of comfort to the occupants of a building which cannot be eliminated. The best that can be hoped for in ventilating a room containing many people is to provide healthful air which will as nearly as possible satisfy the physiological requirements of the average individual.

Imagination has much to do with an individual's comfort or satisfaction. For example, changing the position of a thermometer, has made people who thought they were cold believe they were warm. When the occupants of a building are told that the windows cannot be opened they are very apt to want to open them.

Aside, however, from this consideration, the occupants of a room should have some easy means of changing air conditions within it. No matter how carefully the system is operated, it cannot meet all individual requirements, and not allowing window opening removes the simplest, and often almost the only easy means of controlling certain air conditions.

Where the system is controlled automatically, if the apparatus gets out of order, the room is overheated and discomfort caused if windows cannot be opened. Even if it happens only once in a year; this often engenders an animosity toward the heating apparatus that is astonishing.

Open-window ventilation in rooms occupied by many people is generally suspended in cold weather, because the occupants will not keep the windows open. This means practically no ventilation in the colder weather.

Some claim that air warmed (sometimes styled "canned" air) and then admitted to a room is injurious. If the air is properly washed, very little dust remains, and even if it were all burnt up by the few degrees of warming, the decrease in the oxygen content of the air would be negligible. Air in passing through a steam heater need not be warmed to the excessive temperatures often stated. With steam at 212 deg. F., air cannot be heated in a ventilating apparatus to that temperature, and practically none of it need be heated above 98 deg., or less if desired.

Some engineers maintain that rooms in which windows are opened draw a large part of the air supplied by a fan or blower, keeping other rooms from receiving their share. They may contend that with the ventilating system in operation the various rooms are under considerable pressure. With vent or exhaust flues of any reasonable size, plus the chances for air leakage around the room itself, very rarely will the pressure in any room be sufficientiy above the outdoor pressure to be measurable except by the most delicate instruments. The additional flow of air toward a room with open windows, due to any lessening of pressure within the room, therefore, is usually very small. More likely air will blow into the room from out of doors, reducing the amount taken from the ventilating system and leaving more instead of less for the other rooms.

Assume considerable pressure is carried within the room itself (although this indicates a poorly designed ventilating system) and the wind in a direction to remove air through open windows. An excess of air then will be taken from the ventilating apparatus, but it is not so great as might be imagined, especially if the connections in the branch flues from the main ducts to the individual rooms have some appreciable resistance, as they usually have.

The increase in air supply varies only as the square root of the increased difference of pressure, and if in the original design the branch connections have been slightly choked so as to place the larger part of the duct resistance in these branches, the increase in air discharge into a room with open windows may be made practically negligible.

Some may say that this resistance in the branch connections is very undesirable. It is not if the main ducts are of proper size to minimize the friction within them, and this is desirable, for the main ducts are usually horizontal, and the horizontal ducts should always be large enough to be easily accessible and to keep the total friction resistance within the duct system small. If the friction in the main ducts is small, the friction

*Abstract of paper read before the American Society of Heating and Ventilating Engineers at Detroit.

in the branch connections may also be small, and yet large enough for that in the main ducts to be negligible in comparison.

How small an increase in duct sizes is required to reduce the friction in ducts is often not appreciated. For the same quantity of air delivered the friction varies inversely as the fifth power of the diameter; hence, if a duct or pipe diameter is doubled, the friction is reduced to $\frac{1}{32}$ of its former value. If the duct is increased in diameter only 50 per cent., one of two-thirds its diameter will offer more than 7½ times as much friction, or if the diameter is increased only one-fourth, the friction resistance is reduced in the ratio of more than three to one.

The practice has become common to reduce duct sizes so much that many of the horizontal ducts are practically inaccessible and the friction resistance so great that the use of apparatus frequently is abandoned because of the power required to operate the fan. Main ducts and main branches, therefore, should be made large, while vertical branches should be properly reduced.

Another argument against open windows is that the good air from the ventilating system will escape directly out of doors. Generally speaking, this is not so, and in any event the trouble can usually be rectified.

Some, however, may say that whereas we get pure washed air from the ventilating apparatus, only dusty air comes in through the windows. This is denied by open-window enthusiasts. The truth lies between these two statements. In very dusty localities, air admitted directly from out of doors without washing or cleaning may be objectionable, but usually pretty good air in summer is obtained through open windows, and if it is good enough in summer it should not be rejected because of similar chemical constituents in the colder weather.

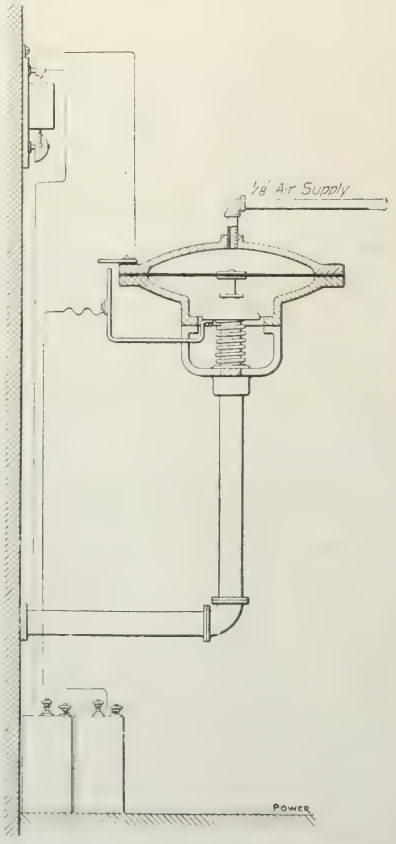
Opening windows to admit air may change the humidity, but if it renders the room more comfortable the author sees no reason why it should not be considered an improvement. There is an element of temperature stratification of the air which seems to be stimulating and invigorating to many people. In other words, air at a dead uniform temperature is not so desirable as air having some differences in temperature. This, insofar as desirable, can often be obtained easily by open windows in conjunction with the ventilating system. It can also be obtained without open windows, but usually not nearly so readily.

Open windows with ventilating systems, in general, therefore, increase the quantity of fresh air supplied; provide means for admitting cool air; give an additional temperature control, and often furnish an easy method of adding to the possible comfort of occupants. Open windows should not, ordinarily, and absolutely cannot, in a well designed ventilating apparatus, interfere with the operation of the system.

CORRESPONDENCE

Air Pressure Alarm

The accompanying sketch shows an electric bell alarm which is used to give notice of a reduction in the air pressure in connection with the operation of thermostatic temperature regulation on radiators and hot-water tanks. This simple alarm has frequently been of great service in giving timely warning of something wrong with the air supply, thereby preventing the hot-water tanks from becoming overheated and the rooms from getting too warm. The alarm is of value in any plant where the air pumps are



ALARM FOR COLD-STORAGE ROOM

widely separated from the rest of the machinery and attendants are not always in the immediate vicinity.

The alarm is made from the head, diaphragm, spider, wooden saucer and stem of an ordinary diaphragm valve as used on radiators. This is screwed onto a piece of ⅜-in. pipe and with an elbow, nipple and flange is fastened to the wall of the boiler room. A ⅛-in. pipe from the air tank operates the diaphragm. One contact, a strip of thin brass, is fastened to the top of the head and extends over the edge; the other, a piece of brass rod flattened at one end, is fastened to the wooden saucer or block and is bent upward around the head. These contacts are connected by wires to a battery and bell. While the air pressure is up the contacts are separated, but when reduced, the spring, which acts against a 15-lb. pressure, will force the diaphragm upward, causing contacts to complete the circuit and ring the bell.

FREDERICK G. LEMKE.

New York City.

Saving 50 Tons of Coal

When engaged for my present position I was requested to fire the boiler and to run a 10x20-in. plain slide-valve engine. There was not a steam trap on the place, not even a reducing valve for the heating system, so that when live steam was used it was necessary to turn it on full pressure. The factory was heated by exhaust steam daytimes, and after heating the feed water there was no steam left for one coil. With the exception of the office, live steam could not be used for heating during the day, as the boiler was overloaded. As a consequence the shop was very cold and much time was wasted by the workmen in getting warm.

It was my idea that a reducing valve would help, but the management would not stand the expense. One day I happened to run across a steel drum with a hole in the top of it and decided to make a steam trap. I drilled a hole for a 1¼-in. pipe on one side and a hole on top for a 1½-in. pipe. I got a 1¼-in. valve, filed the thread off the stem and fastened the float to it and the trap was soon complete. All returns from the upper floor came down in one pipe, so I put the trap on the end of it with excellent results. This left out the first floor with nine steam coils, each having a separate drip, but at that conditions were improved.

Next winter I got the reducing valve and a few fittings and put in return pipes and straightened other pipes that needed it. I put the home-made trap on the return of the heating system and it was possible to heat the factory and office on 4 lb. back pressure. The trap and other improvements saved at least 50 tons of coal during the season and on the coldest winter days the temperature on the first floor was maintained at 60 to 65 deg. F. and the remainder of the factory at anywhere from 70 to 80 deg.

THOMAS D. TRIO.

Winsted, Conn.

In Denmark, all the towns of 5000 inhabitants and over are now provided with public electric service. The largest electric stations are to be found at Copenhagen, and at present there are three large plants in operation giving a total of 27,000 hp. Current is supplied for the city mains as well as for the tramway lines.

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POWER

The Brotherhood of Power Workers

An organization, the aim of which is to educate, protect and uplift all having to do with power-plant work is the Brotherhood of Power Workers, referred to elsewhere in this issue. The coal passer is as eligible to membership as the chief engineer and the educational program provides that his questions shall receive the same consideration as those of his chief.

The organization aims to increase the wages of power-plant workers, not by fostering antagonism between employer and employee, but by educating the worker to a point of efficiency where the employer will be glad to pay for the superior service. It does not participate in labor disputes unless appealed to by a member and then offers the services of an arbitrating committee. However, its members are forbidden to fill positions vacated by organized strikers.

Those desiring to obtain city or state licenses or take civil service or other examinations are coached by an examining board. There is a sick and accident, as well as a death benefit to which all members are entitled. It is proposed to eventually add an insurance department to facilitate payment of the above benefits.

This organization should prosper, grow and realize its aims inasmuch as it is broad in its scope and sound in principle, and it will so long as it allows nothing to divert it from its present constructive policy. We hope the fireman member will receive the consideration it is proposed to show him. Firemen are handicapped in getting the knowledge necessary to make them engineers. In most plants they are not allowed to leave the boiler room while on duty. This is as it should be, but the opportunities for becoming familiar with equipment in other parts of the plant are lacking. The work is hard and he is in no humor to read or study when the day is done. The discussions that he would hear and participate in at the meetings of a good organization would, therefore, be of inestimable value not to the fireman alone, but as much so to the engineer.

We wish success to the Brotherhood of Power Workers. It is founded on right ideals and should be an important factor in improving the conditions and elevating the standards in the field which it serves.

To Foreword Contestants

So numerous have been the replies to the offer for forewords and cartoons printed on the first page of the Aug. 13 issue, that we must bespeak the patience of the contributors until the editors can pass upon them. As far as possible, those obviously unavailable are returned at once. Others surely acceptable or likely to be with a little modification, or suggesting ideas that can be made use of, are put aside for further consideration. The receipt of all matter is immediately acknowledged, but for the above reasons final acceptance may be a little deferred. Contributions will be paid for as fast as accepted, although it may be many weeks subsequently before they appear, since only one can be used each week.

The above will show the necessity of our asking indulgence on the part of contributors that they may not be too hasty in concluding they are being neglected.

A word of caution to those who have yet to submit matter. Many contributions have involved POWER itself in the treatment of an idea. Much as we appreciate the compliment, we do not want to hold up the paper as a panacea for all operating ills. It would seem indelicate and moreover advertisements of POWER are not what we are after. One very carefully drawn cartoon we regretted having to return for this reason. Had we anticipated the possibility of contestants falling into this error, we would have warned against it in our original offer. We trust that all further contributions will be strictly impersonal so far as POWER is concerned.

As stated in the offer, we particularly desire cartoons. The ideal cartoon tells its own story with no explanatory text

beyond a caption and is directly applicable to the operating field. It may point a moral, contrast existing power-plant conditions, etc. Some submitted have been more appropriate for educational textbooks or advertisements and one was strictly allegorical. Such are not wanted, nor those that harp continually on the dirty engine room or the slovenly or bibulous engineer. These conditions and habits are too obviously wrong to need further emphasis.

The range of subjects is wide and it is the quest for new ones that occasioned our offer. Although many contributions have been received already, they have scratched only the surface of the possible field and we trust our readers will continue to put their wits to work and send in more. Good ones will always be available, but efforts must be confined to the power field and the paper left out of sketches or discussions.

Technical Training

About ninety per cent. of our children it is stated finish the grade or grammar schools, five per cent. the high schools and two per cent. the colleges and universities, including the professional schools.

As the industrial tendency is to use machinery more and more, good engineers are needed to plan and design, and skilled men to produce the parts, and assemble and erect them. Capable salesmen also are needed to dispose of the output. The amount of skill and training required varies greatly with the position. There are plenty of men for the lower places, for machine tending, but they become fewer as the demands upon them become greater and more highly technical.

An attempt is being made to meet these industrial needs with engineering, technical, vocational and trade, correspondence, Y. M. C. A. and public evening schools, as well as schools run by the manufacturers in connection with their works. But more are needed and more must come. The wage earner must specialize along definite lines. But what can a man do who has left off his education with, say, the grade school and has started in the industrial field? He has begun to see that more education is needed, more knowledge of the principles and reasons underlying his work or the trade in which he is engaged.

Naturally the question must be answered and a few suggestions may guide him in his choice of methods. He must consider that he has available only a certain amount of energy each day. If he is busy at work all day, he may be sleepy, fagged out and unable to take advantage of night schools or to study evenings by himself in correspondence lessons. Many have, however, been helped by these schools, especially when day duties have allowed a chance to study lessons at the same time. Studying alone and getting out hard problems unaided is fine training, but gives no means of comparing one's self with others; it is like a one-man race, no pace- makers or competitors by which to measure one's ability.

A course of training, taking a subject at a time and studied but a little each day, extends the time needed to complete the desired subjects over many years. Of those persons who start out with strong intentions, very many fail to continue to the end, perhaps because studying is hard for them, or their daily duties change, or their determination gradually diminishes.

As to the length of time needed, it must be recognized that nature has certain laws in the realm of mind as in that of matter, and that the one especially applying here is that the older one grows the longer it takes him to learn new things; and so while a man is slowly completing night and correspondence-school courses, he is growing older and the last work will come still more slowly. Also the advantage of having the training is less, when it has finally been secured.

It would then be better where possible to make a business of getting the needed education and training while young. This means in part attending some school where the student can devote himself to study, can receive needed guidance and help day by day, can measure himself by his classmates, can learn from their questions, can take advantage of the best time of his life for learning. And when he has mastered the principles and has been started in their applications, he can continue to apply them to advantage to himself and his employer in the everyday work of the plant.

If it is a question of money, it is suggested that he write to the different schools that offer studies and training along his line, to trade schools, to institutes giving industrial training, to engineering schools and to state universities. Let him learn first the cost per year and then figure out what he can make per year in the summers and weeks when the school work is not in progress. There are numerous young men already putting themselves through these schools, paying their way by various means, and many of them standing well up in their classes.

If he does rely upon his own exertions to pay his way through, he must be prepared to take extra time for the studies, even an extra year, and study fewer subjects each year to allow the time for the paid work. This is necessary so that he can study at more nearly full pressure and to better advantage. Otherwise he may "sacrifice his education for the means of getting it."

Another plan is to borrow the money. A young man hesitates to run into debt. But one would borrow to go into a profitable business and what better business could a young man engage in than in transmuting borrowed money into the capital of an education. This borrowing must be done by giving a note. With no property for security, let him procure a life-insurance policy of perhaps a thousand or two thousand dollars, and borrow on it from some kind friend or philanthropic party, a sum to cover his estimated expenses plus the insurance premiums and interest for the period of his studying. The life insurance is made payable to the lender of the money and secures him in case of the death of the borrower. It is assumed that the young man is honest and will pay back the money if he lives. The note may be renewed from time to time, after he is again earning money, but for reduced amounts.

It is easy reckoning to see how long it would take him at his present wage to make complete payment; after his course of study he should be able to increase his wage gradually and so clear up the debt sooner. It is quite a surprise to find how many are ready to extend a favor to an earnest, honest, capable fellow who is seriously striving to improve himself.

To the young man with no family ties this suggestion is worth thinking over. What is worth while comes only by effort. Keep striving, but if you expect the chips to fly, use the edge of the axe and see that it is sharp.

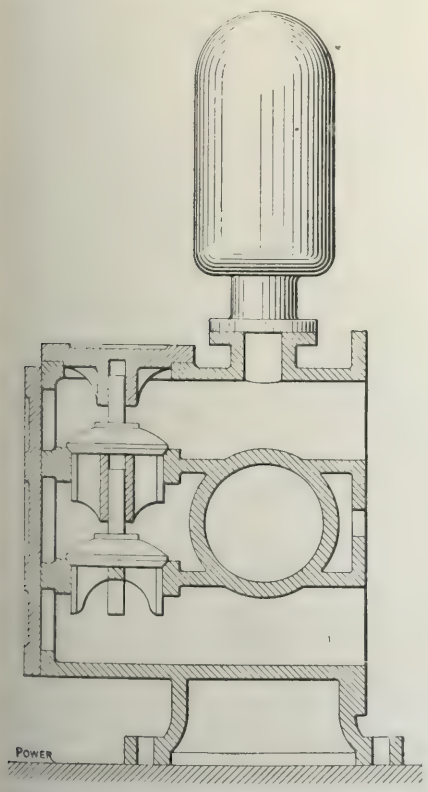
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Effect of Valve Position on Pump Efficiency

The effect of the position of the valves in an ordinary plunger pump upon its efficiency is interesting. Upon several occasions I have observed plunger pumps with the suction valve immediately under the discharge valve, as shown in the illustration. In testing them I was astonished to find that the actual quantity of water delivered was in excess of the theoretical capacity of the pump.

The explanation is very simple as I understand it. A certain momentum is given to the water in the suction pipe during the suction stroke, and when the plunger starts to return, this momentum holds the suction valve open for an appreciable length of time, allowing water to pass directly from the suction through the delivery valve with the re-directed water from the plunger.



SHOWING POSITION OF VALVES IN PUMP

This, of course, would not occur to any extent if there were much head on either valve, but I thought it an interesting point to bear in mind in pump design.

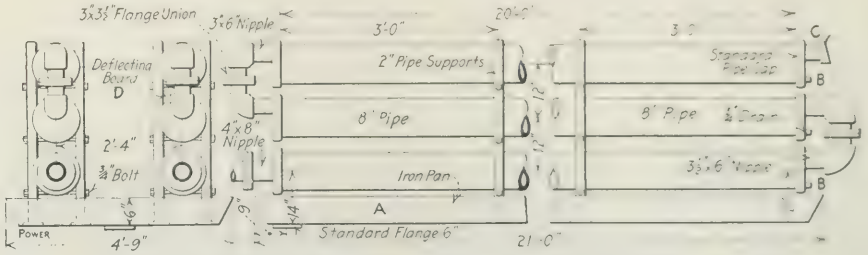
G. H. HUMM.

Loughboro, England.

Condenser Made of Eight Inch Pipe

The illustration gives the details of a steam condenser made of old 8-in. pipe and a few fittings. The iron pan *A* was made at the factory, but a wooden pan could be made more cheaply and be just as serviceable.

The pipes for distributing the cooling water are not shown. Perforated or slotted pipe can be used very satisfactorily. A wooden trough with saw teeth cut along the two upper edges, is an effective and easily constructed means of distributing the water. The ¾-in. drain pipes *B* at the lower end are to be connected with suitable pipe and fittings to take away the condensate.



HOME-MADE CONDENSER CONSTRUCTED OF PIPE AND FITTINGS

The pressure-relief valves *C* are not altogether necessary, their use depending upon local conditions. The deflecting boards *D* (one board only is shown) assist materially in distributing the cooling water over the lower pipes, and also prevent water from splashing outside of the pans. These boards (of 3-in. flooring) should be on both sides of the lower pipes and extend the length of the condenser. The edge next to the pipe should be cut to let the water pass under it.

The writer does not know how much steam, in pounds, the condenser will handle, but the following sizes of pumps, etc., are now exhausting into it:

Four 8x36-in. deep-well pumps, operating at 20 r.p.m.; one 7 and 4 by 10-in. boiler-feed pump; two 9-in. air pumps; one 8 and 6 by 4-in. single, double-acting, brine pump, operating at 40 r.p.m. and one high-speed 9x12-in. slide-valve engine. So long as the condenser is reasonably free from scale, and a thin skim of water is kept running over the 8-in. pipes, no steam escapes from the relief valves, one being shown.

A number of its features make this condenser attractive to the owner, as well as the engineer. The first cost is small, when compared with other condensers of equal efficiency and durability. The one shown cost $90, including new pan, caps, ells, nipples, pressure-relief valves, and labor to assemble the parts. The engineer at the plant can put together and erect the condenser without any trouble. The round surface of the pipe facilitates removing the scale. If the pipes are well painted before being put into operation, a sound blow from a light hammer will loosen the scale. The round surface also aids in better distributing the cooling water, getting at the steam from all sides, as it were.

When the condenser is badly coated with scale, there will be a loss out of the relief valve not only of steam, but of considerable distilled water. This serves to remind the engineer that it is time to clean the condenser.

It is readily seen that repairs can be made to this style of condenser very cheaply and quickly. Should a hole come in one of the 8-in. pipes, tap it out and plug it. The nipples can be renewed easily and the flange unions allow renewing the 8-in. pipe. I have had one of these condensers in operation for six months, using cooling water at about 75 deg. F., having about 30 grains per gallon of scale-forming substances, and have found it very satisfactory.

GAY A. ROBERTSON.

Louisville, Ky.

Steam Gage Problem

Upon taking charge of a plant, I noticed that the steam gage did not agree with the safety valve. It behaved as follows upon firing the boiler:

On Monday morning I could not raise over 20 lb. pressure on the gage, and the safety valve, which was set to operate at 100 lb., would blow off. The gage registered an increasing pressure until

at the latter part of the week it would agree with the safety valve.

I concluded that the gage was out of order, and not having a reliable one, I asked the superintendent to have the gage tested. His answer was that there was no money to pay for it. I called his attention to a circular issued a short time before which stated that the engineer in charge and the superintendent were held responsible for the safety of steam boilers and other machinery. Then I was allowed to have the steam gage corrected, which cost 75 cents.

The gage was corrected with a standard gage and the variation was found to be about 7 or 8 lb. I replaced the gage and after a few weeks it began its same old trick, but this time I thought the trouble was in the safety valve. I overhauled and cleaned it thoroughly and figured the weight that should be placed on the lever to allow it to blow off at 100 lb., and the marks on the lever corresponded with the weight.

To get a new gage was almost financially impossible. The heating season was almost over, and as there was a high-pressure gage on the heating boiler, I placed it in connection with the other. On starting up the gage from the steam-heating boiler registered 100 lb. and the safety valve blew off while the other gage registered about 20 lb. as usual. At the first opportunity I took the steam gage down, examined, cleaned and replaced it, but with no better results.

Can some interested reader explain what caused this trouble? I believe the spring has lost its elasticity, but why did it register the correct pressure toward the latter part of the week?

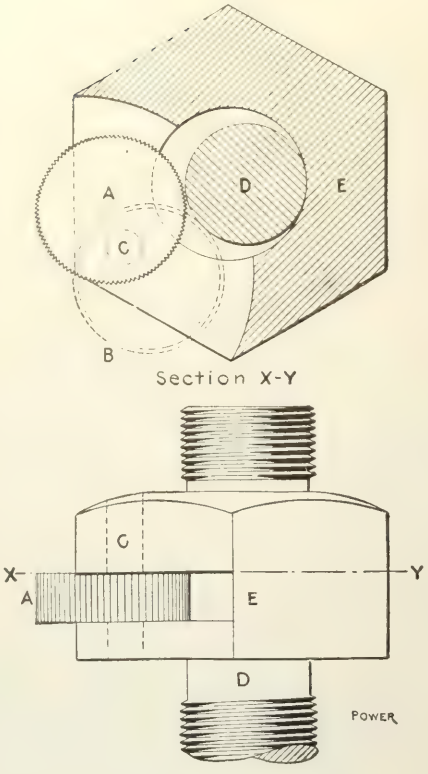
ANGELO BELMONTE.

Stewart, Nev.

A Simple Stud Puller

The illustration shows how a simple stud puller may be made. *E* is a hexagonal piece of machine steel of suitable size, depending on the job. An old nut will not do very well as there is not sufficient metal between the thread and the outside surface. A plain cylindrical hole is made in *E*. A slot is milled in one side, as shown, to accommodate the knurled eccentric *A* and a hole is drilled for the pin *C*. The eccentric *A* should be made of tool steel, hardened and the temper drawn to purple. The pin *C* should also be made of tool steel and the temper drawn to the same color. The body *E* may be left soft but is better case-hardened.

The action is as follows: For screwing a stud in place, the stud *D* is screwed in as far as it will go by hand. The puller is then placed over the stud and the knurled eccentric pressed against the stud *D* in the position shown at *A* in the cut and the wrench applied. Obviously, the greater



SHOWING CONSTRUCTION OF STUD PULLER

the resistance of the stud *D* the greater the grip of the knurled eccentric *A*. To remove a stud the eccentric is swung to the position *B*.

L. A. SUVERKROP.

New York City.

A Thoughtless Fireman

On Sundays and holidays in winter one of our boilers is always under steam pressure as required by insurance companies. Out of twelve boilers, three and sometimes four are washed every Sunday. An 8 and 8 by 6-in. feed pump furnishes washing water at a pressure which quite well taxes the strength of our big fireman who handles the hose.

The regular feed pump moves very slowly and feeds the boiler in service, since the heating system remains in operation perpetually during cold weather. The feed pipe, therefore, always contains hot feed water. An independent line is used to convey the wash water to the boiler room and at two remote points are hose attachments. The wash water comes directly from under ice on the pond, so the temperature can be conjected. Now that I have given a fair idea of the conditions necessary to the understanding of what follows, here is the point of my story:

This fireman had been on the job five years and every winter the washing of boilers was torture to him. He would remove the two lower front handhole plates, the upper front, and so on with the rear plates and then stick the hose in the top hole, start the pump, and wash the crown-sheet in the usual manner, using a 4-ft. pipe screwed to the hose end. The other end had an elbow and nipple to deflect the flow sideways between the tube ends. The scale and other impurities followed the water, falling down into the water-legs. Then, after all crown-sheets had been thus treated, he would stop the pump, remove the 4-ft. pipe and use one a foot long instead, for lower regions. After shoving it into one of the apertures he would start the pump again. Sometimes the force would wrench out the rubber pipe and irrigate the floor, and the big fellow would rush in all excited from the pump room in an endeavor to impede the disturbances. In chasing the wagging hose which twisted and swung with all the nimbleness of a snake, from one end of the room to the other, he would receive a none too gentle whack that would cut and perhaps lay him low for a moment.

I entered one cold Sunday morning, and found him crouched low, sucking his fingers, and on the verge of tears. He said his fingers were frozen stiff from washing the first boiler. I asked him to show me how it had happened. He stuck his stout arm into one of the lower holes and inserted the spout of the hose with it. He pulled out whatever scale he could with his hand while the water washed away the remaining mud.

I felt the water and must admit that it was pretty icy; it was no wonder his hands were stiff. "Keep on," I said, "and I'll fix things so you'll be more comfortable." As I spoke I reached out and cracked the feed valve; we heard the hot water oozing. I opened it a trifle wider and gradually he felt the water grow warmer. It was a simple expedient, but in five years he had never thought of it.

LUKE MARIER.

Fall River, Mass.

Blowing Down Return Tubular Boilers

Some engineers declare that the proper time to blow down a boiler is before starting the day's work or at least some time after closing down and after the sediment has a chance to settle, but I prefer to blow down a boiler while the water is in circulation, for the following reasons:

By giving the sediment a chance to settle, it will collect on the upper part of the tubes and the lower part of the boiler shell. I consider that very little that lodges near the entrance of the blowoff pipe will ever move from its position. But if the boiler is blown down while the water is in circulation a large portion of the impurities held in suspension will immediately head for the entrance of the blowoff pipe, and more of it will

be carried away in the same amount of water than if the sediment is allowed to settle.

A. J. Ferguson.

Vernon, B. C.

Zinc Slabs as Corrosion Preventives

Our insurance company has recommended slabs of zinc suspended in our steam boiler to prevent pitting of the tubes and sheets.

The boiler is of the Scotch marine type, dry back, return-tubular, 50-hp. and used for low-pressure heating. I would be pleased to have suggestions from readers as to the best method of doing this, and of the results obtained.

Howard L. Stafford.

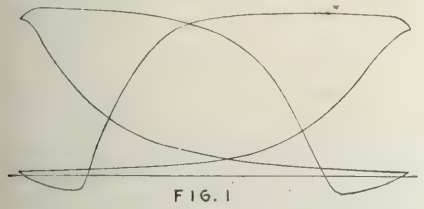
New York City.

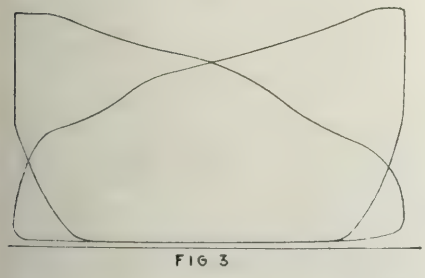
Poor Engine Economy Due to Faulty Valve

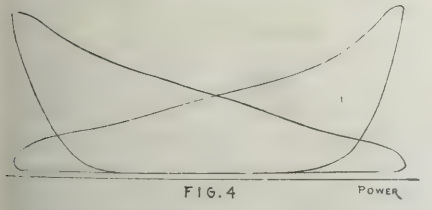
Our large engine would not pull the load, and the owners thought it too small. I blamed the engine and on indicating it obtained the diagram, Fig. 1. This card set some of us guessing, so we removed the valve, Fig. 2, and took it to a near-by machine shop and had $\frac{1}{16}$ in. planed off of the exhaust edges and $\frac{3}{16}$ in. from the steam edges. Then we replaced the valve and advanced the eccentric until the lead was what we thought about right—something less than $\frac{1}{16}$ in. We then took the diagrams, Figs. 3 and 4; they are not bad for a throttling engine. The engine could carry the load, and much more than was needed. I think there are more of these engines running in such condition than many suppose.

C. R. McGahey.

Baltimore, Md.







Diagrams Showing Improvement of Valve Alterations

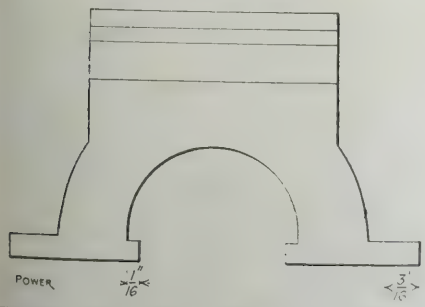
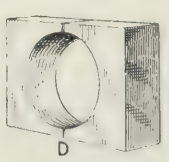
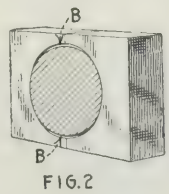


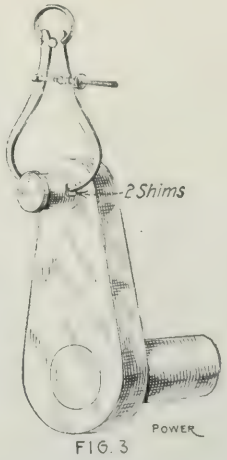
Fig. 2. Slide Valve Showing Amount Cut from Steam and Exhaust Edges

Fitting Crankpin Brasses

A large cross-compound condensing Corliss engine of which I once took charge, I had been warned would give trouble because the high-pressure crankpin was always running hot. My predecessor explained that the trouble started about a month before, when the master mechanic fitted the brasses. I asked the master mechanic how he did the work and found, as I have before, that in fitting them they were put on the pin separately and fitted for high places. After much trouble I refused to keep the job unless the brasses were taken out and fitted according to my method, as life became unbearable under such conditions.







Correct and Incorrect Pin Bore. Calipering Pin for Correct Bore

I first took off the strap, jib and key and put a block next to the inner box to take the place of the end of the rod, and put two $\frac{1}{16}$-in. shims between the boxes at *D*, Fig. 1, and then keyed up solid. Then I took them to the lathe keyed up solid and took two pieces of shims of the same thickness as before at *D*, Fig. 1, and placed them on the side of the pin and calipered the diameter, as shown at Fig. 3. The boxes were then bored to this size. Then the shims were taken out and fitting commenced. Boring the pin hole larger than the pin is done so that when the boxes are fitted there will be a space at *BB*, Fig. 2, which collects oil and distributes it evenly over the pin and allows of further adjustment.

Next I took some Prussian blue paint and put a very thin film over the pin, and then put the boxes on the pin and keyed up easy so there would be no play, using the block in the strap instead of the connecting-rod. The boxes and strap as connected were turned around on the pin, thus marking the high places, then taken off and these spots scraped. This was done several times until a perfect fit was made. The boxes were put on and the engine run slowly for about an hour with no signs of heating, and they are runing today with no tendency to heat. Where loads are light, it may be that brasses, put on the pin and fitted separately, have worked all right, but when the brasses must be keyed up solid on the pin a different condition exists and calls for a more refined fitting of them.

A. C. Waldron.

Revere, Mass.

"Knocking"

Some men have a disagreeable way of "knocking," and creating discord wherever they work, because they think such tactics will gain favor for them from the boss. Some bosses also "knock" to the "old man" on slight provocation.

When a workman is unfortunately responsible for a trivial offense it is better to attribute it to "accident" and not create discord. Nine times out of ten, such procedure, together with a private, but friendly reprimand, will bring about better results than running to the old man; but if the case demands reporting at all, why not give such information to him in the presence of the accused?

A case recently came to my observation where an engineer "knocked" a fellow workman in this manner. While moving some lumber up over a small steam pipe a board was dropped upon the pipe, causing a 2-in. gate valve to rupture. The engineer reported him in such a manner that the firm kept $4.75 of his wages to pay for the damaged valve.

A few days later this engineer's carelessness caused the ruin of a 50-ft. stack which he was lowering. He was quick in reporting the trouble as being due to his negro helpers, and he "got away with it."

I do not advocate this policy of laying many things to unavoidable accidents so that someone may break or tear up anything that suits his fancy, but to give everyone a square deal.

I do not believe that a man should be held responsible for accidents to the extent of being made to pay the damage.

Lloyd V. Beets.

Nashville, Tenn.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Stopping Leaky Rubber Diaphragms

In the July 30 issue, M. S. Weil tells of stopping the leaks in rubber diaphragms with molasses. This treatment may be all right for clear rubber or for diaphragms where the molasses only gets on the surface, but I tried it once for single-tube bicycle tires, the construction of which is much the same as many diaphragms, and found that it separated the plies of the tire and so weakened it that it went to pieces in a couple of days.

A very effective treatment and one that does for any kind of diaphragm is an application of ordinary rubber varnish or roofing cement that comes with some kinds of paper roofing. Leather, rubber or cloth thoroughly impregnated with it will remain air or water-tight for a considerable length of time and it cannot hurt or injure their construction.

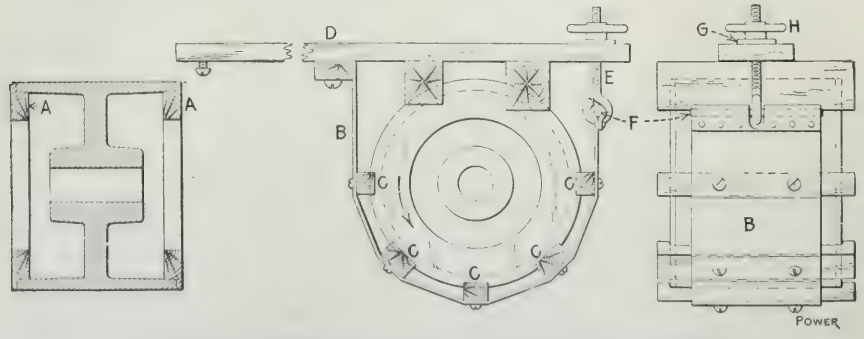
A. A. BLANCHARD.

Lakeland, Fla.

Making a Water-cooled Brake

A. T. Kasley's experience with soap and water to prevent a prony brake from sticking is very similar to mine. Graphite will answer the same purpose.

An inexpensive water-cooled prony brake pulley for temporary use may be easily made for testing small engines or motors without buying a flanged pulley. For testing a 15-hp. motor with a normal speed of 1720 r.p.m., an old 10x12-in. iron pulley that fitted the motor shaft was used. Flanges were made for it by cutting rings from a ¾-in. pine board to make a snug fit on the inside of the pulley rim. The rings were about 1½ in. thick with a small flange to fit against the outer edge of the pulley. After forcing these rings into place on each side of the pulley and soaking them with water, a flanged pulley was produced as good for a temporary job as those with the flanges cast in place. A cross-section of the pulley with the flanges *A* in place is shown at the left of the illustration.



SHOWING CONSTRUCTION OF WATER-COOLED PRONY BRAKE

The brake band *B* was made from an old piece of rubber belting with hardwood strips screwed on the inside to act as as rubbing pieces. One end of this was firmly fastened to one end of the lever arm *D*, the other end was doubled back and riveted, making a small loop. At the middle of the belt a hole was cut into this loop and a long threaded eye-bolt *E* placed with the eye in this hole so that a bolt or short rod *F* could be run through the loop and the eye. The threaded end of the eye-bolt was then run through a hole in the lever arm, and a washer *G* and nut or a handwheel *H* with a threaded hole was screwed on the bolt. This gave a simple means of adjusting the tension of the brake band.

By a little preliminary calculation the length of the lever arm may be found which will materially reduce the amount of the final calculations necessary to determine the horsepower from data taken during the test. In the old familiar formula of

$$hp. = \frac{2\pi Pln}{33{,}000} \text{ or } \frac{Pln}{5252}$$

where

$\pi = 3.1416$;

P = Weight;

l = Length of the lever arm;

n = R.p.m.

If l is made equal to 5.252 ft. we have for the hp. $\frac{Pn}{1000}$. If this is too long or too short an arm for the size of the brake, any even multiple of this length may be used and the constant changed as follows:

10.504 ft.	2 Pn/1000
5.252 ft.	Pn/1000
2.626 ft.	Pn/2000
1.750 ft.	Pn/3000
1.313 ft.	Pn/4000

The brake operated very satisfactory up to 22 hp., which was the maximum capacity of the motor. To supply the water which was evaporated from the inside of the pulley, a short piece of ¼-in. pipe with a cock on one end was fastened into the side of an old gallon tin can, the can was filled with water and placed so that the cock extended to the inside of the outer flange of the pulley, then the cock was adjusted to supply water at just the rate of evaporation.

B. A. SNOW.

Canon City, Colo.

Overspeeding of Turbines

I have read several accounts of turbines overspeeding and was especially interested in Mr. Martin's letter in the June 11 issue and Mr. Hawley's comment on it in the Aug. 13 issue.

A little more than a year ago I was promoted from watch engineer to chief of a 3000-hp. turbine plant having two turbines of 1500 hp. each, where we had considerable trouble with turbines overspeeding. My first endeavor was to overcome this trouble and I was completely successful.

Observations prompted me to make the following changes: We removed the primary valve and thoroughly cleaned the pistons and cylinders. I made new bushings for the valve spindles, allowing 0.007-in. clearance and ground in the needle valves and closed them tight, depending on leakage through the bushings to operate the valves. I also fitted a plug in the hollow end of the spindles so that no steam could pass up through them. Then I drilled a $\frac{5}{16}$-in. hole through the cap under the lower end of the spindle to allow any accumulation of water to pass into the turbine. In assembling the valve I gave the steam piston $\frac{5}{16}$-in. clearance and the dashpot piston ⅛-in. clearance.

As the admission of steam for operating the valves is greatly reduced, the tension on the springs is necessarily reduced in proportion. This is best determined by running the machine and increasing

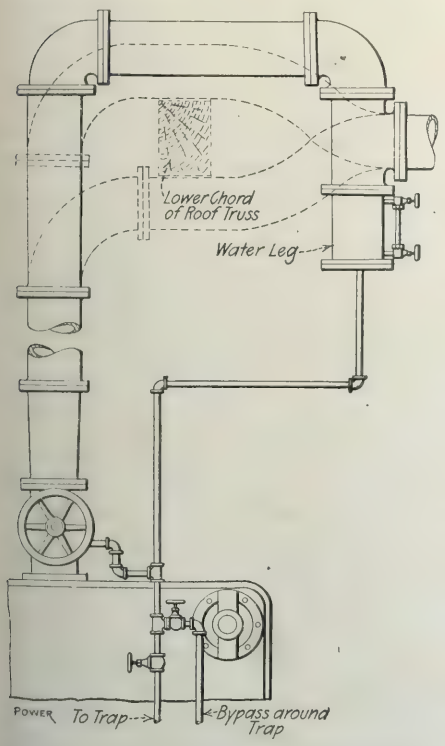
the tension until the valve seats properly. I have also entirely discontinued using oil on these valves, and during the past year our turbines, carrying a load that fluctuates from zero to overload with 175 lb. steam and 29-in. vacuum, have given no indication of overspeeding.

DAVID CRAFT.

Flushing, L. I., N. Y.

Damaged Steam Pipe Due to Faulty Design

In the issue of July 30, under this heading, the original design, and the suggested improvement are alike bad. If the designer of the original arrangement called himself an expert in the design of high-pressure steam-pipe lines, he certainly misunderstood himself. Mr. Parker's suggested improvement would probably be less effective than he believes, as when the water began to trickle back to the boilers (and that suggests that it would be interesting to see the rest of



PIPE LINE IMPROVED TO INSURE DRY STEAM TO ENGINE

that pipe line) there would probably be serious damage done to the pipe.

One of those trickling drip jobs was responsible for an accident in a plant where I worked, making the place resemble what the infernal regions are supposed to look like, with scalding steam, flying bricks, pipe, fittings and pieces of engine all about.

My advice to every man in any way responsible for high-pressure steam work is: Do not take any chances, never under any circumstances design a pipe line with the water flowing against the steam; keep them both going the same way, and get it away from the steam as soon as possible. If a vertical line occurs in a design, always provide a large drip leg the full size of the pipe, the contents of which should be taken care of with a trap. If the trap fails to operate throw the line open to blow the steam through.

The full lines in the illustration show how I would reconstruct the pipe line in question. The dotted lines show the original piping and Mr. Parker's suggested improvement.

J. P. LISK.

Brooklyn, N. Y.

Mr. Parker severely criticizes the design of a 10-in. steam pipe to an engine. He proposes a change in piping to prevent the accumulation of water. Now, if I get the correct idea of conditions, I agree with the owners that the pipe is all right as it is and that they should look elsewhere for the trouble.

In discussing the suggested change we will admit that with the throttle valve closed and no flow of steam in either direction the condensation would trickle back to the boilers after his proposed change, but what would it do while the engine is running? It seems to me that we would have an ideal "pocket," the velocity keeping the water from trickling back and the upset in the pipe preventing it from passing on with the steam or out through the trap, as it should. This trapped volume of water will steadily increase until a "slug" will be taken over into the cylinder and a blown out cylinder head or a wrecked engine may result.

This "trickling back" proposition is not always satisfactory in practice. I would suggest installing a steam separator of ample capacity directly over the throttle valve and a trap that can be relied upon to drain this separator; to be safe two traps might be connected in multiple so that if one failed the other could do the work.

I do not see how, with the throttle closed and the engine shut down and no flow of steam in the pipe, there could be water-hammer to break the elbow referred to after the vertical pipe has filled with water. Some explanation is due here.

Incidentally, the blowing of steam into the engine room after the elbow had broken and having to go to the top of the boilers to close the valves are in themselves a very strong recommendation for using automatic stop valves. I am a great believer in automatic stop and check valves and think they should be used on all boilers, especially where two or more are connected in a battery, as a safeguard against just such occurrences as this one.

M. C. ST. JOHN.

Houston, Tex.

J. W. Parker shows a queer design of piping, but so far as his trouble is concerned I do not believe the piping to be the cause. The drip arrangement should be looked after and should keep the pipe clear of water. I know of no method of piping to a dead end that is not subject to water-hammer.

The change proposed by Mr. Parker is likely to give the engine a dose of water while running if the pipe is of ample size, unless it has a steep pitch toward the boilers, and that would look worse than the present arrangement. I should have a long sweep bend at the engine, and no trap on the steam-line drip to bother me, but instead would put a valve on the drip line at the throttle. When the engine is shut down this valve would be left "cracked" to the atmosphere. With a little experimenting there need be very little waste and the pipe can be easily kept free of water.

LEIGHTON JOHNSON.

Exeter, N. H.

Small Turbine Accidents

Paul Hoffman's article on steam turbines is excellent, but as I came to where he stated that "in small turbines a man looked at them once or twice a watch, etc., and they keep running for years without interruption or repairs and practically no maintenance cost," I wondered if he had stood beside a small impulse geared turbine and listened to it running apparently as smooth as a watch and suddenly to hear a rumble, an explosion of oil and a screech that made one's hair stand on end. This happened to me and upon opening the gear case I found the gears chewed and broken, the teeth on the driving spindle gone and everything more or less damaged.

The cause was never found, to my knowledge, even by experts sent by the company. At another time the outboard bearing on the high-speed spindle showed a slight haze of smoke and I was called to examine it. On touching the bearing with my hand it felt about as usual and the sight-feed oiler was running freely. Picking up an electric hand light, I bent over to examine the open end and was struck on the cheek by a drop of babbitt, while a stream of it poured out of the bearings end. I dismantled the bearing and after cleaning the end of the shaft, placed a new brass in position. Everything started up well and has been running satisfactorily ever since. I was unable to find any cause for this unless the watchman lied about the oiler being clear up to the time I was called.

The governor working loose one day caused the pumps to lose their suction as the turbine would not come up to its speed on starting. It had me guessing for some time before I discovered what was wrong.

These and many other small things that have happened in the last 5 months have made me think there is opportunity for trouble on small turbines as well as on large ones.

This turbine runs day and night, and while its load is steady, it has rather a hard row to hoe and maybe I should not complain if it lays down once in a while.

C. L. MALLERY.

Myerstown, Penn.

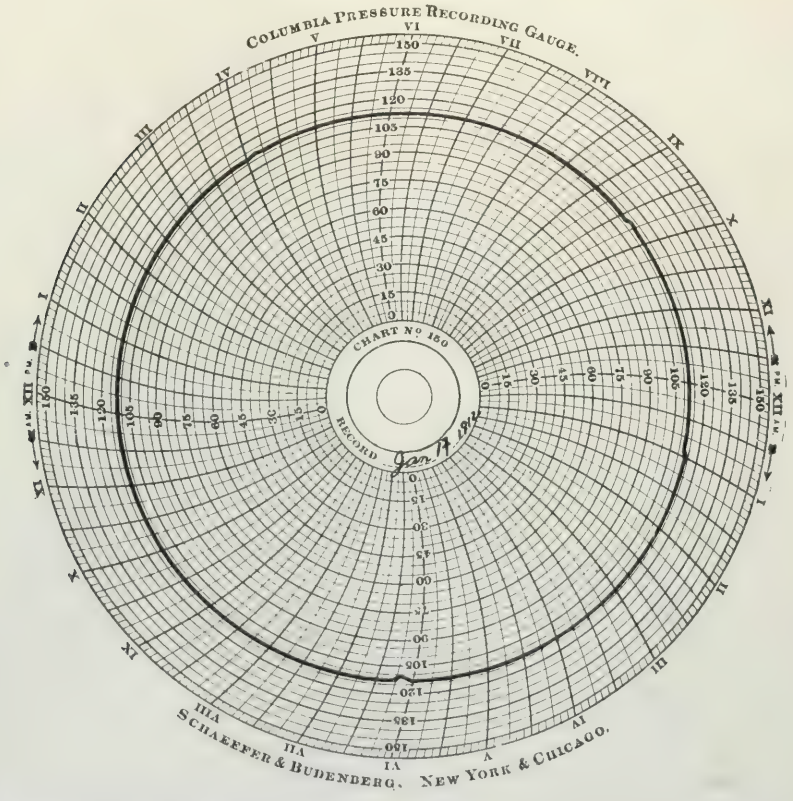
Comments on Pressure Chart

In the Aug. 13 issue, B. M. Seymour shows a recording steam-gage chart and asks for comments on it. I do not think it a very good one as the steam pressure is not uniform but varies over 10 lb. on nearly every firing, which seems to occur about seven times an hour.

Mr. Seymour should fire oftener and in smaller quantities, using the alternate method. Firing one-half of the furnace one time and the other half the next is better for hand firing, and will make less smoke than firing the whole furnace at once, and will keep the pressure more uniform. This method comes as near to automatic stoking as I have been able to get with hand firing. From the amount of coal used I think the boilers are working at about capacity. There may be some cases where automatic stokers will not give as good satisfaction as hand firing, but in many instances where the stokers fail the cause is not in the stoker but in the way in which they are operated.

The accompanying chart was taken during the heavy-load period at a plant using three 200-hp. boilers which frequently carried 100 per cent. overload between 6 a.m. and 4 p.m. and especially from 7 a.m. to 10 a.m., while the buildings were being warmed up.

These boilers were fired by underfeed stokers using forced draft of about 2 in. of water in the air duct, and burning bituminous slack and crushed mine-run coal. The fires are cleaned twice during the 24 hr. One man fires the three boilers, and then sits down one-half the time. The coal is brought in on a small truck and shoveled into the hopper. The dampers in the stack are set at a certain point and only changed as the load varies. The forced draft is controlled by a regulator which operates on one or two pounds variation of pressure. The speed of the fan ranges from 150 to 450 r.p.m. as required.



STEAM CHART FROM PLANT USING MECHANICAL STOKERS AND CARRYING HEAVY OVERLOADS

On this chart the greatest variation of pressure during the 24 hr. was 4 lb., and that was while cleaning fires. No matter how hard the fires are forced there is no smoke except while cleaning fires.

In another plant of which I had charge there were six 250-hp. boilers, all of which were formerly required to keep up the steam pressure. These were hand fired with coal that was bought by the ton and not for what it was worth to us. A new 30x40-in. mill engine was installed, and the boilers thoroughly cleaned. Coal was bought that gave the best results, and not by its looks, name or price per ton. One boiler was taken off the line and held in reserve. Most important, the boiler-room force was reorganized so that better results could be had. The firemen were all foreigners and paid low wages. The pressure was held at 145 lb., with the safety valve set at 150, and although the mill engines started and stopped frequently, the recording steam gage seldom showed a variation of over 10 lb. during the 24 hr. and the safety valves opened not more than three or four times a day. The boilers were fired alternately about every 3 min. using from two to four scoops of coal at a time.

In this plant the firemen did not wait for the steam to drop before firing, but used a pilot light as a guide instead of the steam gage. As the lever on the damper regulator moved on a 2 lb. variation of pressure, I placed a contact on the end of the lever so that when the pressure was below 144 lb., the contact was closed and a red light would show in the boiler room where it could be seen by all. When the pressure came up to 146 lb. and the lever raised to operate the dampers the light went out. The reappearance of the light was a signal to fire one side.

I do not know at present just how much coal was saved by the change, but the output of the mill was increased over 50 per cent. with a small increase in coal consumption, due mostly to good coal and close attention.

J. CASE.

Hyattsville, Md.

What Made the Engine Stop?

In the issue of July 23, Mr. Bullis gave an account of an engine stopping with the throttle and exhaust valve wide open, and asks what the trouble was.

I know of a high-speed engine directly connected to a 100-kw. generator that came to a standstill while about two-thirds loaded, but unlike Mr. Bullis' engine it could be turned by hand shortly afterward. Then the throttle was again opened and the engine gave good service.

Upon examination after the day's run, we found the piston had "frozen" from lack of proper lubrication, and the cylinder scored badly. After the cylinder had been smoothed up and the rings cleaned the engine was again put into commission, and, although it required a lot of oil to keep it quiet, is now running as good as ever.

If Mr. Bullis will look into the cylinder he will very likely find the walls badly scored.

G. B. LONGSTREET.

Somerville, Mass.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address— Not for Publication

Heat Units per Boiler HP.

What number of heat units transferred per hour constitutes a boiler horsepower? Why do authorities vary from 33,000 heat units upward, and are the differences of importance?

G. B. L.

In deference to the popular application of the term horsepower to boilers, the committee of Judges of the Centennial Exposition, 1876, adopted as a unit of boiler horsepower the evaporation of 30 lb. of water fed at 100 deg. F. into steam at 70 lb. per sq.in. above atmosphere, these conditions being considered to represent the average consumption of steam per horsepower per hour by good non-condensing engines in use at that time. According to steam tables quoted by D. K. Clark, mainly based on Regnault's experiments and formulas, in which a heat unit is taken as the quantity of heat required to raise 1 lb. of water at atmospheric pressure from 32 to 33 deg. F., the latent heat of evaporation of 1 lb. of water at atmospheric pressure and 212 deg. F. is 965.2; and reckoned from 32° F. and atmospheric pressure, the total heat of 1 lb. of steam at 70 lb. pressure above atmospheric pressure (84.7 lb. abs.) is 1177.8 + British heat units. According to this nomenclature, the heat required per hour per boiler horsepower by the Centennial standard is

$$[1177.8 - (100 - 32)] \times 30 = 33{,}294 \text{ } B.t.u.$$

and the heat equivalent from and at 212 deg. would be

$$\frac{33{,}294}{965.2} = 34.494 \text{ } lb. \text{ } of \text{ } water$$

The American Society of Mechanical Engineers' Committee on Boiler Tests, 1899, adopted substantially the same value by recommending the definition of a commercial boiler horsepower as the equivalent evaporation of 34.5 lb. of water per hour from and at 212 deg. F. and this value is generally accepted. According to the tables referred to this is equivalent to 34.5 × 965.2 = 33,299.4 B.t.u. Discrepancies frequently arise from employing 1178 in place of 1177.8 and the use of 966 in place of 965.2. Employing the Marks and Davis steam tables the same evaporation would be represented by 34.5 × 970.4 = 33,478.8 heat units, according to the standard and formulas used in the construction of those tables.

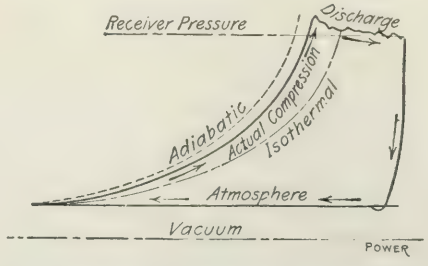
The differences are rarely of importance in an engineering sense, for, although each of the computations are construed as based on the same heat unit, the widest discrepancy would amount to less than 1 per cent.

Air Compressor Diagrams

How does the compression line of an air-compressor diagram compare with that of a theoretical diagram; and why is the discharge line irregular and higher than the receiver pressure?

C. H. B.

In consequence of loss of heat during compression the compression line of an actual diagram will be lower than the theoretical adiabatic line and higher than



the theoretical isothermal line, as illustrated. The discharge line of the actual diagram is irregular on account of vibrations of the discharge and check valves and has to be higher than the receiver pressure to overcome friction of valves, piping and passages employed for conducting the compressed air from the cylinder to the receiver.

Permanent and Electromagnets

What is the difference between a permanent magnet and an electromagnet?

V. F.

A permanent magnet is of hard steel and when once magnetized remains so indefinitely. An electromagnet, on the other hand, is of soft steel or iron and its magnetism is dependent upon the maintenance of a current in the coil surrounding it.

High Duty Pump

What is the difference between a high-duty and a low-duty pump?

D. P.

A high-duty steam pump is one in which provision is made for working the steam expansively—while in ordinary or low-duty pumps the steam follows full stroke.

Increasing Chimney Draft

The force of draft of a chimney 100 ft. high is ⅜ in. of water. How much would the force and amount of draft be increased by increasing the chimney height to 150 ft. with the same size of flue?

J. F. R.

For the same temperature of chimney gases the force of the draft is in direct proportion to the height, and for the increased height of the chimney would be

$$\tfrac{3}{8} \text{ } in. \times {}^{150}/_{100} = \tfrac{9}{16} \text{ } in. \text{ } of \text{ } water.$$

Neglecting a small proportional increase of friction, due to the increased velocity of the gases, the relative velocity and amount of draft would be as the square roots of the heights, viz., as $\sqrt{100}$ to $\sqrt{150}$, or 10 to 12.44, being an increase of 24 to 25 per cent.

Air Pressure and Temperature

What will be the pressure of air if taken at 100 lb. gage pressure and heated to 500 deg. F.? W. C. C.

It is necessary to take the initial temperature into account. When the volume is constant the absolute pressure will vary as the absolute temperature. If the initial temperature were 150 deg. F. the initial absolute temperature would be 150 + 460 = 610 deg. and at 500 deg. F. the absolute temperature would be 500 + 460 = 960 deg. As a gage pressure of 100 lb. = 114.7 lb. absolute, then for an initial temperature of 150 deg. F. raised without change of volume to 500 deg. F. the pressure of the air would become

$$114.7 \times \frac{960}{610} = 180.5 \text{ } lb. \text{ } absolute$$

or

$$180.5 - 14.7 = 165.8 \text{ } lb. \text{ } gage \text{ } pressure$$

Steam Connections for Damper Regulators

Where should the boiler-pressure pipe of a damper regulator be connected?

W. M. M.

The pressure pipe should be connected to the steam space of the boiler whose draft is to be regulated. Connections for diaphragm damper regulators may be placed on the steam connections of water columns, but steam-piston regulators and those which discharge steam should be independently connected with the boiler steam space.

Study Questions

This Week's Questions
Last Week's Answers

(101) A 12-in. ball is in a corner of a room. What must be the diameter of another ball which can touch the first one while both rest on the floor and touch the same walls?

(102) How fast must an automobile go to literally "cut circles around" a man walking at 8 miles an hour? Both start from the same place at the same time and the automobile makes a complete circle around the man and meets him again while he is walking in a straight line.

(103) What should be the rim weight of a cast-iron flywheel 15 ft. in diameter, for an 18x36-in. Corliss engine running at 90 r.p.m.?

(104) Steam at 110 lb. gage is admitted throughout the stroke of an 18x30-in. engine, and exhausted at atmospheric pressure. What is the theoretical indicated horsepower if the engine makes 75 r.p.m.?

(105) Steam enters a condenser at 11 lb. abs. Water enters at 53 deg. F. and leaves at 110 deg. F. The condensed steam leaves at 120 deg. F. If the engine uses 26 lb. of steam per horsepower-hour while running at 53 hp., how much condensing water would be required theoretically and how much cooling surface should the condenser have, assuming that the number of heat units transmitted per square foot per degree mean difference of temperature between the steam and water is 192?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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(96) If the engine were 100 per cent. efficient, the developed horsepower would equal the indicated. When the mechanical efficiency is 90 per cent. only $\frac{90}{100}$ of the indicated horsepower will be available, or $85.6 \times 0.90 = 77.04$ hp.

(97) Let

x = Weight of an ell;
y = Weight of a tee.

Then

$$6x + 7y = 33 \qquad (1)$$

and

$$7x + 6y = 32 \qquad (2)$$

From (1)

$$x = \frac{33 - 7y}{6} \qquad (3)$$

From (2)

$$x = \frac{32 - 6y}{7} \qquad (4)$$

Equating (3) and (4)

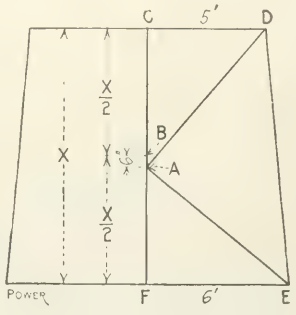
$$\frac{33 - 7y}{6} = \frac{32 - 6y}{7}$$

$$231 - 49y = 192 - 36y$$
$$13y = 39$$
$$y = 3 \text{ lb.}$$

Substituting value of y in (1)

$$6x + 21 = 33$$
$$6x = 12$$
$$x = 2 \text{ lb.}$$

(98) The diagram represents a vertical section of the tank, trapezoidal in form. CF, the vertical axis of the tank, is the altitude of the trapezoid, and must be determined to calculate the volume of the tank.



Lines drawn from A to D and A to E complete two right-angled triangles $ACDA$ and $AEFA$. From the relations of the sides of a triangle

$$\overline{AD}^2 = \overline{AC}^2 + \overline{CD}^2$$
$$\overline{AE}^2 = \overline{AF}^2 + \overline{FE}^2$$

By the conditions of the problem $AD = AE$; therefore

$$\overline{AC}^2 + \overline{CD}^2 = \overline{AF}^2 + \overline{FE}^2 \qquad (1)$$

It was given that

$CD = 5$ ft. (½ the top diameter);
$FE = 6$ ft. (½ the bottom diameter);
$AB = 6$ in. = ½ ft.;
$BC = BF$.

Letting $x = CF$

$$BC = \frac{x}{2}$$

$$BF = \frac{x}{2}$$

and

$$AC = BC + AB = \frac{x}{2} + \frac{1}{2}$$

$$AF = BF - AB = \frac{x}{2} - \frac{1}{2}$$

Substituting in (1)

$$\left(\frac{x}{2} + \frac{1}{2}\right)^2 + 5^2 = \left(\frac{x}{2} - \frac{1}{2}\right)^2 + 6^2$$

$$\frac{x^2}{4} + \frac{2x}{4} + \frac{1}{4} + 25 = \frac{x^2}{4} - \frac{2x}{4} + \frac{1}{4} + 36$$

Reducing

$$x = 11 \text{ ft.}$$

The volume of the tank can now be found by the prismoidal formula

$$V = \frac{h}{6}(A_1 + 4A_m + A_2)$$

where

A_1 A_m and A_2 = Top, middle and bottom areas, respectively;
h = Altitude = 11 ft.

A_1 is the area of a circle 10 ft. in diameter = $0.7854 \times 10^2 = 78.54$ sq.ft. A_2 that of a 12-ft. circle = $0.7854 \times 12^2 = 113.1$ sq.ft. and A_m that of a circle

$$\frac{10 + 12}{2} = 11 \text{ ft. in diameter} = 0.7854 \times 11^2 = 95.033 \text{ sq.ft.}$$

Substituting in the formula

$$V = \frac{11}{6}[78.54 + 4(95.033) + 113.1]$$
$$= \frac{11}{6}(78.54 + 380.132 + 113.1) =$$
$$\frac{11}{6} \times 571.772 = 1048.249 \text{ cu.ft.}$$

Since 1 cu.ft. = 7.48 gal., the contents of the tank will be

$$1048.249 \times 7.48 = 7840.9 \text{ gal.}$$

(99) 1 hp. = 746 watts; hence 25 hp. = $25 \times 746 = 18{,}650$ watts. With 85 per cent. power factor this would be equivalent to

$$18{,}650 \div 0.85 = 21{,}941.17$$

apparent watts.

The total power in a three-phase three-wire circuit is

$$E \times I \times 1.73$$

where E is the voltage between any two wires and I the current in each wire.

$$E \times I \times 1.73 = 21{,}941.17$$

$$I = \frac{21{,}941.17}{500 \times 1.73} = 25.37 \text{ amp.}$$

(100) To raise 200,000 lb. up a grade of 50 ft. per 1000 ft. without frictional resistance would require a force of

$\frac{50}{1000}$ or 5% of 200,000 lb. = 10,000 lb.

and if the friction is 2 per cent., then to overcome friction would require:

2% of 200,000 *lb.* = 4000 *lb.*

making a total of 14,000 lb.

Overcoming this resistance at the rate of 1000 ft. per min. would require

$14{,}000 \times 1000 = 14{,}000{,}000$ *ft.-lb., of energy*

which would be exerted at the rate of

$$\frac{14{,}000{,}000}{33{,}000} = 424.24 \text{ hp.}$$

Sun Power Engine

An interesting report concerning irrigation by sun power is made by the British consul at Alexandria, who comments on the arrival from Philadelphia of the Shuman sun-heat absorber, which, he says, was tested and found to be satisfactory. The plant is being erected at Meadi, near Cairo, and will be used to pump water from the Nile to irrigate the surrounding land. Several improvements have been added to reduce the cost of working.

In Egypt, says the U. S. *Consular Daily Report*, both coal and other kinds of fuel are expensive, but plenty of sunshine can be relied upon at all times of the year, so that the experiments with this sun-power plant will be of interest to farmers.

Chemistry's Earning Power*

It may fairly be claimed for chemistry that it is at once the most fundamental and comprehensive of all the sciences. It is directly concerned with the air, the water, the food, the materials upon which we expend our labor, and the things we buy and sell. Chemistry can now determine the suitability of a given water supply for use in boilers or for the requirements of any special line of industry. Furthermore, it supplies the means for correcting undesirable characteristics in a water supply as by use of filtration apparatus, coagulants, and water-softening systems.

That gas is now so cheap is due in large part to the development of the chemical process for making water gas by passing steam through a bed of glowing coals and to the chemical processes for gas enrichment. By the Blau gas system illuminating gas is now produced in liquid form and distributed in steel bottles to isolated consumers like kerosene.

The gas mantle by which the illuminating power of gas is raised from 16 to 60 candles on a consumption of 3½ ft. an hour constitutes one of the most signal triumphs of chemical research.

In the development of electric lighting the chemist has played a part scarcely less important than that of the electrician. The arc light was first shown by Davy between charcoal points and was maintained by the current developed by the action of chemicals in the enormous battery of the Royal Institution. To Faraday we owe the discoveries upon which our modern methods of generating electricity are based. The early history of the incandescent lamp is a chronicle in equal measure of the difficulties of finding a proper material for the filament and those of producing the requisite degree of vacuum in the bulb. Both problems were solved by chemistry which first supplied the carbon filament made by dissolving cellulose, squirting the solution into a thread of the required diameter, drying and carbonizing the thread and thereafter flashing in an atmosphere of hydrocarbon vapor to deposit carbon on the filament precisely where and in exactly what proportion its original inequalities of resistance to the current made necessary. More recently Whitney and other chemists have since given us lamps of an altogether new order of usefulness by employing new materials, as tungsten, for the filament.

The improvements in incandescent lamps in the last 10 years have resulted in the saving of $24,000,000 a year in the cost of lighting as compared with the cost of equal illumination by the older types of lamp.

To the art of illumination Wohler and Willson have contributed the calcium carbide and acetylene; Pintsch and Bláu have developed separate systems for the lighting of trains and houses; to Hewitt we owe the mercury lamp, to other inventors the flaming arc, to Nernst the high efficiency lamp and, long before them all, to Bunsen the blue-flame burner utilized by Welsbach and which constitutes the basic element in gas stove.

American manufacturing is in many respects the most intensive in the world. Nowhere else is a plant scrapped so quickly to be replaced by larger, faster and more efficient machines or labor so speeded up by piece work, bonuses, motion studies, gang organization and the other devices of the efficiency engineer. Our manufacturers understand these things, and what they understand they want. They do not understand chemistry, naturally they do not propose to have any chemist teach them their business. This is reflected in the attitude of their subordinates, who, like their masters, cut themselves off from that great coordinated and organized body of knowledge brought together by thousands of highly trained minds through the incessant questioning of nature during a hundred years.

Consider a few concrete examples of the earning power of chemistry. Cylinder oils generally cost about what one is accustomed to pay. Plants which employ a chemist pay from 19 to 27c. Manufacturers who do not need a chemist commonly pay 45c., 65c. or even $1.50 a gallon. There is probably not a large plant in the country in which, if it is not already under chemical control, the lubrication account cannot be cut in two. In the engine room of one large cement plant the average monthly cost for lubricants had been $337. It is now $30. A concern paying 37c. a pound for a special grease now buys on specification for 5½c. and the mill still runs. One company saves $12,000 a year on cutting oils alone; another, $3600 on boiler compounds.

Chemistry points out the only proper way to buy supplies which is on the basis of their industrial efficiency by means of specifications defining the quality desired and rigid tests to make sure that quality is secured.

*Abstracted from a lecture by Arthur D. Little, Boston, Mass.

The plants and business of the George V. Cresson Co. and Morris Engineering Co. have been consolidated under the title Cresson-Morris Co., Philadelphia, Penn. The new officers are P. H. Morris, president; A. S. Morris, vice-president; Gouverneur Cadwalader, secretary and treasurer; Henry G. Morris, consulting engineer; Joseph M. Hewlett, office manager; Charles H. Moyer, general representative. The new company claims that by combining the facilities of both old companies it can insure much better service and extend its field of usefulness.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Well, for the love of peat! The people over to Washington are telling us that peat in this country is produced only in small quantities and used chiefly as fertilizer and chicken food. We were of the opinion that we had peat to burn!

Sir James Donaldson, a braw body frae bonnie Dundee, said at the recent meeting of the British Association for the Advancement of Science, that spellin' is of nae consequence. Spell as ye like, says Jamie. Hoot, mon! 'Tis a peetiful time POWER would hae wi' its readers if some restraint were not put on the editors by the circumspect printer lads. Are ye gang daft, Jamie?

A railroad company has restricted its maximum passenger-train speed to 70 miles an hour. Who said we are speed-crazy?

Thomas A. Edison now rings up on time-card No. 1 at his laboratory in West Orange, N. J. His card for one week registered 95 hr. 44 min. This is not only an endurance test, but it also shows one reason why Mr. Edison is called a "wizard."

A Maryland power company has served the 400 villagers of Conowingo with notice to trek elsewhere. It will build a dam across the Susquehanna at that point and wants the ground. "Dam" is *the* popular word in the vicinage of Conowingo today.

Maine is some prohibitor! Now its adjutant-general prohibits the use of the Portland armory for a power-plant machinery exhibit. Perhaps the Solons of the Pine Tree State were fearful that the power-plant folks might become intoxicated with joy. Evidently, anything with a "smile" flavor is frowned upon in Maine.

On the "Baltic's" second day out from England a crazed stoker committed suicide by jumping overboard. Because a wealthy passenger having bowels of compassion got up a $500 purse for the widow, the daily press raised a great howdy-do over it, saying he "spent more time down in the boiler room than he did with the other passengers." Listen, Agnes! It must be turrible to be a millionaire an' have them reporter guys a-doggin' your every footstep.

Flow of Water through Check Valves

Result of tests to determine the flow of water through check valves under different pressures, and with several sizes of disks and seats.

During a recent discussion upon what takes place while water is passing through a check valve and allied subjects before the Modern Science Club of Brooklyn, some of the work of the committee on tests was demonstrated by James A. Donnelly. The committee had undertaken the investigation of the flow of water through steam traps, having various sizes of seats, as well as through standard check valves, ball check valves, etc. It was the committee's idea that no commercial test of an appliance should be made with but a very short pipe connection upon it, and as all are used with moderate lengths of pipe connected, it would seem as if this might be a reasonable assumption.

The first appliance tested was a standard, direct-lift ¾-in. check valve. The valve disk was first removed and the flow through the orifice and 1 ft. of pipe was obtained to determine the maximum amount of water which could be discharged, or what might be termed 100 per cent. efficiency. The valve disk was then placed in the valve and the test again run. In the same way, two tests were run with 25 ft. of pipe upon the outlet of the valve. The diagram, Fig. 1, gives a good idea of the cutting down of the discharge with this length of pipe. The two upper curves indicate the amount of water discharged through a check valve and 1 ft. of outlet pipe with and without the valve disk.

A comparison of the discharge of water through a ¾-in. check valve with and without the disk and discharging through a 25-ft. length of ¾-in. pipe is also indicated by the curves. At 30 lb. pressure the discharge through the diskless valve was 55.5 lb. of water per minute; with the disk in the valve, 53.5 lb. per minute was discharged. The difference in the amount of water discharging through 1 ft. and 25 ft. of piping at 20 lb. initial pressure is 42 and 45 lb. as against 58¼ and 59¾ lb. respectively. This indicates the result of pipe friction.

The tests with the ¾-in. ball check valve were made with 1-ft. and 25-ft. lengths of pipe attached to the outlet side, both open to the atmosphere. The discharge of water through the long pipe at 40 lb. pressure was 41 lb. per minute; with the short pipe the discharge was 47 lb. of water per minute.

The lower curve indicates the flow of water through a ¾-in. valve having a seat 11/32 in. in diameter and with a 1¼-in. flat disk covering the seat as shown in the illustration. At 40 lb. pressure 8 lb. of water per minute was discharged through the outlet. With the disk removed, the discharge was 45 lb. per minute, as indicated by the 11/32-in. orifice curve.

A restricting device of this character was first used for modifying the flow of gas to a burner as early as 1870. An attempt was made to use this principle about 1895 in a reducing valve for water, steam or other fluids. The principle of its operation is the transforming of pressure head existing at the inlet of the valve into velocity head between the disk and seat. The various pressures existing in the valve were tested and calculated as follows:

When 50-lb. gage pressure was used below the seat, a gage pressure of about 1 lb. was registered above the disk. As the disk at this time is held out of contact with the seat, it is obvious that it must be maintained in that position by an equalization of forces or pressures acting in opposite directions.

The area of the 1¼-in. disk, 1.227 sq.in., multiplied by the absolute pressure, 16 lb., gives a total pressure tending to force the disk toward the seat of 19.63 lb. The area of the seat opening, 11/32 in. in diameter, is 0.0928 sq.in. This, multiplied by the absolute pressure below the seat opening of 65 lb., gives a pressure tending to hold the disk up of 6.03 lb. This, subtracted from the total pressure above the disk, gives a force of 13.6 lb. as being present between the disk and seat and tending to raise it. If this is divided by the difference in area of the seat opening and disk, 1.1342 sq.in., it gives an average pressure between the disk and seat of 12 lb. absolute or about 5.5 in. of vacuum. See Fig. 3.

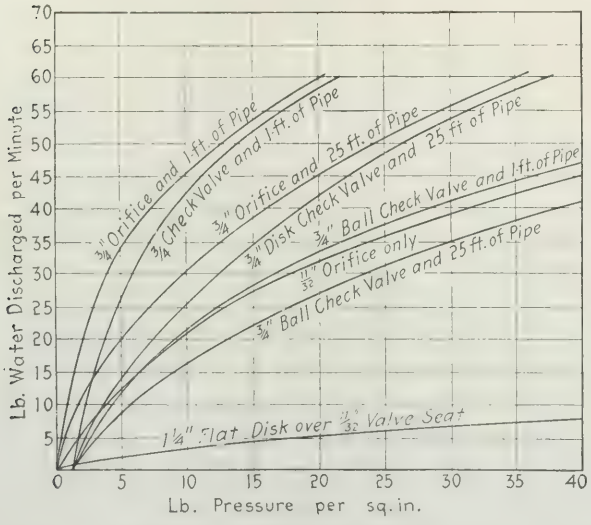

Lb. Water Discharged per Minute
Lb. Pressure per sq.in.
3/4" Orifice and 1 ft. of Pipe
3/4" Check Valve and 1 ft. of Pipe
3/4" Orifice and 25 ft. of Pipe
3/4" Disk Check Valve and 1 ft. of Pipe
3/4" Disk Check Valve and 25 ft. of Pipe
3/4" Ball Check Valve and 1 ft. of Pipe
11/32" Orifice only
3/4" Ball Check Valve and 25 ft. of Pipe
1 1/4" Flat Disk over 11/32 Valve Seat

FIG. 1. DISCHARGE CURVES WITH TWO LENGTHS OF PIPES

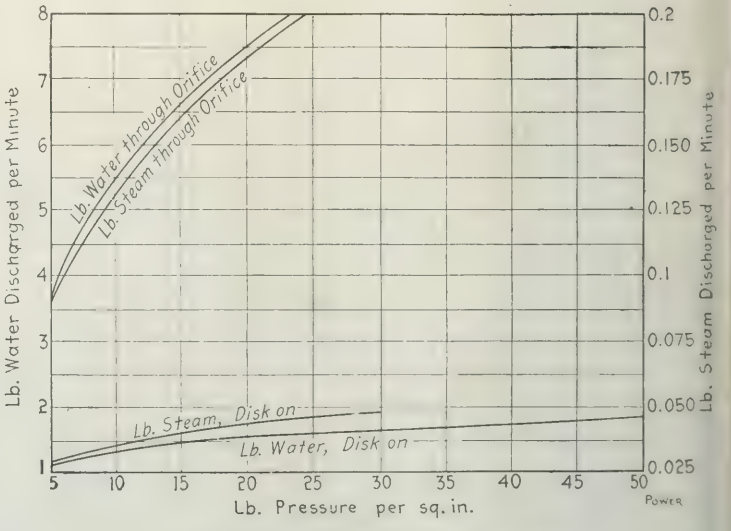


FIG. 2. STEAM AND WATER CURVES

The test of a flat disk steam trap having a ¾-in. connection, seat 3/8 in. in diameter, and a disk 3/8 in. in diameter, both for steam and water, gave the results shown in the curve of chart Fig. 2.

The capacity of the trap for discharging water was first tested by removing the disk. The disk was then placed on the valve and the same test again run. With the disk in place, the percentage of water passed as compared with that of the orifice without the disk was 31 per cent. at 5 lb., 20 per cent. at 20 lb., and 17 per cent. at 60 lb.

The trap was then tested in the same manner with steam and the result plotted

by using a scale for pounds of steam discharged per minute where the quantities are $^1/_{40}$ that of the scale for the discharge of water. This was done that a comparison might be made of the relative discharge with steam and water.

With the disk in place, the percentage of steam passed as compared with the orifice without the disk was 33½ per

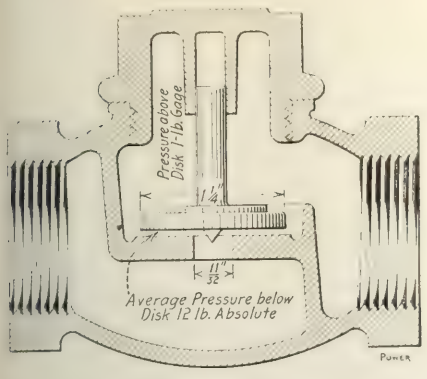


Fig. 3. Section of Flat Disk Test Valve

cent. at 5 lb. and 25 per cent. at 20 lb. pressure. This result shows that the device is more efficient for restricting the passage of water than that of steam. The reason is probably that steam or any fluid which changes in both volume and temperature does not give as high relative velocity for its pressure head as a fluid which, like water, does not change in volume and temperature when flowing through an orifice.

Several tests made upon ball check valves showed that their restriction of

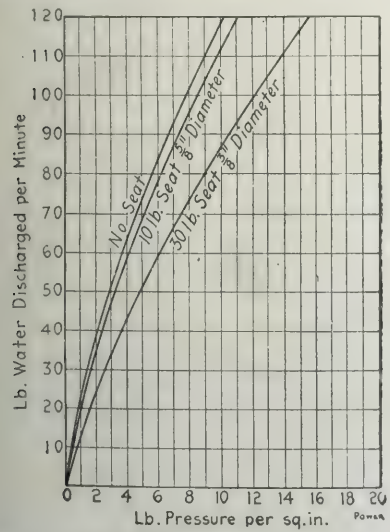


Fig. 4. Discharge of Water through a Steam Trap

the passage of water was practically the same as the restriction of the wide, flat disk check valve, last described.

Fig. 4 represented the discharge of water through a ¾-in. steam trap attached to 25 ft. of ¾-in. piping containing eight elbows. At 3 lb. initial pressure the discharge of water through a seat designed to operate at 10 lb. pressure was 47 lb. of water. At 10 lb. initial pressure the discharge was 87 lb. of water, using a 30-lb. seat. The upper curve indicates the flow of water through the same trap with no seat in the discharge valve.

During the test of the ball check valve a severe chattering was set up which varied in intensity with the initial water pressure. Tapping an air cock in the top of the valve cap and releasing the air stopped the noise. With the ball chattering, the discharge through the valve at a pressure of 25 lb. initial pressure was 45.4 lb. of water per minute. With the air removed from the valve cap the discharge, with the same pressure, was about one-half of the flow with the air present. Why the difference in the amount of water discharged?

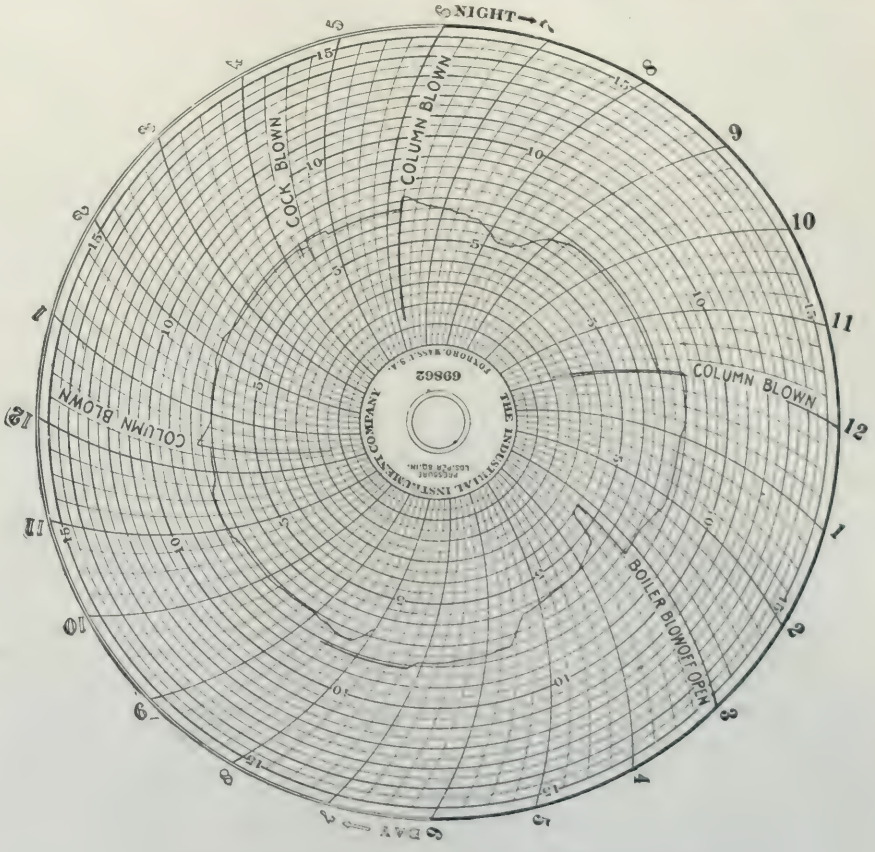
Recording Gage for Boiler Water Level

A pressure-recording gage has ordinarily the pressure of the atmosphere on the outside of the diaphragm or bourdon tube used to measure the pressure, so that the indication of the gage is that of the difference in pressure between the pressure inside the tube and that of the atmosphere. By surrounding the tube with a chamber so that pressure from any source may be maintained on its outside while pressure from another source is admitted to the inside, the gage will indicate the pressure difference.

Such a gage is called a differential-pressure recording gage and is used for various purposes, such as recording the differences in pressure on the up-stream and down-stream sides of calibrated nozzles and venturi tubes, on the static and dynamic orifices of Pitot tubes, etc.

This arrangement offers a solution of the difficulty of making a water-level recorder for steam boilers. The difference in pressure due to the small change in hydrostatic head between the high- and low-water levels is so small in proportion to the steam pressure that any change on a scale which represented the total pressure would be insignificant. By making the steam pressure act upon the outside of the tube and the steam pressure plus the head of the water column on the inside, the whole range of the gage becomes available for the varying pressure due to the varying water level.

The accompanying diagram is a reproduction of a record received from the Industrial Instrument Co., of Foxboro, Mass., made by one of their differential gages used as a water-level recorder on a 125-hp. horizontal return-tubular boiler in a small electric-light plant operating



Recording-gage Chart for Boiler Water Level

24 hours per day and carrying 90 lb. pressure. The diagram shows, not only the height of water in the glass at any moment of the day or night, but just when the column was blown down and the boiler blown off.

Steam Power Plant Piping Materials

By Wm. F. Fischer

An explanation of the materials used in steam power plant piping and the strengths and weights of the various thicknesses.

Directions are given for determining the proper weight pipe to use under given conditions.

Most of the pipe used in modern steam power plants is known as wrought-steel pipe, claimed by many as more reliable than wrought iron, especially where used to convey superheated steam at high pressure, or where used in expansion bends subject to severe bending strains, vibration, etc., and because mild steel has a higher tensile strength than wrought iron.

Although the tensile strength of new steel pipe exceeds that of wrought-iron pipe, after several years of service, the latter shows a higher tensile strength due to a greater resistance to corrosion and the repeated shocks of vibration than steel pipe.

Expansion and Contraction

For ordinary power-plant purposes the factor of safety of standard weight pipe is so large that the difference between the ultimate tensile strength of the two metals may be ignored as far as the bursting pressure is concerned.

Welded steel pipe has an average ultimate tensile strength of about 50,000 lb. per square inch. Wrought-iron pipe has an average ultimate tensile strength of about 40,000 lb. per square inch.

Wrought-iron pipe costs about 10 per cent. more than steel pipe, and is to be recommended where the pipe is exposed to the weather, buried underground, or where excessive corrosion might be expected, as in drain pipes, feed pipes, blowoff pipes, and those which are not to be kept constantly in service.

Manufacturers Standards

When ordering commercial wrought-iron pipe mild steel pipe will be furnished as a rule. If wrought-iron pipe is required, the specifications should distinctly state it. Mild-steel and wrought-iron pipe are graded as merchant or standard-weight, full-weight, extra-strong and double extra-strong pipe. Full-weight pipe will not vary more than 5 per cent. above or below the weight specified in the manufacturer's catalog.

Merchant or standard weight may vary as much as 6, 8 or even 10 per cent. less than card weight and the thickness will vary accordingly.

Table 1 gives the manufacturer's standard dimensions and card weights of mild-steel and wrought-iron pipe. The outside diameter remains the same for standard, extra-strong and double extra-strong pipe, so that any grade of pipe of the same size may be threaded into any grade of fitting, flange, valve or coupling of the same size.

Columns 3, 4 and 5 give the thickness of the metal in the different grades. The inside diameter of the pipe varies according to the thickness.

Columns 6, 7 and 8 give the nominal weight per foot of length. Unless otherwise ordered, manufacturers will ship standard and full-weight pipe with threaded ends and one wrought-iron coupling to each full length; extra-strong and double extra-strong pipe will be shipped with plain ends, unless otherwise specified.

Generally pipe up to 3-in. size is butt-welded and the larger sizes lap-welded. The standard and full-weight pipe is tested to from 600 lb. in the small sizes to as high as 1500 lb. per square inch in the larger sizes, and extra-strong pipe is tested to considerably higher pressures, depending upon the size. When tested to destruction by the Crane Co., a piece of 10-in. standard-weight wrought-iron pipe burst at 1900 lb. pressure per square inch. A 10-in. standard-weight mild-steel pipe burst at 3000 lb., a 10-in. extra-strong wrought-iron pipe burst at 2700 lb., a 2-in. standard-weight mild-steel pipe burst at 5800 lb., and a 2-in. standard-weight wrought-iron pipe burst at 4500 lb. pressure. None of this pipe burst at the weld, but some distance from it. An 8-in. standard wrought-iron pipe was tested to 2800 lb. without bursting, and an 8-in. standard wrought- or mold-steel pipe was tested to 3200 lb.

From the foregoing there seems to be no reason why pipe heavier than merchant or standard weight should be used for ordinary purposes and pressures up to 200 lb. per square inch. For pressures above this and up to 250 lb. per square inch full-weight pipe should be used. Extra-strong pipe is used for higher pressures, or for lower pressures also, where the pipe is subject to severe shocks, heavy vibration or excessive corrosion. Double extra-strong pipe is approximately twice as heavy as extra-strong pipe and is used mostly on hydraulic work, step-bearing oiling systems, etc., and is frequently subjected to pressures as high as 4000 lb. per square inch. For making up pipe bends no pipe lighter than full-weight should be used.

Large O. D. Pipe

All steel and wrought-iron pipes above 12 in. are specified by their outside diameters and known as O. D. pipe. Thus, a 14-in. pipe 3/8 in. thick will be 14 in. outside diameter and 14 — 3/4 = 13 1/4 in. inside diameter. Large O. D. pipe varies in thickness from 1/4 in. to 3/4 in., depending upon size and service for which it is required.

The terms "merchant pipe," or "standard pipe" should never be applied when ordering pipe larger than 12 in., but the outside diameter and thickness should always be stated in the specifications. When the thickness is not specified manufacturers will generally supply pipe 5/16 or 3/8 in. thick in sizes from 14 to 16 in. inclusively, and 3/8 in. thick in the larger sizes. For threaded or screwed work 1/4-in. metal is too light; 5/16 in. is considered a minimum thickness and 3/8-in. thickness is preferable for good results.

TABLE 1. STANDARD DIMENSIONS OF WROUGHT IRON AND STEEL PIPE, FROM 1 INCH TO 12 INCH INCLUSIVE

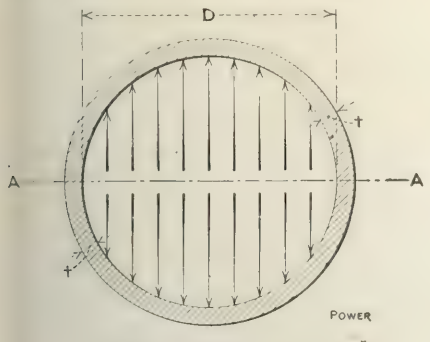
1	2	3	4	5	6	7	8
		Thickness in Inches			Nominal Weight per Foot		
Pipe Size	Outside Diameter Inches	Standard Weight	Extra Strong	Double Extra Strong	Standard Weight	Extra Strong	Double Extra Strong
1	1.315	0.134	0.182	0.364	1.668	2.17	3.65
1 1/4	1.660	0.140	0.194	0.388	2.244	3.00	5.20
1 1/2	1.900	0.145	0.203	0.406	2.678	3.63	6.40
2	2.375	0.154	0.221	0.442	3.609	5.02	9.02
2 1/2	2.875	0.204	0.280	0.560	5.739	7.67	13.68
3	3.500	0.217	0.304	0.608	7.536	10.25	18.56
*3 1/2	4.000	0.226	0.321	0.642	9.001	12.47	22.75
4	4.500	0.237	0.341	0.682	10.665	14.97	27.48
4 1/2	5.000	0.246	0.360	0.718	12.34	18.22	32.53
5	5.563	0.259	0.375	0.750	14.502	20.54	38.12
6	6.625	0.280	0.437	0.875	18.762	28.58	53.11
*7	7.625	0.301	0.500	0.875	23.271	37.67	62.38
8	8.625	0.322	0.500	0.875	28.177	43.00	71.62
*9	9.625	0.344	0.500	...	33.701	48.25	...
10	10.750	0.366	0.500	...	40.065	54.25	...
12	12.750	0.375	0.500	....	48.985	65.00	...

*Note: 3 1/2-7 and 9 in. pipe are odd sizes not so frequently used for general power plant work as the other sizes. For more complete tables of dimensions see manufacturers catalogs.

The standard thickness and weights of large outside-diameter pipe are given for reference in Table 2.

Bursting Pressure and Factor of Safety

The following rules and examples show the method of calculating the bursting pressure and safe working pressure for pipes of known internal diameter and thickness. A pipe or cylinder, when subjected to internal pressure, is strained equally in all its parts and if rupture occurs it is always in the direction of the length.



Cross-section of Pipe

The illustration is a cross-section of a piece of pipe of unit length. The arrows show the direction of the pressure tending to burst the pipe along the line *AA*, or in the direction of the length. The formula

$$P = \frac{2ts}{D}$$

gives the absolute safe static pressure which may be put upon a pipe in which

P = Internal fluid pressure in pounds per square inch;
D = Inside diameter of pipe in inches;
L = Length of pipe in inches;
t = Thickness of metal in inches;
S = Tensile strength of pipe metal in pounds per square inch.

The total bursting pressure, or the pressure tending to burst the pipe and tear the plates apart along line *AA* equals

$$PDL$$

This bursting pressure is resisted by the stress in the ring of metal composing the pipe, which is

$$2\,tLS$$

Therefore,

$$PDL = 2\,tLS$$

or canceling L from each term of the equation we get

$$PD = 2\,tS$$

and transposing, we get,

$$P = \frac{2ts}{D} \qquad (1)$$

and

$$t = \frac{PD}{2S} \qquad (2)$$

To find the internal pressure necessary to burst a piece of 8-in. standard-weight mild-steel pipe, assuming the ultimate tensile strength of the metal to be 50,000 lb. per square inch we find from Table 1, column 2, that an 8-in. standard pipe is 8.625 in. outside diameter and from column 3 is 0.322 in. thick. The inside diameter is, therefore,
8.625 — (2 × 0.322) = 7.98, *say* 8 *in.* Then substituting in formula 1,

$$P = \frac{2 \times 0.322 \times 50{,}000}{8} = 4025\ lb.\ per\ sq.in.$$

Assuming this pipe to be subject to only 200 lb. internal pressure per square inch the factor of safety would be

$$\frac{4025}{200} = 20.12$$

or $^1/_{20}$ of the bursting pressure.

The thickness of metal required in pipe of 8 in. inside diameter to burst under an internal pressure of 4025 lb. per square inch, assuming the ultimate tensile strength of the pipe material to be 50,000 lb. per square inch, is found by substituting in fomrula 2

$$t = \frac{4025 \times 8}{2 \times 50{,}000} = 0.322\ in.\ or\ \tfrac{5}{16}\ in.$$

To find the unit fiber stress of the pipe material in pounds per square inch, due to the internal fluid pressure; or, in other words, to find to what extent the metal is strained, use the formula

$$f = \frac{PD}{2t}$$

where

f = Unit fiber stress in pipe material in pounds per square inch;
P = Internal fluid pressure in pounds per square inch;
D = Inside diameter of pipe in inches;
t = Thickness of pipe metal in inches.

To find the unit fiber stress in the metal of an 8-in. standard mild-steel pipe of 8 in. inside diameter and 0.322 in. thick, considering the pipe to be subjected to an internal pressure of 200 lb. per square inch, substituting in the formula

$$f = \frac{200 \times 8}{2 \times 0.322} = 2485\ lb.\ per\ square\ inch$$

and

$$\frac{50{,}000}{2485} = 20,\ the\ factor\ of\ safety$$

An average tensile strength of 50,000 lb. for mild-steel or wrought-steel, and 40,000 lb. for wrought-iron pipe is the average for pipe material as adopted by pipe manufacturers.

For steam piping the factor of safety employed should never be less than 6 as there are stresses, due to expansion and contraction, vibration, shocks, due to water-hammer, etc., and there is also the corrosive action of the metal to be taken into account, as corrosion gradually decreases the strength of the pipe metal.

For feed-water mains and similar piping subjected to excessive corrosion, the factor of safety should never be less than 8 and from 10 to 15, or more is desirable. For feed mains, water mains and all pipe subjected to excessive corrosion or pitting, galvanized wrought-iron pipe is preferable to plain black pipe. For all ordinary pressures as carried in the average run of power work the pressure necessary to burst or rupture merchant or standard-weight pipe is so far in excess of the ordinary working pressure that no calculation for strength is necessary.

For hydraulic work, however, where the pressures to be carried are considerably higher than in the average power plant, it is advisable to determine by formula 3 the unit fiber stress of the pipe material, due to the working pressure, and from that find the factor of safety for the given conditions, or the bursting pressure may be calculated from formula 1, and this bursting pressure divided by the working pressure will give the factor of safety for the given conditions

TABLE 2. STANDARD DIMENSIONS AND WEIGHTS OF LARGE O. D. WROUGHT PIPE

Outside Diameter of Pipe Inches	Weight per Foot of Length—Pounds — Thickness Metal in Inches							
	$\frac{1}{4}''$	$\frac{5}{16}''$	$\frac{3}{8}''$	$\frac{7}{16}''$	$\frac{1}{2}''$	$\frac{9}{16}''$	$\frac{5}{8}''$	$\frac{3}{4}''$
14	36.75	45.72	54.61	63.42	72.16	80.80	89.36	106.20
15	39.42	49.06	58.62	68.10	77.50	86.81	96.03	114.20
16	42.09	52.40	62.63	72.78	82.85	92.83	102.70	122.20
17	44.76	55.74	66.64	77.46	88.19	98.84	109.40	130.30
18	47.44	59.08	70.65	82.14	93.54	104.80	116.10	138.30
20	52.78	65.76	78.67	91.49	104.20	116.90	129.40	154.30
21	. .	69.10	82.68	96.17	109.60	122.90	136.10	162.30
22	. .	72.44	86.68	100.80	114.90	128.90	142.80	170.30
24	. .	. .	94.70	110.20	125.60	140.90	156.20	186.30
26	. .	. .	102.70	119.50	136.30	152.90	169.50	202.40
28	. .	. .	110.70	128.90	147.00	165.00	182.90	218.40
30	. .	. .	. .	138.20	157.70	177.00	196.30	234.40

Note: For threaded ends use nothing less than $\frac{5}{16}$-in. thick. It is not advisable to thread pipe above 20 in. O. D.

The bursting of a water turbine on Sept. 13 at Station No. 1 of the Shawinigan Water & Power Co., at Shawinigan Falls, Que., resulted in drowning an employee and flooding the station.

International Union of Steam Engineers Convention

The third biennial convention of the International Union of Steam Engineers was held at the Hotel Ryan, St. Paul, Minn., on Sept. 9 to 14. The first session opened at 10 a.m., Monday, Sept. 9, when V. E. Patnaude, business agent for Local No. 84, St. Paul, introduced Mayor H. P. Keller as the first speaker. The mayor declared that the people of St. Paul were ardent believers in the value of organization.

H. E. Hall, president of the Minnesota State Federation of Labor, spoke briefly of the work of the federation and made a plea for greater coöperation by the rank and file of organized labor. A. L. Rich, president of the Trades and Labor Assembly, of St. Paul, and Collis Lovely, general vice-president of the Boot and Shoe Workers' Union, made brief speeches.

General President Matt Comerford, after a short speech of thanks to the speakers, formally opened the convention and appointed the committee on law. After a partial report of the committee on credentials was adopted, the convention adjourned at 12 noon.

At the afternoon session, President Comerford appointed committees on rules, officers' reports, resolutions, grievance, insurance and on charters. First Vice-President Milton Snellings presided while President Comerford made his official report. The figures for the organizing work done during the past two years showed that the net gain in membership was 4432. During the past term much attention was paid to the Pacific Coast, where an organizer has been in the field almost continuously. The most gratifying results were obtained in Michigan, where one of the general organizers has been at work for more than a year.

In the discussion of legislation for examining and licensing engineers, a system of uniform license laws was urged as beneficial to both the engineer and the general public. The recommendation that firemen and oilers be admitted to the union as apprentice engineers on the principle adopted by the American labor movement of "one trade, one organization," was adopted. The general secretary-treasurer was allowed to have his report printed as the remaining time would not permit its being read. This report showed the organization to be in sound financial condition, having some $31,650 on hand July 31, 1912. During the past two years 85 new charters were issued, 6 locals reorganized, 25 local charters lapsed or dropped and 2 charters revoked.

Other business sessions were held on Tuesday morning and afternoon, Wednesday morning and on Thursday, Friday and Saturday mornings and afternoons. The bulk of the business transacted was routine or local. A matter of general interest was changing the name, the "International Union of Steam Engineers," to "International Union of Steam and Operating Engineers."

The following officers were elected for the present term: Matt Comerford, general president; Milton Snellings, first vice-president; T. J. Roberts, second vice-president; M. Kelly, third vice-president; M. Murphy, fourth vice-president; W. M. Finlay, fifth vice-president; James G. Hannahan, general secretary-treasurer; trustees, Louis Thurber, chairman; M. J. Cassidy, P. J. Horan.

Peoria, Ill., was selected as the next convention city.

Entertainment Program

The elaborate entertainment program provided reflected much credit upon the committee. The ladies' committee of the "Twin Cities" gave an informal reception to the visiting ladies in the hotel parlors on Monday afternoon. On Tuesday afternoon the ladies attended a party at the Empress Theater. On Wednesday afternoon the delegates and guests enjoyed an automobile ride, visiting historic Old Fort Snelling, the Soldiers' Home, Minnehaha Falls, and points of interest in Minneapolis, returning by way of the State Agricultural School, State Fair Grounds, Como Park and the State Capitol. An unfortunate accident marred the complete enjoyment of this trip. In trying to avoid colliding with another automobile, the driver of a delegates' car ran into a sidewalk in Minneapolis. The machine struck a tree and painfully injured M. Mangan, of Chicago. Mr. Mangan was forced to remain several days in the hospital because of a rather serious injury to his neck.

On Thursday evening the delegates and guests attended a ball and banquet at Minnehaha Hall. On Friday, Mrs. J. J. Hannahan entertained the lady guests at her home on West Lynhurst Ave. The ladies were given an automobile ride about the "Twin Cities" on Saturday, and a tea party at the Emporium tea rooms.

Exhibits

The convention committee had invited T. S. F. Hayes, St. Paul representative of the Dearborn Drug & Chemical Works, to arrange for an exhibit in connection with the present convention. After consulting with William P. Lyons, of the Lyons Boiler Works, and F. J. Camitsch, of the Western Supply Co., St. Paul, space was engaged in the Ryan Hotel corridor, and on Aug. 12 invitations to exhibit and have a representative on the ground were extended to about 200 manufacturers and supply houses. Of this number, 31 engaged space and either sent a representative or arranged for a display. The short time allowed prevented a larger number of exhibitors. As it was, the exhibitors' committee was pleased with the results obtained. The following houses had exhibits and representatives in attendance:

Dearborn Drug & Chemical Works, represented by F. F. F. Hayes, E. T. Ward, A. S. Cook; Western Supply Co., F. J. Camitsch; Quaker City Rubber Co., Charles M. Pinchon and J. B. Freye; Lyons Boiler Works, W. P. Lyons; St. Paul Western Coal Co., H. O. Brown and Charles Wanemaker; POWER, A. R. Maujer; Viscosity Oil Co., W. L. Beardsley; R. B. Whitacre & Co., A. E. Elliott; Platt Iron Works, J. F. Gould; Jenkins

Delegates and Visitors at the St. Paul Convention, I. U. S. E.

Brothers, J. C. McHugh; North Western Electric Equipment Co., Neal Snyder; U. S. Graphite Co., Charles H. Schenk; W. S. Nott Co., J. H. Vailes; Crane & Ordway, C. L. Olson; Westinghouse Air Brake Co., W. E. Lean; Hawks Price Co., Ben Hawks; Osborne Valve Co., A. H. Breach; St. Paul Electric Co., Charles Horton; Valley Iron Works, Mr. Camron; Rex Oil Co., Paul Swanson; American Huhn Metallic Packing Co., J. J. Schwends; George M. Kenyon, C. L. Smith; Sim D. Rollins & Co., Sim D. Rollins; The Hill Pump Valve Co., William F. Birch; Robinson, Cary & Sands Co., Charles Peters; Lunkenheimer Co., H. A. Burdorf; Plant Rubber Co., Charles Faucett; The Sutherland-Carson Land Co., A. W. Sutherland.

The exhibitors' committee consisted of F. J. Camitsch, chairman; T. S. F. Hayes, secretary-treasurer, and W. P. Lyons.

Employees' Safety Rules

The Minneapolis General Electric Co. recently issued to its employees a carefully prepared booklet of safety rules, including copies of the state labor laws and suggestions by the state commissioner of labor. Each book is numbered and the recipient is held responsible for its safe keeping as well as for familiarity with its provisions.

The first page is a perforated leaf carrying a blank statement that the employee has read and thoroughly understands the rules. Within 10 days after receiving his copy the employee must sign and return this sheet to the office. Besides general operating rules, there are instructions for the use of special safety devices, descriptions of methods of resuscitation from electric shock, etc.

This book is but one of a number of measures to conserve employees' safety recently carried out in Minneapolis. By a recent city ordinance there, all poles carrying 2300 volts or more must be marked with a 3-in. ring of bright red paint just below the lowest crossarm. Each ring must be kept bright.

Refrigeration Exposition for 1913

The Third International Congress of Refrigeration, to be held in Chicago, September, 1913, was brought to this country through the invitation by the President of the United States under authority of the joint resolution of Congress and extended to the delegates in attendance at the meeting held in Vienna in 1910. The wonderful progress made in refrigeration methods should make this the most important congress yet held.

The organization of an association to manage this exposition is now under way and as soon as incorporated, the work will be commenced. The men behind the enterprise have devoted time and energy to securing this congress, and urge that the industries participating in the cold-storage division make all preparations now.

The notice may be accepted as official that the show will be held, and as rapidly as possible the allied interests through their associations and individual organiations will receive communications direct as to its extent and scope.

NEW PUBLICATION

MODERN ILLUMINATION. By Hortsmann and Tousley. Published by Frederick J. Drake & Co., Chicago, 1912. Size, 7x4½ in ; 265 pages ,illustrated. Price, $2, net.

This book, by two well known authors on electrical subjects, has been prepared as a ready reference for architects, managers, electricians and such as have occasion to deal with illumination problems in general. Hence only such theory is included as will be readily comprehended by the practical man.

The text is a compilation in logical order of much material which has heretofore been widely scattered and includes chapters on light, the principles of vision, photometry, characteristics of electric illuminants, reflectors, choice of lamps and indirect lighting. A chapter upon "practical considerations" contains many useful suggestions as to the lighting schemes for various types of buildings and the different rooms therein.

Numerous tables are supplied and enough problems are solved to make the text easily comprehensible.

SOCIETY NOTES

Smoke abatement will be the subject of the monthly meeting of the American Society of Mechanical Engineers in New York, at the Engineering Societies Building, 29 West Thirty-ninth St., Tuesday evening, Oct. 8.

The discussion will be opened by George H. Perkins, Lowell, Mass., with a report on the International Smoke Abatement Exhibition and Conference held in London last March and April. Mr. Perkins was the representative of the Lowell Textile School at this conference and informally of the society also.

Authorities from other cities where the problem has been seriously attacked, and engineers and manufacturers generally will take part in the discussion. The meeting is open to the public and all in any way concerned with the smokeless combustion of coal are invited to attend.

From the standpoint of residents generally the importance of reducing the smoke nuisance is obvious, but it is hoped that this meeting, which will be one of the most important on this subject ever held, may bring out a fund of data in substantiation of the contention that reduced smoke can and should mean greater operating economy.

The Central States Water Works Association held its sixteenth annual convention at Detroit, Mich., Sept. 24 to 26.

No. 3 Association, N. A. S. E., Newark, N. J., has moved into its new quarters at 101-103 Market St. The association announces that it will be pleased to receive catalogs, literature, models, etc., of power-plant equipment. All communications should be addressed to C. L. Johnson, 42 Rowland St., Newark, N. J.

The Fourth National Conservation Congress will be in session at Indianapolis, Ind., on Oct. 1 to 4. The vital resources of the country will be discussed in their many important divisions by men prominent in the field. The National Association of State Conservation Commissioners will also meet in special session during the congress.

The following members were appointed delegates to represent the American Society of Heating and Ventilating Engineers at the Fourth National Conservation Congress at Indianapolis, Ind., on Oct. 1 to 4: Samuel R. Lewis, N. L. Patterson and J. H. Davis, Chicago, Ill.; Theodore Weinshank and Joseph G. Hayes, Indianapolis, Ind.

PERSONAL

A. G. White, electrical engineer of the Tri-State Railway & Electric Co., with headquarters at East Liverpool, Ohio, has resigned. It is understood that Mr. White and a number of business men of Toledo, Ohio, will engage in the manufacture of cement in Chicago.

Peter Junkersfeld, of the Commonwealth Edison Co., Chicago, is in Europe. While in England, Mr. Junkersfeld will visit the Heaton works of C. A. Parsons & Co., where that concern is building the new 25,000-kw. turbo-generator for the Fisk St. Station in Chicago.

H. G. D. Nutting, for several years manager and superintendent of the municipal light and water plant at Fort Atkinson, Wis., recently became associated with the Central Illinois Public Service as district superintendent. Mr. Nutting will be succeeded by W. D. Leonard at the Fort Atkinson plant.

Hans C. Specht, for several years electrical engineer with the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Penn., will soon leave America for an indeterminate stay in Germany. While there, Mr. Specht will be associated as chief engineer of construction work with the Norddeutsche Wollkämmerei & Kammgarn Spinnerei, a textile firm near Bremen. Mr. Specht is a member of the American Institute of Electrical Engineers.

Moments with the Ad. Editor

Here's a question for you—

Suppose every advertiser in this paper "cut out" simultaneously and henceforth your copy of Power came to you without any advertising at all, what would be the result?

Would Power be as interesting to you as it is now?

Would it be as valuable?

If there is any engineer who can answer "yes" in these two instances, we would like to know how he spells his name.

We would like to know what new line of work he is contemplating taking up.

You'll admit that Power would not be as interesting or as valuable without its advertising pages.

Then you *are* getting some good out of Power advertising.

Do you realize, too, that whenever you read one of these advertisements you have given an audience to a sure-enough salesman? That is to say, the *properly written* ads are such.

And whenever this salesman persuades you to buy, do you know *exactly* what has transpired in your mind?

'Tis this—the mental law of sale.

Whether you buy a spool of packing, or a 1000 k. w. unit, or what not, it's there.

First this salesman got your *attention*. How? By a catchy headline, by a pertinent question, a direct command, or possibly by an attractive arrangement or layout.

Then it developed an *interest*. This interest properly augmented then changed into *desire* and the desire intensified developed at once into *action*, on your part.

If this salesman carries you successfully through these four steps it contains *all* the elements of good salesmanship.

It has succeeded in getting you to write for something for which a desire was created in your mind.

But that's not the end—The employer of this salesman has more to perform.

He must obtain and maintain your confidence in his goods.

He does this by rendering you a service through these goods.

If he can render you no material service he fails—and this salesman fails on the next visit.

And remember this—no one knows this truth any better than an advertiser in this paper.

No advertiser in Power sees any profit in selling you something without selling you satisfaction and confidence too.

The modern business slogan reads: "He profits most who serves best." It's universal.

Our advertisers are not so narrow or unbusinesslike that they fail to see beyond the passing of the voucher from your hands to theirs.

They know that only reliable products can be continuously advertised.

They know that they themselves must maintain an unquestionable reliability if they are to remain as an advertiser in this paper.

Now then, as you read the advertisements in this issue bear out these thoughts.

Notice the approach of these many salesmen.

Try yourself—and see how many of them get your attention—

How many of them get you interested—how many of them are able to create a desire in your mind to actually *want* their products—and how many of them persuade you to write for a catalog or free samples or further information, etc.

Look upon these advertisements as so many salesmen calling on *you*, and Power will be more interesting. It will be more valuable than you perhaps have realized before.

POWER

Vol. 36 NEW YORK, OCTOBER 8, 1912 No. 15

Your Son?

Give Him a Man's Job!

New Roseland Pumping Station

By Edward K. Hammond

The pumping engines are placed in a pit above a basin into which the lake water flows by gravity. The four pumps are each of twenty-five million gallons capacity.

An increasing demand for water throughout the southern part of its territory has made it necessary for the City of Chicago to add to its facilities for supplying the requirements of that district. This need has been met by the construction of the new Roseland pumping station, now operating under part of its full rated capacity. The equipment is the most modern and, as a power-plant installation, presents many interesting features.

The building is of pressed brick and stone, with a tiled roof. Its main section is divided by a wall; the boiler room occupies the south end of the building, and the engine room is north of it. The building faces north and the engineers' offices are situated at both sides of the entrance.

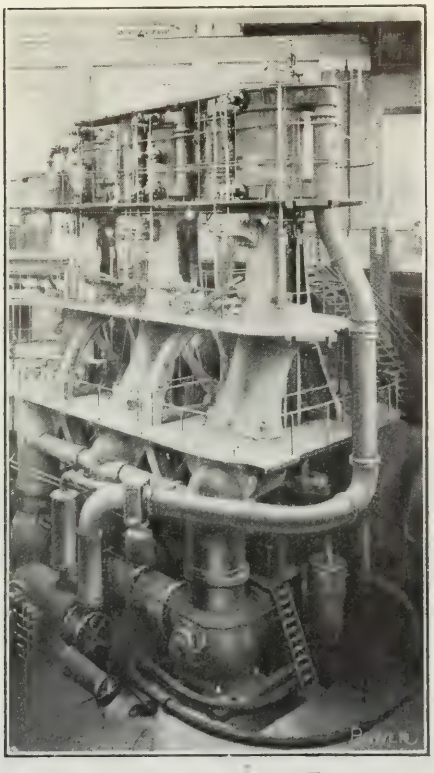


FIG. 1. WEST PUMPING ENGINE

PUMP ROOM

The pump-room floor is on the ground level, and the pumps are set in a circular pit in the center, Fig. 1, the walls and floor of which are concrete. A concrete basin is in the center of the engine pit, and at the top of a shaft at the end of the new southwest land and lake tunnel, which forms a connection through bedrock, between the station and the two cribs in Lake Michigan, from which the water is pumped. The depth of the engine pit is such that the level of the water in the basin is at Lake level, consequently the basin is filled by gravity.

The two pumping engines now in service have been installed in the southern half of the pit, leaving space for two similar units; arrangements have been made for installing a third engine, which will be of the same type as those already in use. These units consist of vertical 24, 48 and 70 by 60-in. triple-expansion Allis-Chalmers pumping engines, each having a capacity of 25,000,000 gal. of water per 24 hr., under a normal head of 140 ft., or 60.6 lb. per sq.in. The engines can pump against a maximum head of 200 ft., or 86.6 lb. per sq.in.; they are operated under 170 lb. steam pressure at the throttle.

The steam cylinders are steam jacketed and between the high- and intermediate-pressure and between the intermediate- and low-pressure cylinders are reheating receivers.

The basin from which the water is pumped furnishes the condensing water for the engines. Each exhaust pipe contains an oil separator and a primary heater between the engines and condensers. The surface condensers are designed to maintain a vacuum within 2 in. of barometric pressure. The condensed water goes to the hot well and is pumped back through the primary heater to the feed-water heater in the boiler room.

Each engine has a direct-driven air pump, feed pump and air compressor; there is also an independent air compressor and air pump, to provide the necessary compression and suction in the cylinders to start the engines. It has been found more convenient, however, to use water pressure for this purpose. Two pipes fitted with valves are connected to each water cylinder, which permit the water pressure to be turned on or released under any plunger, according to its position. The water pressure starts the pump when steam is turned on to take up the load.

FIG. 2. VIEW OF THE BOILER ROOM

ELECTRIC GENERATORS

All electric current is furnished by two Curtis turbines directly connected to direct-current generators of 75 kw. capacity. In summer, the exhaust from these turbines goes to the feed-water heater in the engine room, but in winter it will be used in the building heating system.

Oiling System

The automatic oiling system with its gravity tanks makes it unnecessary to handle any oil in the plant after it has been deposited in the tanks in the oil room. These tanks are connected with two supply tanks, on the south wall of the engine room near the floor; the latter are kept filled by gravity feed. The oil passes through filters on its way to the supply tanks. Oil is pumped from the supply tanks to a third pair of tanks near the ceiling on the north wall of the boiler room. Connections lead from the lower of these two tanks to all bearings, guides, etc., and from the upper tank to all of the engine cylinders in the station. The supply tanks hold 5 bbl. and the gravity tanks 3 bbl. of oil.

The used oil runs through a system of drip pipes and is collected in reservoirs,

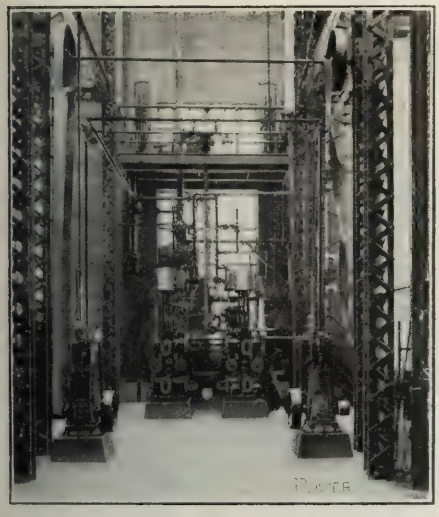


Fig. 3. Stoker Engines, Feed-water Heater and Feed Pump

from which it is pumped into oil filters on the wall of the engine room above the oil-supply tanks; the oil is pumped from the drip reservoirs into the filters and then drains to the supply tanks.

Boiler Room

The boiler room, Fig. 2, contains two batteries of two 300-hp. Edge Moor water-tube boilers, designed for 171 lb. steam pressure. The equipment is sufficient to take care of the four pumping units which will eventually be installed; only one battery is now used, but the third pump when installed, will require putting the second battery in service.

The boiler feed water is heated by the exhaust from all of the auxiliary engines in the plant.

A Link-Belt bucket conveyor runs over the boilers to deliver coal to the bunkers and down through the basement to take away the ashes. Coal is received at the plant on a spur from the C. & E. I. R.R. track, and is dumped into a hopper in the basement. After passing through a 40-ton crusher the coal is carried through the basement to the east end of the building, up through a sheet-metal casing surrounding the conveyor to the two double 170-ton bunkers over the boilers. One bunker feeds each boiler.

When the conveyor is to be used for removing ashes, the tripping mechanism is reversed. This throws out the trip over the coal bunkers and sets the one over the ash hopper at the west end of the

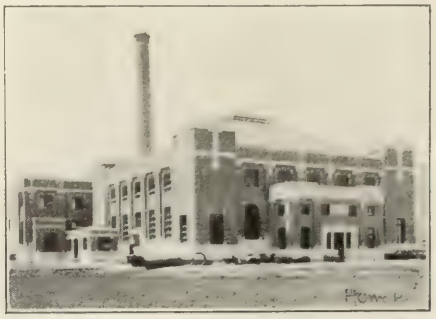


Fig. 4. New Roseland Pumping Station

building. The ashes are then taken up by the conveyor and carried around to this hopper, where they are dumped. The same siding over which coal is brought to the plant runs under this hopper, and the ashes are discharged into cars for removal, or they can be dropped through a second chute into wagons drawn up on a driveway beside the railroad siding. A system for wetting down the ashes before they are removed from under the grates has been installed.

The boilers are fitted with engine-driven chain-grate stokers, a single shaft extending over both batteries of boilers providing the drive for the eccentrics which actuate the rods connecting with the grates. Fig. 3 shows the arrangement of the engines which drive the stokers; also the feed-water heater and the feed-water pump.

The steam mains are piped from each boiler down through the floor near the north wall of the boiler room. Beneath the floor, these pipes make a bend through the wall into the south end of the engine room, where connection is made with a double header. All four boilers connect with each header, but the arrangement of valves is such that either header can be cut out when desired. The pumping engines take their steam from these headers through 6-in. pipes, which are trapped before reaching the engines.

All piping is coupled by steel flanges welded to the pipe. The method of arrangement has been particularly well worked out, and an admirable combination of efficiency and simplicity have been secured.

Shafting and Belting Calculations

By Miles Sampson

The accompanying charts represent a very simple graphical solution of the problems which arise concerning shafting and belting. While presenting nothing new in theory, the application will be found very useful.

Shafting

The chart for shafting is based upon the commonly used formula for the strength of shafting:

$$Horsepower = \frac{(diameter\ of\ shaft)^3 \times r.p.m.}{constant}$$

In explanation of the constant, it may be noted that 100 is used for head-ends of shafts receiving large amounts of power. By largely increasing the size of

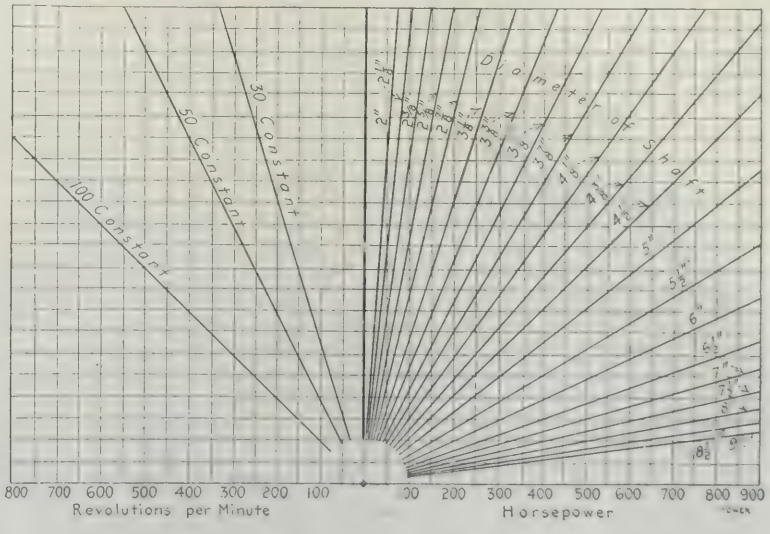


Fig. 1. Chart for Calculating Shafting

shaft necessary to carry a given power, it takes account of the large transverse stress due to the belt pull, and of the necessary stiffness. Any shaft carrying pulleys with belts over 6 to 8 in. wide should be figured using this constant. The constant 50 should be used to calculate countershafts driving ordinary machinery, and shafting carrying small belts. When

30 is used only the transmission of power should be expected of the shaft, no transverse stresses being considered.

The method of using the chart follows: On the scale representing the given speed follow vertically to the line denoting the proper constant, then across horizontally to the line representing the shafting size. The abscissa of this point on the horsepower scale gives the allowable load. Thus, at 300 r.p.m. a 3⅛-in. shaft carrying small belting (constant 50) would transmit 180 hp. Problems having different given conditions may be solved in a similar manner. For example—to carry 400 hp. at 400 r.p.m. a head-end shaft must be 4½ in. diameter.

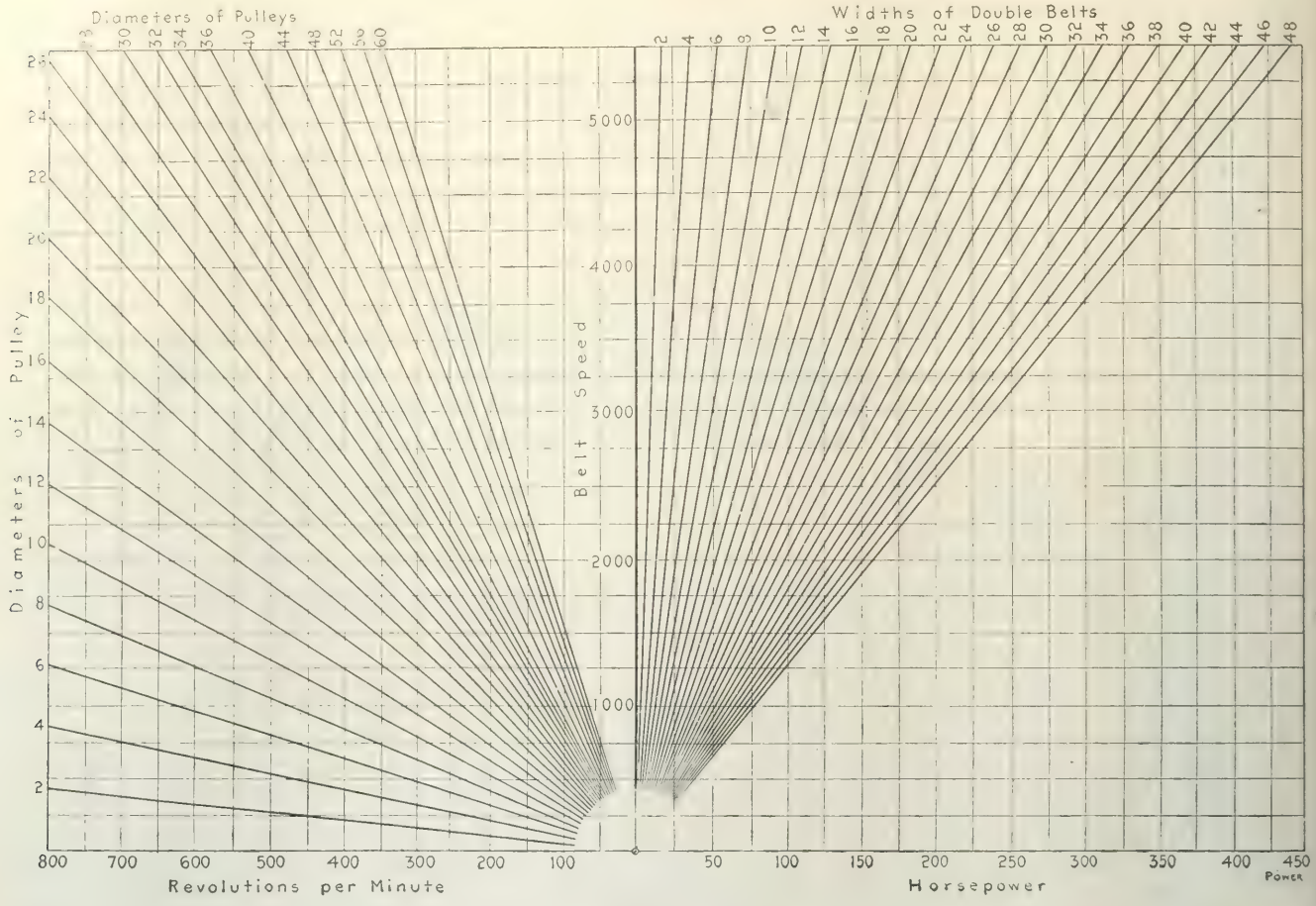


FIG. 2. CHART FOR DOUBLE BELTING

BELTING

The chart for belting is based upon a speed for double belts of 600 ft. per minute per horsepower per inch of width. This is conservative in the light of present practice, but represents a standard toward which belt users are tending. For very wide belts it may doubtless be decreased to 500 ft. per minute with perfect safety. For single belts the 600 becomes 1200, and for triple belts, 400.

For using the chart at any number of revolutions per minute follow vertically to the line representing the pulley diameter, which gives, on the vertical scale, the belt speed in feet per minute. Follow across horizontally to the belt width and down to the required horsepower. Thus at 400 r.p.m. on a 36-in. pulley (3750 ft. per minute) a 20-in. double belt will transmit 125 hp. Similarly, if a 30-in. belt is the maximum width available for transmitting 250 hp., a belt speed of 5000 ft. per minute is necessary; this requires a 60-in. pulley running 320 r.p.m.

Ever Ready Revolution Counter

The illustration shows a new type revolution counter particularly easy to handle, being shaped somewhat like a revolver

NEW REVOLUTION COUNTER

and about the same size. The starting and stopping of the counting mechanism are not dependent on the degree of pressure with which the spindle is held against a revolving shaft but solely upon the operation of a trigger, thus eliminating chance for the counting to begin too soon or too late.

The spindle is held against the shaft and revolves freely with it, but the counter remains at zero until the trigger is pulled. The counting continues as long as the trigger is held, but stops the instant it is released. The figures, which are located in the head of the counter, can easily be seen and are quickly set back to zero. It counts forward whether the shaft is turning right or left. This counter is a recent product of the American Ever Ready Co., 304 Hudson St., New York City.

The engineering work in connection with the Panama-Pacific International Exposition at San Francisco, Calif., is proceeding rapidly. A department of mechanical and electrical engineering is at present engaged in studies for the underground distribution system and wiring systems of buildings for electric power. Careful detail estimates of the amount of power required for a great variety of purposes are necessary. Studies are being made for the selection of designs and sites for the service and other buildings.

Calculations of Pump Slippage

By W. G. Kirchoffer

Results of tests for slippage on several large pumping engines and a convenient diagram for determining the discharge of a pump with a given slippage.

It is a common practice among superintendents of water-works and engineers in charge of pumping plants to compute the discharge of their pumps on the basis of the actual displacement of the pump plungers. Such computations often lead to the statements made in the annual reports of the department that vast quantities of water are being used or wasted which cannot be accounted for. As a matter of fact, few, if any, pumps in this service ever discharge as much water as the displacement of the plungers would indicate.

The slippage of pumps depends upon many factors and cannot well be determined except by actual tests. In some cases where the water is pumped from or into reservoirs of known capacity, it is a comparatively easy matter to determine the actual capacity of the pump.

Besides the satisfaction of knowing that the consumption is not as large as supposed, such tests for slippage are of value in showing to the engineer the true condition of the pump. If such tests are undertaken, they should be made under practical operating conditions, for only then can the true slippage be determined. For example, suppose a pump is normally working against a pressure of 75 lb. per square inch, raising water out of a well or reservoir under a suction lift of 18 to 20 ft. The slippage under such conditions very likely would be much different from the slippage of a pump which might be tested by pumping out of a shallow tank or by allowing the water to discharge over a weir onto the ground.

In some tests which the writer recently made on pumping plants of various kinds, an opportunity was afforded for measuring the slippage of a number of those pumps. Table 1 gives the principal facts observed and recorded, but to make the results of greater use some explanation is necessary.

At Elkhorn and Platteville, Wis., the sources of supply are from deep wells which are pumped with an air lift. Here it was possible to fill the reservoirs previous to the test and then pump the water out, noting the strokes of the pump, the depth to which the water has been lowered and the time required to pump this amount. At Elkhorn, the direct-acting steam pump was short-stroking so that the stroke was only 8½ in. in place of 10 in. The displacement and slip given in the table are computed for an 8½-in. stroke, but the same figures for a 10-in. stroke would be 6.65 gal. and 22.4 per cent. slip. At Platteville, the displacement was based on the full stroke.

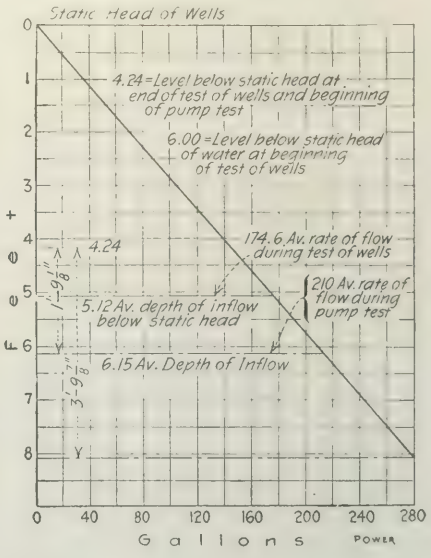

Static Head of Wells
4.24=Level below static head at end of test of wells and beginning of pump test
6.00=Level below static head of water at beginning of test of wells
174.6 Av. rate of flow during test of wells
210 Av. rate of flow during pump test
4.24
1'-9⅛"
5.12 Av. depth of inflow below static head
6.15 Av. Depth of Inflow
3'-9⅞"
0 40 80 120 160 200 240 280
Gallons
POWER

FIG. 1. METHOD OF CALCULATING FLOW OF WELLS

At Kaukauna, Grand Rapids and Watertown the supplies are obtained from flowing wells discharging into masonry reservoirs. At the two latter places the conditions were such that the flow of the wells was not affected by the level of the water in the reservoirs so that it was possible to gage the flow previous to the test, when the pumps were not operating, by noting the rise of the water in the reservoir in a given time. The actual quantity of water pumped during the test is the amount pumped from the reservoir as measured by the depth pumped plus the flow of the wells for the given time.

The supply at Kaukauna is from two artesian wells which discharge against the head of water in the reservoir. When the water was pumped down, the wells increased in flow, so it was necessary to determine not only the flow of the wells at any one level but the rate of increase of flow as the water in the reservoir receded due to pumping. The approximate static head of the wells was obtained from the engineer and a record of the time and depth of inflow was recorded for a period of 3 hr. 46 min. During this time the water level raised 1 ft. 9⅛ in., corresponding to a flow of 39,400 gal. in 226 min., or an average of 174.6 gal. per minute. At the end of the test for inflow, the pumps were started and operated for 1 hr. 45 min., during which time the water was lowered 3 ft. 9⅞ in.

By constructing a diagram as shown in Fig. 1 and assuming that this rate of increase would be uniform, which is fairly correct for artesian wells, and placing the average flow of 174.6 gal. at an average depth of the inflow, or at 5.12 ft. below the static head, it is possible to read from the diagram the average rate of inflow during the period of pumping, which is approximately 210 gal. per minute or a total inflow during the test of the pumps of 22,050 gal.

This test could have been made by the method used at Wausau, Wis., which was as follows: At Wausau one of the supplies is from a round well 36 ft. in diameter by 35 ft. deep, the water en-

TABLE 1. RECORD OF TESTS FOR PUMP SLIPPAGE

Location	Type of Pump	Size of Pump, Inches	Capacity of Pump, Gal. per Stroke	Diameter of Reservoir	Capacity per Foot of Depth	Depth Pumped out of Reservoir	Gallons of Water Pumped out of Reservoir	Flow of Wells during Test	Total Gallons Pumped	Duration of Test, Hr.-Min.	Str'kes of Pump	Actual Discharge per Stroke, Gal.	Slip in Per Cent. of Displacement
Elkhorn, Wis.	Compound duplex	8, 12&7x10	5.65	18′	1,904	16′– 3¾″	31,038 0	0	31,038	2–13	6013	5 16	8 6
Kaukauna, Wis.	Compound duplex	14, 20&12x18	35.3	61 7′ 1–30′	22,400 5,199.57	1′– 9⅛″	85,600	22,050	1,076,501	1–45	3212	31.92	9 5
Platteville, Wis.	Compound duplex	12, 18&10x12	16 32	1–29′	5,285.00	2′–10⅜″	29,635	0	29,635	2– 0	2336	12.68	22 3
Grand Rapdis, Wis.	Double-acting triplex	8x8	10 1	10′	9,424.5	4′– 0″	37,698	17,200	54,898	2-11½	6580	8.35	17.3
Wausau, Wis.	Crank and flywheel Compound duplex	14½x24	64 12	36′	9,613.7	2′–10″	21,580	28,764	50,344	0-35	1016	49 52	22 8
Watertown, Wis.	Compound duplex	14, 20&12½x18	38.3	60′	21,054.0	1′– 1″	28,000	14,220	42,220	1– 0	1204	34 9	8 2

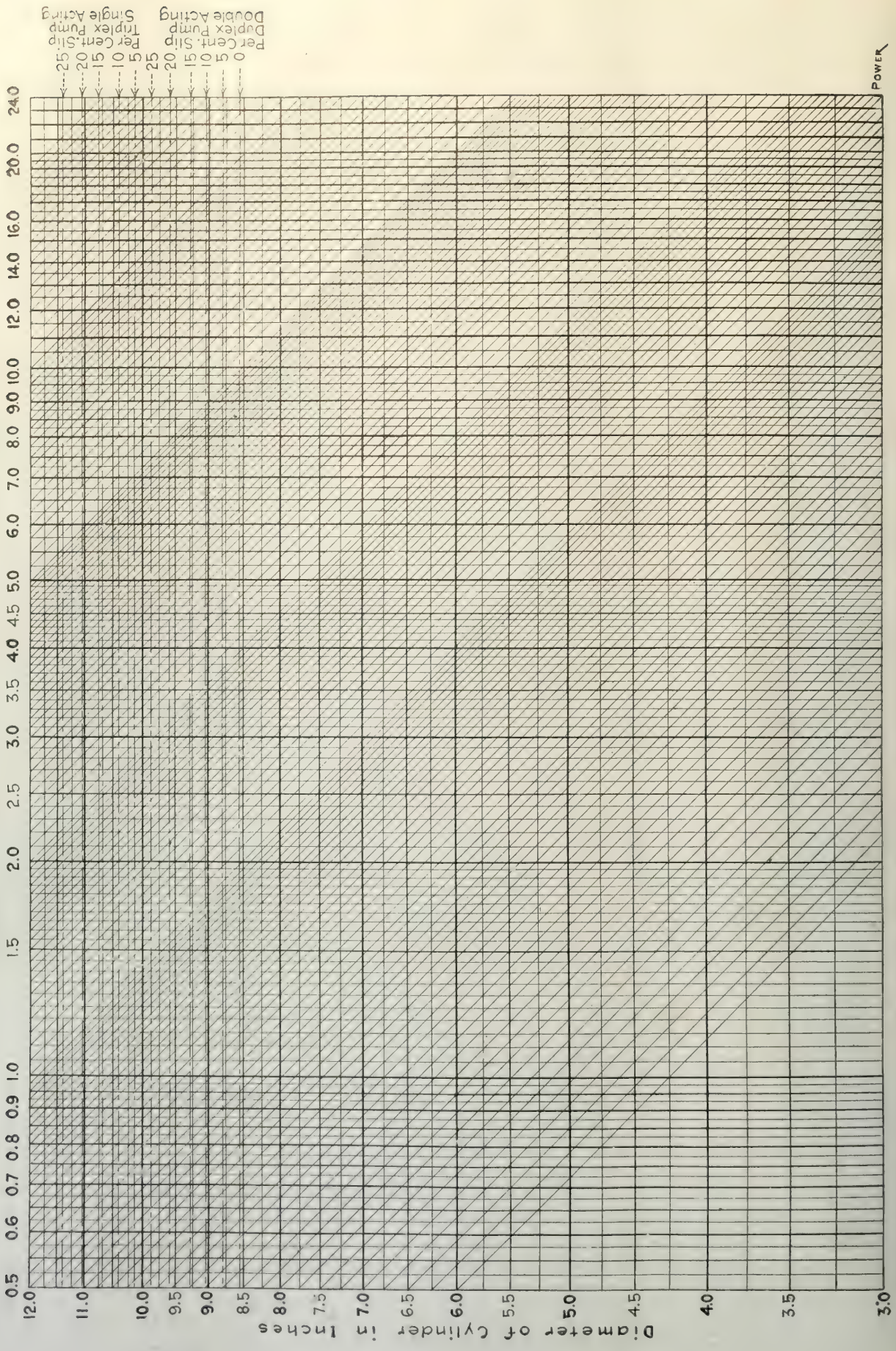

Per Cent. Slip
Duplex Pump
Double Acting
25 20 15 10 5
Per Cent. Slip
Triplex Pump
Single Acting
25 20 15 10 5 0
POWER
Length of Stroke in Inches and Gallons per Stroke
0.5 0.6 0.7 0.8 0.9 1.0 1.5 2.0 2.5 3.0 3.5 4.0 4.5 5.0 6.0 7.0 8.0 9.0 10.0 12.0 14.0 16.0 20.0 24.0
Diameter of Cylinder in Inches
12.0 11.0 10.0 9.5 9.0 8.5 8.0 7.5 7.0 6.5 6.0 5.5 5.0 4.5 4.0 3.5 3.0

FIG. 2. CHART FOR DETERMINING PUMPAGE FOR A GIVEN PERCENTAGE OF SLIP

tering through the open bottom. The water is pumped by two Holly-Gaskell direct-pressure crank and flywheel pumping engines. As there were no storage reservoirs and the pressure had to be kept up by constant pumping, it was necessary to make the test by observations upon this round well. As the flow of water came in through the bottom and increased more rapidly than was proportional to the lowering of the water, it was necessary to resort to other methods than those used at the other places mentioned. The rate of inflow for short periods of time, such as intervals of 5 min., no doubt increased fairly uniformly.

Proceeding on this assumption a hook gage was arranged in the well so as to have a long travel and readings on the level of the water in the well and the revolutions of the pump were taken every 5 min. for a period of 35 min. The period was short because the inflow soon become nearly equal to the pumpage and the high velocity made the water turbid. The pumping was continued for a few minutes after the pumping test was completed, or until the suction could be changed over to a driven well system. Then a record was made of the time it took the water to rise to each of the readings made when pumping out the well. A copy of this record for one of the pumps is given in Table 2. The quantity of water pumped during any 5-min. period is represented by

$$Q = 7613.7\ L\left(1 + \frac{t}{t'}\right)$$

in which t' must correspond to the time required for filling the well a corresponding distance of L ft. The last part of the formula within the brackets represents the flow of the well in the time t and for sake of clearness will be explained.

If a quantity of water flows into a well, or reservoir, represented by a depth L, which was observed as shown in the table, then the average flow per minute is 7613.7 L divided by t' minutes and the flow in the time t during which the pumping took place is t times this amount or

$$7613.7\ L\left(\frac{t}{t'}\right)$$

The 1 in the brackets represents the water in the well as storage which was pumped out during the time t.

The data given in Table 1 are also shown graphically in Fig. 2. The slip of the pumps was computed by the following formula:

$$S = \frac{Q - q}{Q} = \frac{64.1179\ R - 7613.7\ L\left(1 - \frac{t}{t'}\right)}{64.1179\ R}$$

where,

Q = Displacement of pump in gallons in time t;
R = Revolutions of pump in time t;
q = Actual discharge of pump in gallons;
L = Lowering of water in time t;
t' = Interval of time for well to fill up distance L which was pumped out in time t;
t = Interval of time in taking readings during pumping; it must be a constant interval;
S = Slip of pump in percentage of displacement;
64.1179 = Displacement in gallons for one revolution of pump;
7613.7 = Capacity in gallons for one foot of depth of reservoir;

$$Q = 64.1179\ R;$$

$$q = 7613.7\ L\left(1 - \frac{t}{t'}\right).$$

This test was applied to both pumps, one at a time. The average of a number of determinations of each was 22.8 per cent. for one and 22.66 per cent. for the other. In using a method of this kind extremely accurate readings must be taken; otherwise the results will be of little value.

Discussing the results of all the tests briefly, it is the belief of the writer that the results found at Elkhorn, Kaukauna and Watertown show a fair average working condition of the pumps, considering that they have been in service from five to ten years. At Platteville the high percentage of slip, no doubt, was due in part to short-stroking the pump. At Grand Rapids, the pumps were not kept in a good condition from a mechanical standpoint. At Wausau, the high percentage of slip was due in part to the age of the pumps (having been in use about 25 years) and to a heavy suction lift.

In making tests of this kind, or in any engineering work connected with pumps, it is often convenient to have at hand some ready means of determining the displacement of pumps and the actual capacity at differtnt percentages of slip. For this purpose the writer has devised the diagram shown in Fig. 2. The use of this diagram perhaps can be explained best by a few examples.

EXAMPLE—What is the capacity of a 6x8-in. double-acting duplex pump, assuming no slip?

From the left-hand side of the diagram follow the horizontal line opposite 6 in. until it intersects the vertical line through 8 in., the length of the stroke; then follow the diagonal line from this intersection upward to the left until it intersects the horizontal dot-and-dash line of zero slip; now follow the vertical line to the top of the diagram and read off 3.92 gal. per stroke. If the capacity is desired when the slip is 5 per cent., follow the diagonal lire upward to the horizontal dot-and-dash line representing 5 per cent. and read off as before at the top of the diagram 3.7 gal. per stroke. Similar determinations can be made by this diagram for any percentage of slip up to 25.

If the pump were a 6x8-in. single-acting triplex the same procedure would be followed out except that in following upward on the diagonal line, proceed farther up until the line of zero slip is intersected for these pumps. If the size and type of pump are known and the actual discharge has been determined by test, such as described in the first part of this article, the percentage of slip may be readily read from the diagram by following down a vertical line from the actual capacity until it intersects the diagonal line from the intersection of the horizontal and vertical line representing the diameter and the stroke of the pump.

The diagram can also be used for determining the discharge of the pump per minute when the number of strokes is known. For example, take the same size of pump used in the previous problem and assume that it is making 40 strokes per minute. Forty times eight equals 320. The scales being logarithmic, this may be read as 3.2 on the diagram.

TABLE 2. RECORD OF TEST FOR SLIPPAGE OF PUMP AT WAUSAU, WIS.

Time of Pumipng, Read Down	Time Interval of Pumping, t	Time Filling Reservoir, Read Up	Time Interval of Filling, t'	Gage Reading in Well	Lowering in Feet, L	Reading of Revolution Recorder	Revolutions per Interval of 5 Minutes	$L\left(1 + \frac{t}{t'}\right)$	Displacement of Pump, Gallons	Actual Discharge, Gallons	Per Cent. Slip
8:40	5	10: 9:10	.....	7.970	.....	3249	.....	.....	.....	.....	.....
45	5	9:58:38	10 32	7.375	0.595	3388	139	0.877	8,915.0	6680.00	25
30	5	9:51:15	7–23	6.780	0.595	3541	153	0.998	9,815.0	7598.48	22.5
55	5	9:46:07	5–08	6.195	0.585	3723	182	1.155	11,665.0	8793.84	24.6
9:00	5	9:43:01	3–06	5.745	0.45	3905	182	1.174	11,665.0	8938.50	23.4
05	5	9:40:57	2–04	5.395	0.35	4086	181	1.197	11,600.9	9113.61	21.5
10	5	9:39:35	1–22	5.135	0.26	4265	179	1.221	11,476.0	9220.20	19.7

Average, 22.8

From the intersection of the horizontal line through a diameter of 6 in. and the vertical line through 3.2 in., proceed as before and read the true displacement at 156 (actual reading 1.56) gallons per minute at the top of the diagram. Where the intersection of the horizontal and vertical lines come above the horizontal lines of slip, such as would be the case for a 10x10-in. cylinder, the diagonal line is followed downward to the right in place of upward to the left. For the larger sizes of pumps represented on the diagram where the intersections of the diagonals with the slip line would come off the diagram, such as 10x20 in., the capacity may be determined by using a 2-in. stroke in place of 20 and multiplying the result by 10.

The diagram is also useful in finding what cylinders of different dimensions and strokes have the same capacity. In this connection it is interesting to note that cylinders of one-half the diameter and four times the stroke of a given cylinder have the same capacity; that is to say, a 12x1½-in. cylinder is equivalent to a 6x6-in. or a 3x24-in. cylinder in capacity per stroke.

This diagram is limited in diameters of cylinders between 3 and 12 in. and for strokes from 0.5 to 24 in., but by making use of its logarithmic properties, it may be used for any stroke. It is also limited to the use of double-acting duplex pumps and single-acting triplex pumps. Similar diagrams could be readily prepared for single-acting single-cylinder or for double-acting triplex pumps and for greater ranges of diameters.

Care and Management of Water Tube Boilers

By C. K. Heasley

Care should be exercised in providing boilers of sufficient capacity for the work required. Many steam plants equipped with water-tube boilers are so arranged that the engineer does not have the opportunity to properly care for the boilers. A sufficient boiler capacity should always be provided so that one boiler may be down for overhauling and washing out. The water-tube boiler should be washed out just as often as any other type and under ordinary conditions it should be done at least every 30 days, although local conditions will determine just how often in the individual case.

The following method of washing out horizontal water-tube boilers gives best results: First, the boiler should be cut out of battery and allowed to stand with the dampers closed until the pressure has been reduced to zero. Then, open the blowoff and allow the water to run out until the drums are empty. Remove the manhead and thoroughly wash the drum. Next allow the water to run out to the bottom level of the top row of tubes. Close the blowoff and remove the caps or handholes on the top row of tubes and pass the scraper or turbine or cleaner through each tube. Allow the water to run out so as to empty the second row of tubes. Repeat this until all of the tubes have thus been scraped. When the last or bottom row is cleaned, the boiler is empty and clean and the heat from the furnace walls, arches and baffles has not baked any of the mud onto the tubes.

Care should be taken to clean out the combustion chamber and the space underneath the steam drums, particularly between the tubes, where the ash and soot usually clinker together and fill up between the tubes at the rear end. It is just as important to keep the tubes clean externally as internally, for if the space between becomes filled up this localizes the heat and often is the direct cause of a ruptured tube. Care should be taken never to blow out between the tubes with wet steam. Always allow the steam to blow through the blow pipe for a sufficient length of time to allow all of the wet steam to escape. If steam with excess moisture is used for this purpose, it will dampen the soot and ash and clog up between the tubes. After the boiler is fired up, these will burn hard and become like clinkers, and it will be found very difficult to remove them, especially in boilers where the tubes are staggered.

In erecting boilers they should be set, if possible, at least 4 ft. apart, to allow of cleaning out between the tubes from the side by having side doors located between baffles. Many manufacturers do not provide openings necessary for the proper access to the boiler so as to care for and keep it in good condition. Engineers in charge of the erection of boiler plants should see that these provisions are made.

The mud drums and bottom nipples should be thoroughly scraped each time the boiler is washed out and coated with some good nonrusting paint. A good paint for this purpose and one that can be relied upon is made by mixing a good quality of graphite with raw linseed oil to the consistency of thin paint and then either rub or paint on with a brush, going over all bottom nipples and the mud drum, also the rear end of all the bottom tubes which may draw dampness from the bottom. When the combustion chamber is wet down for cleaning or when the boiler is being washed out, the water will soak up among the tubes, particularly the first and second rows. The blowoff pipes should receive the same treatment.

If the circulating nipples and the rear ends of the bottom tubes are kept dry and free from moisture they will not corrode externally or become thin except by the wear and tear caused by the passage over them of the escaping heated gases. It is seldom that tubes waste away on the inside, although this is sometimes the case where the feed water contains properties that cause pitting and corrosion. All baffle plates and arches should be examined carefully each time the boiler is washed out. The arches and furnace walls should be kept in perfect condition so as to insure a proper distribution of the heated gases. All manhead plates should be grooved and treated with a solution of graphite and oil; also all handhole plates and caps, as this will prevent corrosion and the wasting away of the plates in case of leakage.

If handhole plates leak and corrosion commences when a boiler is in operation, they should be treated with a thin solution of graphite and oil or kerosene oil. Kerosene is a most satisfactory remedy. From an oil can pour the kerosene on, commencing at the top of the header, and allow the oil to run down all over the headers, covering the handhole plates and all. This will also prevent corrosion and wasting away of the plates. If such precautions are taken, there is no good reason why water-tube boilers should not last twice as many years as they do; if they were properly constructed, erected and cared for, their life would be doubled.

The writer has in mind four water-tube boilers of 400 hp. capacity each, located in a large office building in a Western city. These boilers are used two at a time for 30 days; in the meantime they are thoroughly washed out and properly cared for. They have been in constant use for the past 10 years and they have never had one cent's worth of repairs outside of those necessary to the furnace walls and arches. These boilers are equipped with flat grates, hand fired in dutch ovens; the arch is extended back to about 6 ft. from the rear end of the tubes.

The evaporation under ordinary running conditions is between 8 and 9 lb. of water per pound of coal, and there has been no complaint from the smoke inspector. These boilers are cared for and managed along the lines suggested in this article. There is another plant, a duplicate of this one, where the management is just the reverse. The boilers are washed out once every four to eight months and new tubes are provided from time to time. When the inspector made an examination of the tubes recently, they were found almost half filled with scale. I doubt very much if the evaporation under ordinary circumstances would exceed 4 or 5 lb. of water per pound of coal. If the loss in fuel in operating this plant is compared with the one first mentioned, a good illustration will be had of what proper care and management mean in the economical operation of a plant.

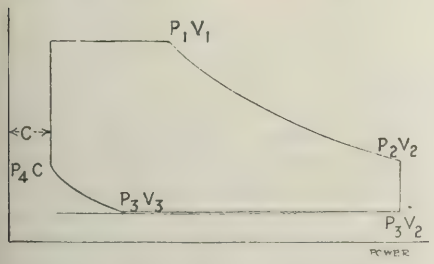
Altitude and Power Plant Economy

By Prof. A. G. Christie

In general, engine room economy is practically the same regardless of location. Boiler evaporative efficiency is independent of altitude, but an increase in grate area or draft is necessary.

Power plants in many of the new cities of the Western mountain states and of western Canada are located at altitudes greater than 2500 ft. above sea level. The writer was recently interested in the design of a plant to be about 3500 ft. above sea level and the question of the effect of altitude on the operation and efficiency of power-plant machinery, received considerable attention. Some engineers maintained that relatively as high vacuum or as good engine economy cannot be obtained at high elevation as at sea level. The following is taken from notes prepared at that time, in which the effect of altitude on both engine and boiler equipment is considered. It is hoped that this discussion will be of assistance to engineers who have to deal with plants similarly located.

In considering reciprocating engines at the two elevations, ideal indicator diagrams were analyzed both for condensing and noncondensing conditions with various degrees of clearance. Certain conditions were assumed and the mean effective pressure of each theoretical diagram was calculated as follows:



THEORETICAL INDICATOR DIAGRAM

The illustration represents a theoretical indicator diagram where P_4 is the admission pressure in the cylinder, V_1 the total volume at cutoff, P_2 the pressure at release, V_2 the volume swept over by the piston together with the clearance C. The back pressure is P_3, and V_3 is the total volume at the beginning of compression. Admission takes place at pressure P_4 and at the end of the stroke. The expansion and compression lines are assumed to be rectangular hyperbolas; that is, the product of absolute pressure and total volume at any point is a constant for each curve. If the pressures are measured in pounds per square inch, absolute, and the volumes in cubic feet, then the mean effective pressure of the diagram can be represented by the following formula, which is of the same form as the one given on page 931 (eighth edition), of Kent's "Mechanical Engineer's Pocket Book."

$$M.e.p. = \frac{P_1 V_1 \left(1 + hyp.\ log.\ \frac{V_2}{V_1}\right) - P_3\left[(V_2 - V_3) + V_3\ hyp.\ log.\ \frac{V_3}{C}\right] - P_4 C}{V_2 - C}$$

NONCONDENSING ENGINES

A small noncondensing engine, such as is often used with a throttling governor on auxiliaries, was considered first. Its operating conditions were assumed to be inlet pressure 100 lb., gage; cutoff, one-third, release at the end of the stroke, back pressure 1 lb. above the atmosphere, compression commencing at 20 per cent. of the stroke and the clearance volume 12 per cent. of the piston displacement. First consider such an engine in a plant at sea level with a barometer of 29.92 in. of mercury, equivalent to 14.7 lb. per sq.in., absolute, pressure. Representing the volume swept over by the piston by 100, and substituting in the equation at the bottom of the column

$$M.e.p. = \frac{114.7 \times 45.3\left(1 + hyp.\ log.\ \frac{112}{45.3}\right) - 15.7\left[(112 - 32) + 32\ hyp.\ log.\ \frac{32}{12}\right] - 114.7 \times 12}{112 - 12} = 67.48$$

Suppose this engine moved to 3500 ft. above sea level where the barometer normally reads 26.25 in. of mercury or 12.9 lb. per sq.in. pressure. If the valve setting remains the same the only values that will change in the above equation will be the absolute pressure. Then

$$M.e.p. = \frac{112.9 \times 45.3\left(1 + hyp.\ log.\ \frac{112}{45.3}\right) - 13.9\left[(112 - 32) + 32\ hyp.\ log.\ \frac{32}{12}\right] - 112.9 \times 12}{112 - 12} = 68.17$$

The power that can be developed in an engine cylinder is proportional to the mean effective pressure; hence, the power of this engine at the two altitudes is in the proportion of their relative mean effective pressures. Therefore, the power is increased by

$$\frac{68.17}{67.48} = 1.01\ per\ cent.$$

by moving the engine to the higher altitude.

To determine whether clearance and compression have any effect on the result, the same pressures were assumed as in the preceding case, while the clearance was made only 4 per cent. of the piston displacement, compression commencing at 10 per cent. and cutoff taking place at one-quarter stroke as in a small Corliss engine. At sea level

$$M.e.p. = \frac{114.7 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 15.7\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 114.7 \times 4}{104 - 4} = 54.27$$

At 3500 ft. elevation

$$M.e.p. = \frac{112.9 \times 29\left(1 + hyp.\ log.\ \frac{104}{29}\right) - 13.9\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 112.9 \times 4}{104 - 4} = 55.09$$

In this case the power of the same engine at high altitude exceeds that at sea level by

$$\frac{55.09}{54.27} = 1.0151$$

Several other cases were computed besides those given above. From these results the capacity or power of an engine running noncondensing evidently is practically unaffected by altitude as the increases shown above are of a very small order.

An engine's steam consumption cannot be accurately determined from an indicator diagram. The Mollier diagram is therefore used to compare the relative economy of an engine at the two elevations.

The efficiency of a steam engine or turbine is defined as the ratio of the percentage of heat turned into work in the actual engine per pound of steam to that which could be turned into work in an ideal engine without friction, radiation

or other losses and with the steam expanded adiabatically. This definition is fully explained in POWER, May 2, 1911, where a Mollier heat-entropy diagram is also shown. As far as present knowledge goes, the efficiency ratio of a given engine or turbine may reasonably be assumed to be constant at all elevations. Hence, the heat energy turned into work at the two altitudes will be proportional to the heat available from an adiabatic expansion from initial to final conditions in the two cases, the steam being assumed to be initially of 98 per cent. quality, which is common in practice.

By the above mentioned diagram, the heat available from adiabatic expansion from 114.7 lb. per sq.in. abs. and 98 per cent. quality to 15.7 lb. abs. exhaust pressure at sea-level conditions, amounts to 142.7 B.t.u. per lb. At 3500 ft. elevation 149.4 B.t.u. are available from a similar expansion from 112.9 lb. abs. and 98 per cent. quality to 13.9 lb. abs. at exhaust. Hence, each pound of steam at 3500 ft. elevation will produce

$$\frac{194.4}{142.7} = 1.047 \text{ lb.}$$

more work than at sea level, or, conversely, at this altitude the engine will only require 95.5 per cent. of the steam used at sea level. Apparently, therefore, a noncondensing engine or turbine is more economical at higher altitudes than at sea level.

Condensing Engines

Assume a compound Corliss condensing engine with 4-to-1 ratio of cylinders and cutoff at one-quarter stroke in the high-pressure cylinder. Let the steam pressure be 150 lb. gage and the clearance in each cylinder 4 per cent. Let compression commence 10 per cent. from the end of the stroke in each cylinder, and assume an exhaust pressure of 2 lb. abs. As the receiver losses and diagram factors will remain practically the same in both cases, they will not be considered and the mean effective pressure will be calculated for the whole of the combined diagram. At sea level conditions

$$M.e.p. = \frac{164.7 \times 10.25\left(1 + hyp.\ log.\ \frac{104}{10.25}\right) - 2\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 164.7 \times 4}{104 - 4} = 47.38$$

at 3500 ft. altitude

$$M.e.p. = \frac{162.9 \times 10.25\left(1 + hyp.\ log.\ \frac{104}{10.25}\right) - 2\left[(104 - 14) + 14\ hyp.\ log.\ \frac{14}{4}\right] - 162.9 \times 4}{104 - 4} = 46.77$$

The heat available from adiabatic expansion under sea-level conditions with 98 per cent. initial quality is 283.4 B.t.u. per lb. and 283 B.t.u. in the second case, so that the steam consumption of an engine under such conditions evidently is practically unaffected by altitude. On the other hand, the power of the engine, as shown by the mean effective pressure at 3500 ft. elevation is only 0.9885 of its power at sea level with the same valve setting. Hence, power decreases with altitude while the steam consumption may be said to remain constant. With a steam turbine, the exhaust pressure would probably have been maintained at 1 lb. abs., and superheat would also likely be used. Assuming 100 deg. F. superheat at admission, the heat available at sea level would be 343 B.t.u. per lb. and at 3500 ft. 342.3 B.t.u., which again shows that the steam consumption is practically unaffected by changes in altitude. These deductions are important to purchasers of machinery to be located at high altitudes for manufacturers should give practically as good guarantees on their turbines under these conditions as at sea level.

At the higher altitudes the air will be less dense and hence the loss due to air friction and windage in the flywheel and generator of an engine or in the rotor of a turbo-alternator will be less than at sea level. This increase in efficiency will be relatively small.

With inclosed turbo-alternators, a greater volume of air will have to be forced through the rotor to produce the same cooling effect as at sea level through the work done in forcing such air through will remain practically the same.

With regard to the heating of generator windings at high altitudes, the Standard Rules of the American Institute of Electrical Engineers for testing electrical machinery, as given in Foster's "Electrical Engineers' Pocket Book," state that for every 0.39 in. difference in barometric pressure from 29.92 in., the observed temperature rise is to be reduced 1 per cent. to compare with standard guaranteed temperature rises. For the case under consideration at 3500-ft. elevation a correction of 9.54 per cent. would be made. This rule fits all cases very well, whether the generator has open ventilation or forced ventilation as in a turbo-alternator. This increased temperature rise indicates that the heat losses in a generator will increase with altitude. Hence the generator efficiency will decrease slightly at high elevations.

Engine Room Auxiliaries

Some engineers believe that as good a relative vacuum cannot be maintained at a high altitude as at sea level. The factors affecting the vacuum in any condenser, if sufficient cooling water is available, are chiefly air entrained in the boiler-feed water, air entrained in cooling water, if a barometric or jet type of condenser is used, and air leakage.

The amount of air entrained in the boiler-feed water, and thus carried through the engine to the condenser, depends largely on the temperature of the feed water. As the temperature of this feed water approaches the boiling point at atmospheric pressure, it will contain less air if an open heater is used. Apparently, therefore, this is a question of temperature alone and hence is not affected by altitude.

A bottle filled with water saturated with air at sea level and then uncorked at a high altitude will behave similarly as a freshly opened ginger ale bottle—bubbles would appear as soon as the pressure is released. Hence at a high altitude water will not carry as great a weight of air when thoroughly saturated and at the same temperature, as at sea level. The amount of entrained air in conditions between absolutely pure water and saturated water will vary with local conditions, but generally less entrained air will be in the cooling water at high altitudes than at sea level.

Regarding air leakage, the hole through which the leak occurs, whether a pin hole or a gasket leak, may be considered as a small orifice. Air practically follows the laws of perfect gases. Hence, if the absolute pressure on the inside of the vessel is less than 0.58 of the outside pressure, the quantity of air flowing in will remain the same. The absolute pressure in the condenser is usually less than 58 per cent. of the atmospheric pressure regardless of the altitude. Therefore, with the same conditions as regards air-leakage holes, the amount of air leaking into the condenser will be constant regardless of altitude.

The foregoing shows that probably the air pump would have less air to handle at the higher altitudes than at sea level, due to the smaller quantity of entrained air in cooling and feed waters, and hence with the same sized air pump a higher vacuum might be carried with jet or barometric condensers.

The work done by the air pump will be less at the high altitude. For example, at 3500 ft. altitude and 1 lb. absolute condenser pressure the air pump has to compress less air from 1 lb. abs. to 12.9 lb. abs. pressure, while at sea level it compresses more air from 1 lb. abs. to 14.7 lb. abs.

The work required to drive the circulating pumps with either barometric or surface condensers under similar conditions of temperature and vacuum will be

practically the same regardless of altitude. The wet-air pump of a jet condenser has to lift the water against a smaller absolute pressure at the high altitude than at sea level, and hence does less work.

Therefore, relatively as high vacuum can be obtained at the high altitude as at sea level, and this can be produced at a lower operating cost. Consequently, striving for the highest possible vacuum at the high altitudes when steam turbines are used is advisable.

A thermos bottle would lead one to conclude that, in general, radiation losses from boilers, piping, etc., would be less at the lower atmospheric pressure of elevated locations. While such a decrease in air density no doubt has some slight effect, experiments along a similar line failed to show sufficient decrease in radiation loss to warrant consideration. Hence, in a commercial sense, radiation losses are independent of elevation.

Summarizing briefly, all noncondensing engines as exciters, air pumps, steam-driven circulating pumps and boiler-feed pumps will show better steam economy at the higher altitudes, the condensing apparatus will also take less power while the main engines will consume very slightly more steam. In general, then, as far as the engine room is concerned, the steam economy probably will be practically the same regardless of location.

Effect of Altitude on Boiler Plant

Under usual operating conditions 20 lb. of air is required per pound of coal burned in the boiler furnaces. If this air enters the furnace at 60 deg. F., 1 lb. of air will occupy a volume of 13.07 cu.ft. at sea level, while at a 3500-ft. elevation it will occupy 14.89 cu.ft., at the same temperature. That is, the furnace at the high elevation must be supplied with 14.5 per cent. greater volume of air than a furnace at the lower elevation. Then, either the area of the grate must be increased 14.5 per cent. with the draft remaining constant, or the draft must be increased sufficiently to provide for this extra volume of air. Therefore, with the same furnace economy and burning the same amount of coal per hour, furnaces at the higher altitudes evidently must either have increased grate area or more draft than similar furnaces at sea level. Neglect to consider this requirement has several times resulted in limiting the capacity of boilers to loads much lower than they would be capable of carrying under more favorable conditions.

It has been suggested that such an increase in the volume of air would promote better combustion in the furnace as each particle of combustible would come in contact with a greater volume of air. The proportion of oxygen and nitrogen remains practically the same at all points on the earth's surface, so that each proportional volume of air has a combining power with carbon exactly in proportion to its weight. In the ordinary boiler furnace, apparently large volumes of air pass through the furnace without even coming in contact with the carbon. It would therefore seem probable that if the draft were increased at the higher altitude, the furnace area remaining constant, improved combustion would result from this increased volume of air striking the burning carbon at increased velocities over that occurring in a similar furnace at sea level.

If the absolute pressure is the same altitude has no effect on a steam boiler, for the heat required to produce steam at a given absolute pressure from feed water at a constant temperature, remains the same. This is shown by any steam table. If, however, the pressure above atmosphere, or, the gage pressure, is the same in both cases, there is a slight difference which will be illustrated by an example.

Suppose the boiler gage pressure to be 150 lb. and the feed-water temperature 182 deg. F. both in a boiler at sea level and in one at 3500 ft. elevation. According to the steam tables the boiler at sea level and at 164.7 lb. absolute pressure requires

1194.97 — (182 — 32) = 1044.97 *B.t.u.*

to evaporate 1 lb. of steam from feed water at 182 deg. F. A similar boiler at 3500 ft. elevation or 162.9 lb. absolute pressure would require

1194.79 — (182 — 32) = 1044.79 *B.t.u.*

to evaporate 1 lb. of steam from the same feed-water temperature. Hence, a difference of elevation of 3500 ft. decreases the heat required to evaporate 1 lb. of steam under the given conditions by 0.18 B.t.u. or 0.018 per cent. approximately. This fraction is so small that for practical purposes it is negligible. Therefore, with regard to evaporative efficiency, in commercial operation, altitude has no effect on steam boilers.

That a greater volume of air will be required in the furnace at the higher altitude has already been shown, so that a greater volume of furnace gases must pass into the boiler tubes or flues from the furnace itself. Boilers are usually built from standard designs so that the areas for the passage of gases are the same in all similar boilers. These areas are usually figured for sea-level conditions. Evidently, therefore, since a greater volume of gas passes through the same area, the gases at the higher altitude must pass through the flues or tubes at a much higher velocity than in a similar boiler at sea level, if the same amount of coal is burned with the same excess of air in both cases. Experiments recently made in England show that increased velocity of the gases through a boiler results in increased boiler efficiency. That the boiler at the higher altitude should show better efficiency than the one at sea level would therefore be expected, though it is hard to predict the amount of this increase from the available information on the subject.

With the higher flue-gas velocities, one can expect higher friction losses in the boiler flues or tubes. Hence, at the high altitude the draft required to overcome these friction and eddy losses will exceed that at sea level irrespective of the area of the grate itself. When the grate area remains constant, the draft at high altitudes will be considerably greater than at sea level, as already pointed out. Therefore, if an induced-draft equipment is provided, the fans, if of the same size in both plants, must be run faster at the high altitude. This would result in a very decided increase in the power required. A larger fan provided in this plant would also require more power to drive it than the one at sea level. Evidently, then, an induced-draft plant requires more power in a plant at high altitudes than in one at sea level.

The height of a chimney required under any given condition is determined by the formula (from Gebhardt's "Steam Power Plant Engineering").

$$H = \frac{D_1}{0.52 P_2 \left(\frac{1}{T_2} - \frac{1}{T_1}\right)}$$

where

H = Height of chimney in feet;
D_1 = Required draft in inches of water;
P_2 = Observed barometric pressure;
T_2 = Absolute temperature of outside air;
T_1 = Absolute temperature of the flue gases in the chimney.

This shows that the height of a chimney varies inversely with the barometric pressure, if the draft remains the same, which means the higher the altitude the higher the chimney required. But, as has been shown, friction losses in the boiler increase at the high altitude and hence greater draft will be required, resulting in a still higher chimney. If the grate area is not increased, still more draft will be necessary, and hence more height must be added.

Kent's formula for area of chimneys gives effective area in square feet.

$$E = \frac{0.3\ hp.}{\sqrt{H}}$$

where $hp.$ = Horsepower of boilers served and H = Height of stack in feet.

The higher the chimney, then, the less is the effective area required. But the effective area varies inversely as the square root of the height, so the height will increase relatively faster than the diameter decreases. A chimney's cost is usually determined largely by the height. Hence, at a high altitude a chimney must be higher with somewhat smaller diameter, but with better foundation, and will cost more than one for similar boiler power at sea level.

Designing Brick and Steel Chimneys

By Everard Brown

Determining height and area of stack, wind pressure and stability, methods of construction and conditions governing size of foundation.

In designing chimneys the height is generally decided upon first. This involves consideration of the height of surrounding buildings or hills, the length of all horizontal flues necessary, the character of fuel to be used, etc., as well as any local laws or ordinances that may apply. The chimney diameter may then be calculated, based upon the selected height and the boiler horsepower or the amount of fuel to be consumed.

The minimum height necessary, however, will be governed to a great extent by the fuel—wood usually requiring the least and fine anthracite coal the greatest—the character of the installation and the number of furnaces served. Obviously the smaller and more round-about the flues, the higher must be the stack; likewise, a single furnace will require less height than several discharging into a common flue. On account of these variables, it is practically impossible to evolve a general formula applicable to all cases. Except in rare instances, extremely tall chimneys are not necessary and any increased efficiency gained by them hardly ever justifies their greater cost. The tendency is to build two or more smaller ones rather than one high one, which can very often be done at less cost.

A chimney should be proportioned to give sufficient draft for the boiler to develop much more than its commercially rated power, or, according to Kent, to bring about the combustion of 5 lb. of fuel per rated boiler horsepower per hour. By assuming such a liberal rate of combustion, the stack will certainly be large enough. Based on this assumption Kent gives the following equation:

$$E = \frac{0.3\,Hp.}{\sqrt{H}} = A - 0.6\sqrt{A}$$

where:

E = Effective sectional flue area of stack in square feet;
$Hp.$ = Horsepower of boilers;
H = Height of chimney;
A = Actual sectional flue area of stack in square feet.

The actual is not the effective sectional area because friction retards the ascending gases. The annular stratum of gas retarded by the wall in reality diminishes the flue area.

Molesworth gives:

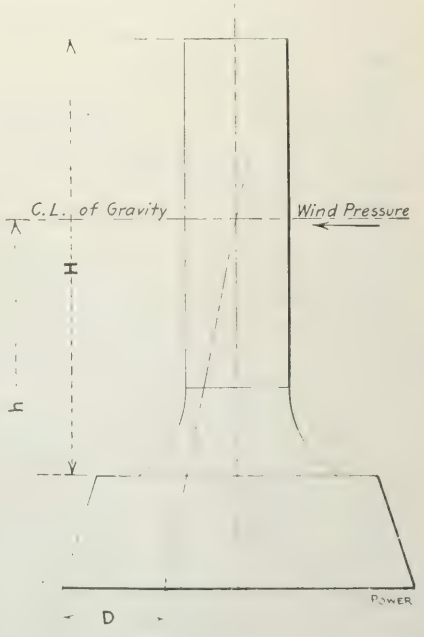
$$A = \frac{Hp.}{1.28\sqrt{H}} + \frac{C}{12\sqrt{H}}$$

where

C = Pounds of coal consumed per hour.

From this equation, H may be found if the sectional area is known or assumed, or by the same process the horsepower may be found for which an existing stack is sufficient. Sometimes windcaps are used to increase the stack capacity, but this is only rarely advisable because of their dangerous position.

Stability and wind resistance are next to be considered. Wind pressure is usually assumed to be horizontal and uniformly intense at all levels. Where a chimney is shielded by buildings, only the part actually exposed need be considered. Generally a maximum pressure is assumed of 50 lb. per square foot,



DETERMINING STABILITY OF CHIMNEY

although this would require a wind velocity seldom reached. A simple rule for calculating wind pressure is:

The square of the velocity in miles per hour divided by 200 *equals the pressure in pounds per square foot.*

The total pressure against a round chimney is about one-half that against the diametral plane of that chimney. It is therefore the product of the pressure per square foot by one-half the total projected area of the exposed part. Similarly the total pressure may be calculated for chimneys of almost any form.

The resultant of this pressure may be assumed to act horizontally through the center of gravity of the exposed part. The stability of a wholly exposed chimney, therefore, is determined by finding the moment of this pressure acting through the center of gravity. From the accompanying illustration this moment is

$$h \times P_1 = H \times P \times \frac{A}{2}$$

where

h = Height of the center of gravity above the top of the foundation;
P_1 = Total pressure exerted;
H = Total height of the stack;
P = Pressure in pounds per square foot;
A = Total vertical sectional area.

For chimneys with vertical axes, the moment of stability is the same in every direction, but few chimneys have exactly vertical axes; therefore, the least moment of stability must be considered as that which opposes the lateral pressure in the direction in which it leans. The stability at any height of the chimney, for example at a certain bed-joint of a brick structure, may be established in the same way by simply considering that part of the chimney above the point at which the stability is to be determined. Opinions differ regarding the strength of brickwork, which, together with the varying qualities of materials and workmanship, makes it exceedingly difficult to accurately calculate the power of resistance. Consequently a large factor of safety is always advisable.

Fundamentally, a chimney's stability depends upon the weight of its outer shell and the diameter or width of its base and, while the cohesion of the mortar in a brick stack may add something to its strength it is too uncertain to be relied upon. The effect of the two forces—the weight of the chimney and the pressure of the wind—is to shift the center of pressure at the base from the axis toward one side, extent depending upon the relative magnitude of the two forces. A comparatively safe rule to follow is to make the base diameter not less than one-tenth of the height. This rule has been approved by many years of actual practice.

STEEL CHIMNEYS

Probably the cheapest chimney is a straight steel shell guyed with rods or wires. These are secured to the shell, usually by an encircling band, and should be anchored to incline at an angle of not more than 45 deg. with the ground. Where other structures are convenient the guys may be fastened to them. This type of stack is rapidly being supplanted by the so called self-sustaining stack. It can be lined or not. Ordinarily when

working to full capacity, the high velocity of the ascending gases allows little heat to be lost by radiation even through an inclined shell. If the gas temperature is high, a lining is desirable at least part way up. Firebrick is sometimes used, but probably more often hard-burned red brick that can withstand temperatures up to 750 deg. without injury. This lining should never be less than 4½ in. thick at the top, whether it goes up only part or the full height of the stack, and should be thicker toward the bottom by 4½ in. for about every 35 or 40 ft. Not less than 1 in. should be allowed between the shell and the lining with no rigid bond between them, thus making both self-sustaining and allowing for expansion and contraction.

For ease of construction and economy, uniform diameter of the shell for all sections is preferable; the bottom of each section should lap outside the top of the next section below, except for brick-lined structures, when the reverse is often preferred. The rivets in horizontal joints should be double staggered, thus giving more lap for the sheets and, consequently, more stiffness to the shell. Single-riveted joints usually answer for vertical seams. If possible, each section should be made of a single plate, to avoid the extra riveting. In the flaring base both the vertical and the horizontal seams should be double staggered. The cone-shaped base is naturally the strongest, but bases curved like a bell are quite common, and may be equally strong if thicker material is used. Better appearance is the only advantage of the curved base.

Occasionally instead of either cone- or bell-shaped bases, the shell is made perfectly straight down to the foundation and anchored to it by steel braces or cables with turnbuckles. The latter are usually fastened to the stack at a height of about three diameters of the flue and have approximately the same slope as the conical base. This type of construction is hardly advisable.

Caps or other ornaments at or near the top of a steel chimney are no use and a poor investment, but if desired should be of copper to be quite light and rust proof, and firmly attached to prevent risk of being blown down, causing damage or human injury.

The chief advantages of the self-sustaining steel chimney over all others are less space occupied above the ground, and easier and more rapid construction and erection; and over brick chimneys in particular, lighter weight often saving pile driving in low load-sustaining soils; less area presented to the wind; elimination of air infiltration and consequent checking of the draft; greater strength and safety, and smaller cost. Places and conditions where steel shell construction would not be at all practicable nor economical are, for example, along the sea coast or where acid fumes are present in the atmosphere.

Brick Chimneys

The most common brick chimney is the radial type. While initially costing somewhat more usually than a steel structure, it does not have the latter's expense for painting, replacements, etc. A brick chimney is usually made up in two shafts, the shell or outer casing and the lining or core. There is no bond between these rigid enough to retard movement due to expansion caused by the hot gases. Generally the core is run up to within a few feet of the top and then battered to connect with the shell. The outer casing diminishes in thickness, section by section toward the top. The joints between sections and at the base, called bed-joints, are less stable than intermediate ones and always must be given primary consideration. A failing chimney nearly always begins by opening one of these bed-joints on the windward side and cracking along both sides diagonally downward, tending to separate the structure into two parts: an upper leeward and a lower windward. Therefore, to withstand extreme intensity of wind pressure, the vertical pressure should be sufficient to counteract and prevent tendency for the joint to open.

The lengths of each section vary according to the height of the chimney and the quality and kind of brick used. For ordinary red brick the lengths may be from 25 to 30 ft. with an outside batter of 1 to 30 or 35, or they may increase gradually from the lowest division to the highest. Using bricks of greater crushing strength, these divisions may be made longer. The wall thickness and the batter must conform to the various lengths to insure stability. The former, giving weight to the structure, and the latter, presenting a sort of lever arm, combine to resist wind pressure. Since a swaying motion is usually set up by the wind, it is never desirable to connect the chimney rigidly with any other structure.

Naturally, much depends upon the workmanship and material; both should be of the best. Important, also, is not to erect too rapidly for the mortar to properly set and the foundation should have plenty of time to harden and settle. In several ways the brickwork may be bonded satisfactorily. The most common bond is probably one row of headers for every five rows of stretchers, although alternate rows of headers and stretchers, or two of the latter for one of the former, are also used. Often in large chimneys an iron or steel hoop or tee iron is laid within the brick wall every few feet. This is good, especially where the wall thickness exceeds, say 10 to 12 in.

Concrete Chimneys

Reinforced-concrete chimneys are least common and were introduced only about 10 years ago. Information regarding them is meager and they have hardly had time to demonstrate their durability. Ordinarily, they have inner and outer shells with an annular air space between. The shell thickness depends upon height, diameter, etc. The steel reinforcement is usually vertical bars with ends overlapping, spaced according to the size of the chimney. These bars are encircled by steel rings and are extended down into into the foundation to insure a good anchorage. They are calculated to resist all pressure by the wind and any tensile strength in the concrete is usually considered an addition to the safety factor.

Foundations

Logically, the first detail is the foundation and in its design must be considered any local building laws that apply. These laws usually specify the loads allowed in tons per square foot of area for ground of various kinds. Where the soil is very loose or sandy, piling is often necessary, but, regardless of expense, a solid, amply large foundation is imperative. Brick, stone or concrete may be used either individually or collectively. Concrete is commonly used for the subfoundation with brick or large stone above. The area of the foundation should be sufficient to prevent unequal settling. It will receive greater pressure from above on the leeward side to the prevailing wind, resulting in a canting of the chimney if the foundation is not broad and substantial. The foundation should be such that wind pressure against the chimney will not unduly increase the load on it at any point.

To determine the stability of the chimney in relation to its foundation: first, find the greatest total wind pressure to which the chimney might be subjected and its moment about an axis in the plane of the base of the foundation; second, the total weight of the chimney, including its lining and also that of the foundation itself. The former divided by the latter will give the distance from the outer edge of the foundation, or the lever arm of the combined weight of the chimney and foundation producing equality of moments. If this distance D is one-half the foundation diameter at the base, the chimney may be considered as just stable with no margin of safety; as it is lessened the factor of stability is, of course, increased.

This distance should never be more than one-third that across the base of the foundation even with the best construction and anchorage. In addition some strength will be obtained from the ground surrounding the foundation, but it will be too variable to be counted on. Evidently, the greater the combined weight of the foundation and chimney, the more stable will the whole structure be made.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Three Wire Direct Current Generators

By A. M. Bennett

Three-wire generators are designed to furnish two voltages, generally 250 and 125, the former between the two outside wires and the latter between each outside wire and the middle wire.

The chief advantage of a three-wire system over the ordinary two-wire installation is that of economy in distribution. In a two-wire system, with a given load and a given percentage of voltage drop, the distribution at 250 volts requires only one-quarter the weight of copper

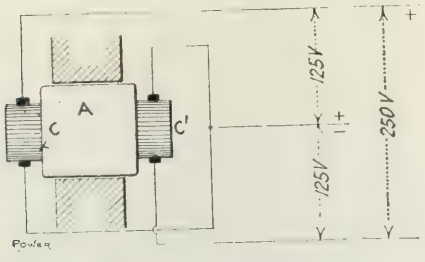


Fig. 1. Hopkinson Three-wire Generator

required for distribution at 125 volts. A third or neutral wire in the three-wire system will modify this proportion of copper, the final saving depending on the size of the neutral. In well designed systems, the maximum unbalanced current carried by the neutral will be about 25 per cent. of the full load. Therefore, the size of the neutral need not be larger than 25 per cent. of the capacity of the outside mains. The weight of copper in this case would be $\frac{3}{32}$ of that used in distributing the same power by a two-wire system.

There is also the advantage in the three-wire system of having at hand two voltages; the higher can be used for power purposes and the lower for lighting. Motors can be designed to work equally as well on the higher voltage as on the lower, and in the larger sizes their cost will be less. They are connected directly across the two outside wires and do not affect the unbalanced load. The lower voltage, namely, 110 or 125 volts, has been found most satisfactory for lighting, the lamps being connected between the neutral and each outside wire; the total load of this nature is divided so that the loads on the two sides are at all times as nearly balanced as possible. The difference between these loads constitutes the unbalanced load of the system, which is carried by the neutral wire.

As explained, the difference between the two-wire and the three-wire generator lies in the ability of the latter to furnish an extra voltage at one-half that which obtains between the regular leads of the generator. This must be done in all cases with close voltage regulation between the two sides of the system when carrying an unbalanced load. Only when the two sides are equally loaded is the voltage between the neutral and the outside wires the same. As one side becomes more heavily loaded and the unbalance increases, the voltage drops below that on the side with lighter load until at the point of maximum unbalance, for which the generator is designed, there will exist the maximum difference of voltage between the two sides of the system. Any great difference that may exist will have a detrimental effect on the life and efficiency of the lamps.

Some of the best known three-wire generators now in use, and which most successfully accomplish the desired results are included in the following:

Hopkinson Three-wire Generator

This is represented in Fig. 1, and is among the oldest types of three-wire machines. The armature *A* has two similar windings, and two commutators *C* and *C'*. Each winding is designed for one-half the total output of the generator at 125 volts. The two sides of the machine are connected in series, and their voltages are added to produce 250 volts on the outside leads. The neutral is taken off at the point of connection between the two windings. There exists, therefore, 125 volts between the neutral and each outside main. The result is similar to that obtained by the Edison three-wire system, where two 125-volt generators were run in series, with a neutral taken off at the point of connection between the two machines.

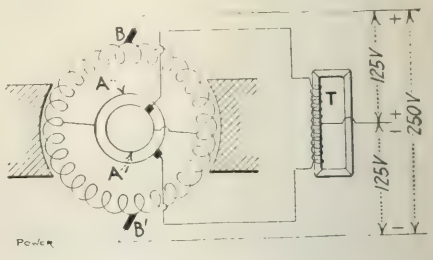


Fig. 2. Dobrowolsky Three-wire Generator with Single-phase Balancing Coil

In the Hopkinson generator, however, since the two armature windings are influenced by the same field, there is no means of compensating for the difference in voltage between the two sides with an unbalanced load, except by shifting the brushes. This would hardly be practical, as it would have to be done on each change of load from one side of the system to the other.

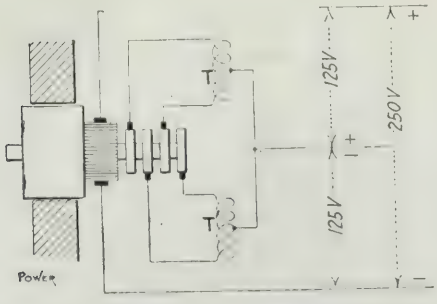


Fig. 3. With Two-phase Balancing Coil

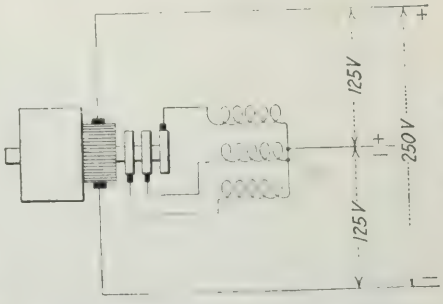


Fig. 4. With Three-phase Balancing Coil

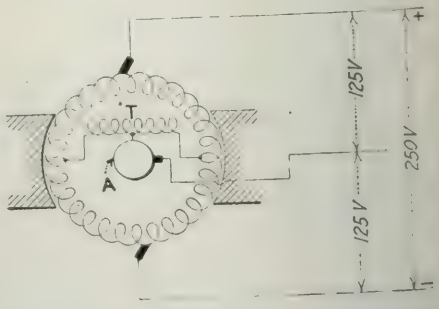


Fig. 5. With Self-contained Balancing Coil

Modifications of the Dobrowolsky Three-wire Generator

Dobrowolsky Three-wire Generator

The majority of straight three-wire generators running at present are of this type or one of its modifications. It may be regarded as an adapted two-wire generator, in that it consists of such a machine working in connection with an ordinary auto-transformer, or balancing coil. This transformer has a single winding on a laminated core. Its outer terminals are connected to brushes working on two slip rings mounted on the armature shaft, these rings in turn being tapped on to the armature winding at two points of opposite potential, forming a single-phase connection to the armature. The neutral

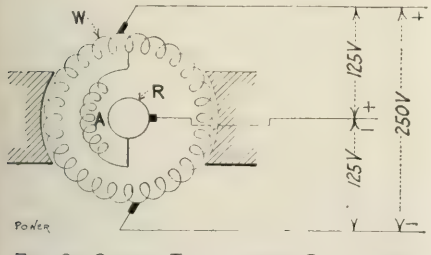


Fig. 6. Osana Three-wire Generator with Single-phase Balancing Coil

wire is taken off at the middle of the transformer. This is shown diagrammatically in Fig. 2, where T is the auto-transformer, A and A' the slip rings connected to the armature winding, and B and B' the main brushes.

As the coils of the transformer are wound on a laminated core, they have a high inductance. They are also subjected to an alternating potential from the armature, and by reason of the counter electromotive force generated in them, do not short-circuit the armature. The middle point of the transformer is always at a potential halfway between that of its two terminals; since the potential between the neutral wire and either outside lead of the generator will be one-half that between the two outside wires. The unbalanced current on the system may flow in either direction in the neutral, depending on which side is more heavily loaded. In any case, the neutral current is divided in the balancing coil. This current meets with no opposition in the coil other than that of ohmic resistance, and as it has opposite directions around the core, its magnetizing effect on the latter is zero. There is, therefore, no interference from this source with the choking effect of the balancing coil on the alternating current to which it is subjected.

In practice the auto-transformer is generally built in compact form and mounted in a cast-iron case filled with oil. It can be placed in any convenient location, the leads from it being taken to the slip rings on the generator.

A modification of the Dobrowolsky system is shown in Fig. 3. Two balancing coils, T and T', and four slip rings are employed, and the connection of the latter to the armature is two-phase. The middle points of the two balancing coils are connected and the neutral is taken off from this common connection. By this construction the path of the neutral current through the armature winding is shortened somewhat, and there is a more uniform action of the coils, with less tendency to cause a flickering of lamps on the system in the case of slow-speed machines, where the alternations are low, than with the single-phase connection. There is the disadvantage, however, of two added slip rings with their brushes, and extra leads to the machine; also the machine is longer.

Still another modification of this system, claimed to be particularly free from unsteadiness of the neutral current is shown in Fig. 4. Here three balancing coils are employed connected in "star," and three slip rings, whose connection to the armature winding is three-phase. The neutral wire is taken off at the point of common connection of the three coils.

There is a fourth modification of this type in which the balancing coil is mounted with the armature on the shaft. The outer terminals of the coil can be connected directly to the proper points on the armature winding. The neutral is taken from the middle point of the balancing coil as before, and is carried to a single slip ring, from which connection is made to the outside circuit. This is shown in Fig. 5, where T is the auto-transformer with its middle point connected to the slip ring A, and its terminals connected to the armature winding. This construction gives a very compact arrangement and the auto-transformer gets the benefit of ventilation, due to rotating with the armature. The balancing coils in this construction can be connected either single-, two- or three-phase, with the same advantages as where these coils are separate from the machine, but in any case only one slip ring is required.

The Dobrowolsky generator and its modifications have the disadvantage that it is not possible without extra mechanism, to regulate the voltage on the two sides of the system. At a given unbalanced load, the difference in voltage is fixed and is determined when the machine is designed.

Osana Three-wire Generator

In this generator, which has now come into extensive use in this country the unbalanced current is taken care of by an auxiliary winding laid in the bottom of the armature slots, beneath the main winding. This is represented diagrammatically in Fig. 6, where W is the main winding, A the auxiliary winding, and R the slip ring. The auxiliary winding is tapped to the main winding, and between the point of connection to the armature and the slip ring, has just one-half the number of turns that are in the main winding, in series between any two sets of brushes. In the position shown it is evident that the coil is at its maximum generating point, and in this position, the voltage generated by it, and existing between the neutral and outside mains, will be just one-half that between the outside mains. At 90 deg. to this position it is at its minimum or zero generating point, and the neutral voltage is generated in the main armature winding itself, the auxiliary winding serving merely as a conductor for the neutral current. Between these two positions the neutral voltage is the sum of the voltages generated in the auxiliary winding, and in that part of the main winding between the point where the former is tapped to the latter, and one of the brushes. This will always be one-half the voltage between the outside mains of the generator.

The single-phase winding shown in Fig. 6 concentrated under a single pole, has not given a steady voltage, and in practice, a polyphase winding (Fig. 7) connected at three or more points to the armature is used. This winding is distributed in such a manner, that the average flux cut by it is at all times practically uniform.

This type of machine is particularly efficient, inasmuch as core losses and excitation losses present in the Dobrowolsky type are removed. It has the disadvantage, however, of requiring an armature of larger diameter, on account of the extra room needed in the slots for the auxiliary winding. The same fault that characterizes other three-wire generators,

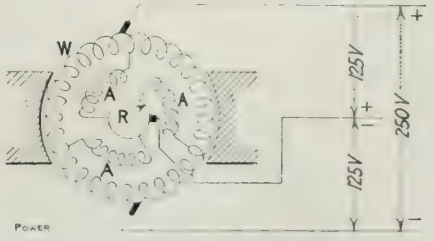


Fig. 7. Osana Three-wire Generator with Balancing Coils

namely, lack of means for regulating the voltage on the two sides of the system, is also present in this generator.

Dettmar and Rothert Three-wire Generator

While the Dobrowolsky and Osana generators bear some relation to each other in the manner in which the neutral voltage is generated, the Dettmar & Rothert machine is a distinct departure from either of these types. In this generator, shown diagrammatically in Fig. 8, a two-pole armature is rotated in a four-pole field. Poles of like polarity are placed next to each other forming in reality a two-pole field, with a gap in the center

of each pole. Opposite this gap a third brush is placed on the commutator, and the neutral wire is led off from this point. The effect of armature reaction, unlike that in the machine with normal poles, is to strengthen opposite north and south poles, and to weaken the remaining two. Like poles on each side of the neutral connection thus have unlike strength, with the result that the voltage on one side of the system is higher than on the other, even on a balanced load. This is

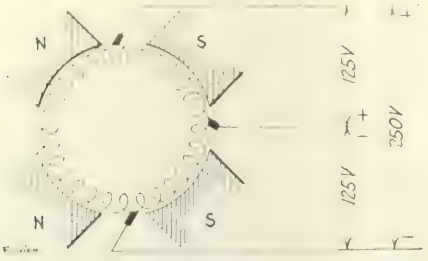


Fig. 8. Dettmar & Rothert Three-wire Generator

compensated for by the manner of connecting the compound winding on the fields, that on the strengthened poles being connected differentially, and on the weakened poles cumulatively. This method of winding regulates the voltage automatically on the two sides of the system, raising it on one side and lowering it on the other, depending on which side has the greater load. The no-load voltage on the two sides of a system fed by such a machine can also be regulated by dividing the shunt field windings into two circuits and placing a rheostat in each.

All three-wire generators, in order to regulate the voltage between the outside mains, have their series fields divided into two circuits, alternate poles being connected in the positive lead from the generator, and the remaining poles in the negative. They can be operated in parallel as satisfactorily as two-wire machines, but two equalizing connections must be provided, one for each series field circuit. Inasmuch as these equalizers are each connected to one terminal of the machine, they have the full potential of the generator across them. Adjacent series fields are also subjected to this same condition. Care must be taken, therefore, that there is no chance for a short-circuit between any of these parts.

Three-wire generators can be operated in parallel with two-wire generators without trouble, but the series fields of the two-wire machine must be arranged in two circuits, and two equalizers provided. In this case the three-wire machine handles the unbalanced load as if it were operating alone. In any installation, therefore, it is necessary to operate only one three-wire generator, provided that it is designed to take care of the full unbalanced load of the system.

In Fig. 9 are shown the connections of a three-wire generator to its switchboard. There are six cables to the board, each carrying the full armature current, and one for the neutral, which is smaller than the others. This number of cables is necessary, for in order to fully protect the generator the full armature current must flow through the circuit-breaker coils. This means that the armature leads must go direct from the machine to the switchboard and back, before passing through the series fields. From the series fields the leads are taken to the busbars on the board. The equalizing connection is made at the switchboard instead of at the machine, as is the case with two-wire generators. The ammeter shunts are also connected in the leads from the armature ahead of the equalizer. A two-pole circuit-breaker is necessary to fully disconnect the generator from the line, and its two poles must be interlocked, so that both go out together. Two double-pole and a single-pole switch are also necessary, placed as shown. This allows the series field to be thrown into circuit before starting the machine. The circuit-breaker must be open till the voltage is built up to that of the line, otherwise the armature of the incoming machine would

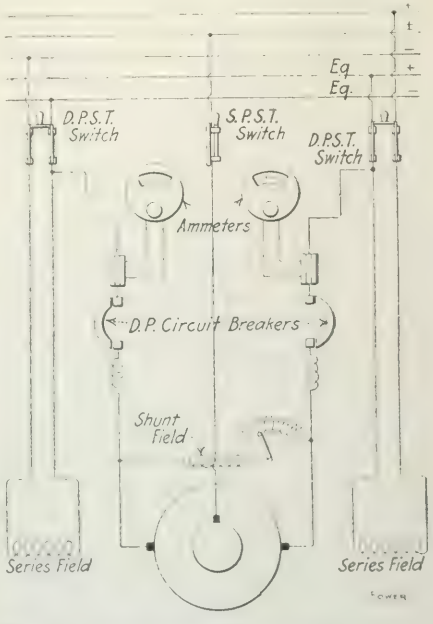


Fig. 9. Switchboard Connections for Three-wire Generator

act as a dead short-circuit to the rest of the machines on the line. Two ammeters are employed, one to read the current on each side of the system, the difference between the two readings being the unbalanced current. It is not necessary to have an ammeter in the neutral line.

It is not good practice to insert a circuit breaker in the neutral unless it is also designed to open the main lines; for if the neutral alone were opened, the neutral load would be thrown across the outside mains of the system. For the same reason fuses should not be used in the neutral line.

There is another method of connecting the generator and switchboard, in which two of the leads shown in Fig. 9 are dispensed with. In this case the ammeter shunts are mounted on the frame of the generator, in circuit between the armature leads and the series fields, and the circuit-breakers are placed in the outside lines. It is necessary in this case that the circuit-breaker open the equalizer circuit at the same time that the main circuits are opened, otherwise the machine is not disconnected from the line.

CORRESPONDENCE

Reversed Polarity

A letter by James Mitchell in the July 2 issue headed "Reversed Polarity," recalls a personal experience with reversed machines. A few years ago, I was in charge of an electric plant consisting of five generators, belted to three engines and supplying power to motors and lights in a machine shop. The generators were compound wound, each being connected to the busbars by a three-pole switch with the equalizer busbar mounted on the same board.

One day as we were starting up after the noon hour the engines were up to speed and four of the generators were connected to busbars, the fifth was just being thrown in when there was a sudden squealing of belts, and the breakers opened on the machines already on the line. It was at once evident that the machines were reversed in polarity, but we did not know how many were in this condition. We tried each machine and found one that was correct, and three of them reversed. As it was getting near 1 o'clock we opened the largest feeder switches so that the motors in the shop could not be started until everything was all right.

We threw in the good machine on the busbars and proceeded to correct the polarity of the others in the following manner: We first opened the field switch on the machine to be connected and disconnected the main lead of the machine on the side opposite from that which contained the series winding; then after closing the field switch we closed the main switch and let it remain closed for perhaps 15 sec. Next, we opened the field switch again and connected the main lead previously disconnected. On closing the field switch the polarity was found to be correct. This was repeated with the other machines, and we were straightened out by 5 min. past one.

Instructions for correcting the polarity of reversed machines, usually say "lift the brushes so that they are clear from the commutator," but when one is in a hurry to get the machines on the line, the method just described is much quicker.

L. E. Thompson.

Medford, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Utilization of Blast Furnace Gas

By Everard Brown

A blast furnace, when operating normally, will generate approximately 150,000 cu.ft. of gas per ton of pig iron produced. This gas, which is really a by-product, is sufficient for the total heating and power requirements of the furnace itself and still leave considerable that may be used for other purposes.

The tendency in the past few years has been to utilize all, or as much of this fuel as possible, although a few of the older plants still use only enough to heat the blast. The more modern plants utilize all of the waste products possible even to the slag which is now ground up and takes an important part in the manufacture of portland cement. Many furnaces are also favorably situated to furnish fuel for power to adjoining steelworks; very often this is a consideration when building a new plant.

Surplus Available

Consider, for example, a blast furnace producing 500 tons of pig iron per 24 hr. At the rate of 150,000 cu.ft. of gas per ton, it will generate 75,000,000 cu.ft. every 24 hr. Of this amount 10 per cent. must be allowed for losses and leakage; 2.5 per cent. for cleaning, if the gas is to be used in gas engines; and 30 per cent. for the stoves that heat the blast. Summarizing, it will be:

Losses and leakage	7,500,000 cu.ft.
Cleaning	1,875,000 cu.ft.
Hot blast stoves	22,500,000 cu.ft.
Total	31,875,000 cu.ft.

Deducting this from the 75,000,000 cu.ft. generated, leaves 43,125,000 cu.ft. every 24 hr. or 1,796,875 cu.ft. per hr., available for use either under boilers or in gas engines. The thermal value of this gas will range from 90 to 110 B.t.u. per cu.ft., depending upon the character and quantities of raw material charged into the furnace; therefore, assuming an average value of 100, there are 179,687,500 heat units every hour that can be utilized. This is equivalent to 12,835 lb. or 5.73 gross tons of coal at 14,000 B.t.u. per lb. With coal costing, say, $2 per ton, the saving would be $11.46 per hr. or $101,265.60 per year of 365 working days of 24 hr. each. Blast-furnace operation is continuous and is never interrupted except for relining or other extreme causes. If the fuel is to be used entirely for generating steam, the saving would practically be net but, where gas engines are employed, there would be a small deduction, owing to the necessity for additional gas-cleaning apparatus. An item of this size, however, is too large to be easily lost sight of, particularly in view of the present-day tendency toward conservation.

Disposition of Gas

The use of this energy to the best advantage seems to be a problem not yet fully solved; in some instances it is utilized entirely to make steam and in some it is all converted into electric energy through gas engines, and in still others a combination of boilers and gas engines is employed. All three have advantages and disadvantages. Perhaps the most important arguments in favor of gas-fired boilers with steam engines are reliability of operation, lower initial cost, less thorough cleaning of the gas required and less skillful attendance. On the other hand, some of the advantages of the gas engine are greater thermal efficiency, elimination of boiler plant, less ground area required, etc.

Many plant managers are still doubtful about the gas engine, particularly when used as a blowing engine, in spite of the fact that its efficiency is greater than that of the ordinary reciprocating steam engine. They seem to think that the costs for repairs, maintenance, etc., might become excessive, and object to the high first cost of such an installation. It is undoubtedly true that the charges for upkeep and capitalization on a gas-engine installation are greater than for a steam plant, but large gas engines are being developed rapidly and before long will, no doubt, be more favorably regarded than at present. Whether gas engines will be able to compete successfully with reciprocating engine and turbine combinations is yet to be settled, particularly as far as blowing engines are concerned.

Use under Boilers

To develop one boiler horsepower with blast-furnace gas having a thermal value of 100 B.t.u. per cu.ft. will require approximately 525 cu.ft. This is equivalent to a consumption of 3.75 lb. of coal per boiler horsepower, the coal having a heat value of 14,000 B.t.u. Dividing 1,796,875 cu.ft., the total amount available as previously mentioned, by 525 gives 3423 boiler horsepower. Good compound-condensing engines should develop two engine horsepower on one boiler horsepower, therefore, the total power available would be 6846 engine horsepower. Furthermore, to supply the necessary blast for a 500-ton furnace should not require more than 2500 hp., so 4346 hp. would still remain—ample to take care of the skip hoist, pumps and miscellaneous items and also leave considerable for purposes other than the actual requirements of the furnace.

The saving resulting from this use of the gas is equal to the cost of the coal displaced plus the cost for labor and maintenance, as well as interest on the difference in the initial cost of coal-fired over gas-fired boilers, where stokers are employed. The saving in labor is in itself quite an item as there are practically no coal or ashes to handle and no fires to clean. As this saving in the boiler house is due directly to the use of gas as fuel, the blast furnace should be given credit for a certain part of it; that is, credit should be given for the amount of coal displaced less the cost of any additional cleaning the gas might be given over and above that given the gas used in the hot blast stoves. Such extra cleaning would be chargeable directly to the boilers.

Use in Gas Engines

While some appliances and operations in connection with a blast furnace under all conditions require steam, their aggregate is hardly more than 200 boiler horsepower for a 500-ton furnace; therefore, being very small in comparison to the total amount generated, it can be ignored. This is assuming that gas engines are used for creating the blast as well as generating electric power for pumps, hoist, etc.

A gas engine operating normally and with a good load factor will develop 1 hp. on about 150 cu.ft. of gas of 100 B.t.u. heat value. The total energy that can be generated by this means will be,

$$\frac{1,796,875}{150} = 11,979\ hp.$$

Deducting from this the 2500 hp. required for blowing, leaves 9479 hp.—an amount far in excess of the requirements of the furnace. In this case, however, as compared with gas-fired boilers there would be a somewhat larger deduction on account of the extra cleaning and purifying required by the gas preparatory to its introduction into the engines. As for the accounting, the blast furnace should be given credit for the actual number of heat

units delivered, basing the calculation upon their coal equivalent minus the cost of cleaning.

From these figures it can be seen that blast-furnace gas plays a very important part in the economical operation of the furnace and can be made the means of increasing the profits and conserving the coal pile for future and other uses.

Cooling Water System for Suction Producers

By W. E. Wort

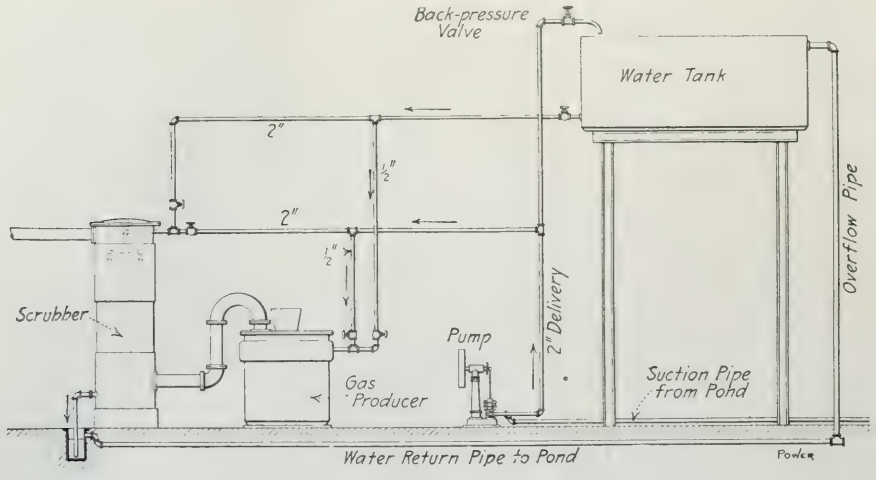
Frequently a drawback to installing a producer-gas plant for moderate power is the difficulty in obtaining a supply of cooling water, or if the water is taken from a town supply the cost is often a large factor. It sometimes happens that a suction producer plant is desired for light and power around a country estate far removed from a town supply of water, and at the same time where only a limited supply is obtainable from the well on the premises. The writer recalls a case of this kind where the capacity to be installed was about 130 b.hp. with an average daily run of 10 hr. The water required by the scrubber and the vaporizer amounted to between 350 and 400 gal. per hour which meant 3500 to 4000 gal. per day. With a limited water supply this was quite an item; hence the problem was solved in the following manner:

While installing the plant, a 50x25x3-ft. reservoir was dug, and into this all the surface water from the land was drained. In order to make the pond water-tight it was first puddled with clay over which was poured a thin layer of concrete and this, in turn, was faced with cement. At a convenient place in the pond a suction pipe was placed in a cylindrical tank which had a number of holes drilled around its sides, so as to alow the water to flow in. The suction pipe was led to a pump driven from one of the engines, and this pumped water into a tank placed about 35 ft. above the ground. From this tank a 2-in. pipe was run to the producer house, a ½-in. branch going to the producer boiler and the 2-in. pipe continuing to the scrubber. The supply in this instance was by gravity. It was found that when working the plant to its full capacity that the head of water was insufficient, so a spring back-pressure valve was fitted at the discharge end of the delivery pipe, the spring being set to give a pressure of 60 lb. per sq.in. in the delivery pipe, against which head the pump was always working. A branch pipe was fitted in the delivery pipe on the pump side of the back-pressure valve and led into the producer house, and coupled up to the scrubber and vaporizer in a similar way to the gravity feed pipe. In this way it was possible to use either the pressure or the gravity supply. The pressure supply has been found more efficient for washing and cooling the gas in the scrubber, as the gas is caused to pass through a much finer spray; consequently there is no danger of the gas short-circuiting past the larger jets such as when the water supply is by gravity.

One advantage of having the gravity supply is that it can be used when the pump is not working. In such an arrangement care must be taken that too much water is not used in the scrubber; that is, more than can be carried away by the outlet pipe, or the water will rise too high, and the gas will be delivered to the engines in a wet state, especially if precautions are not taken to dry the gas after leaving the scrubber. Moisture when carried over by the gas is apt to cause misfiring, and will also make the valve spindles gum and stick.

The water after passing through the scrubber is drained into the pond and again passes through the same cycle. One would naturally expect the water to become foul in consequence of being so often in the scrubber, but this does not occur, because the large area of the pond allows the escape of all noxious gases, and the water is frequently reinvigorated by the addition of fresh rain water which is drained into the pond. At no time does the water become more than slightly warm. To prevent any deposit of lime from adhering to the vaporizer a small quantity of boiler compound is added to the supply at this point. Where a producer is not fitted with manholes the top of the producer must be taken off periodically and the vaporizer thoroughly cleaned. It is necessary to clean out the pond periodically, but not more frequently than once in about six years, so that the expense on that score is not very large.

The cooling of the engine cylinders is effected through the ordinary thermo-siphon system, the jacket water passing through a series of cooling tanks.



Cooling Water System

CORRESPONDENCE

Avoiding a Shutdown

Our plant contains two 150-hp. gas engines operating on suction producers and equipped with a hot-tube ignition system, the tubes being heated by a bunsen burner supplied from the city gas mains.

About 9:30 one Monday morning we were notified that the gas supply would be cut off that afternoon and as we had no other means of ignition, a shutdown seemed inevitable. After some pondering, however, I procured two old bunsen burners, fitted with needle valves and stop-cocks, and plugged up the air holes at the bottom. I then soldered them into the bottoms of two quart cans and inserted a piece of candle wick in the burner and around the base of the ignition tube.

About noon I disconnected the gas supply and attached the cans and bunsen burners. The cans were then filled with wood alcohol, a little of which was also applied to the candle wick and the latter lighted. As soon as the base of the tube became sufficiently hot, the flow of alcohol was regulated by means of the needle valve. This alcohol coming in contact with the hot tube was vaporized and burned, thus keeping the tube hot. The cans were protected from the heat by a sheet of asbestos and all the joints were carefully soldered.

The engines were run in this way for about 5 hr. and it required less than a pint of alcohol for the ignition of each engine.

S. G. Rose.

Brockville, Ont., Canada.

Tests made by Prof. R. C. Carpenter, of Cornell University, on a sun-power steam engine, using steam of about 1 lb. above atmospheric pressure and a vacuum of 28 in., showed that it required about 31.6 lb. of steam per hp.-hour.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Cost of Making Ice in Small Plants

By R. P. Kehoe

The business of ice manufacture is sometimes disparaged and manufacturers of such machinery are criticized for the apparent failure of a few plants. The cause is usually attributed to inefficient equipment or misleading statements on the part of the builders. Occasionally the purchaser is convinced that the plant is properly constructed, but believes his operators to be at fault.

In almost every instance the lack of success arises from local conditions which the most perfect plant could not overcome. This is especially true in installations of small capacity. Large plants are only installed where there is considerable demand which lasts for several months during each year, but small plants may be installed in localities having little population and requiring ice for only a comparatively short period.

In cities of any size there is a certain demand throughout the year. Hotels, saloons, butchers and the like require ice even in the coldest weather. Inhabitants of cities are also accustomed to use ice constantly. But in suburbs, country towns and villages, little or no ice is used during the cooler months. June, July, August and September are practically the only months which can be counted on to produce a maximum demand. Moreover, some days during these months will be cool and the business will fall off. Under such conditions the entire output of a plant may not exceed 30 to 40 per cent. of the full yearly capacity.

If the percentage of yearly output were precisely the same for two plants, one of large and one of small capacity, the manufacturing cost per ton should be and is less in the large plant; but usually the small plant must bear the burden of a low yearly load factor, so that it is actually necessary to make a more careful investigation of local conditions governing demand and selling price in the building of a small plant than a large one.

The daily manufacturing cost when the plant is operating at full capacity gives positively no idea of possible success; such figures are entirely misleading. Many small plants now in operation show little or no profit at the end of each year and the owner cannot understand it when the daily operating cost per ton is only about half the selling price. The cost of upkeep, depreciation and labor during shutdowns and comparatively low yearly consumption are not seriously considered.

Very often in a small business the cost of delivery and manufacturing are confused, and the fact that a loss in one end eats up part or all of the profit in the other, may never be discovered. The average owner of a small ice plant is not often an expert business man and cannot analyze the combination of manufacturing and delivery of ice with much accuracy.

The accompanying tabulation clearly shows how much the yearly consumption will affect the total cost of production. The figures covering labor and fuel represent an average and can be adjusted to suit any special locality, but the comparison is a true one in any case.

COST OF ICE PER TON IN DIFFERENT SIZED PLANTS AND AT DIFFERENT LOAD FACTORS

Capacity, tons ice per day of 24 hr.	5				10				15				20			
Total period of full operation, months	3	4	5	6	3	4	5	6	3	4	5	6	3	4	5	6
Yearly load factor, per cent	25	33	42	50	25	33	42	50	25	33	42	50	25	33	42	50
Investment																
Mechanical equipment complete		$6000				8500				12,000				15,000		
Building		1500				3500				5,000				6,500		
Total investment (excluding land)		$7500				12,000				17,000				21,500		
Daily Operating Expense																
One day engineer		2.50				3.00				3.00				3.50		
One night engineer		2.00				2.50				2.50				2.50		
One day tankman						2.00				2.00				2.00		
One night tankman										2.00				2.00		
Coal @ $3.50 per ton		3.50				7.00				10.00				13.00		
Ammonia, oil, waste and supplies		0.50				1.50				2.25				3.00		
Net operating expense per day		8.50				16.00				21.75				26.00		
Net operating expense per day per ton ice		$1.70				1.60				1.45				1.30		
Total Cost of Operation per Year																
Daily Operating Expense during period of full operation, dollars	765	1020	1275	1530	1440	1920	2400	2880	1958	2610	3263	3916	2340	3120	3900	4680
Half of labor expense for balance of year	608	540	473	405	1013	900	788	675	1283	1140	998	855	1350	1200	1050	900
Fixed Charges: 5% depreciation on cost of mechanical equipment	300	300	300	300	425	425	425	425	600	600	600	600	750	750	750	750
Fixed Charges: 2% depreciation on cost of building	30	30	30	30	70	70	70	70	100	100	100	100	130	130	130	130
Fixed Charges: 5% on total investment for repairs, taxes, water and incidentals	375	375	375	375	600	600	600	600	850	850	850	850	1075	1075	1075	1075
Total annual expense, dollars	2078	2265	2453	2640	3548	3915	4283	4650	4791	5300	5811	6321	5645	6275	6905	7535
Number of tons of ice produced annually	450	600	750	900	900	1200	1500	1800	1350	1800	2250	2700	1800	2400	3000	3600
Total cost per ton ice per annum	4.62	3.78	3.27	2.93	3.94	3.26	2.86	2.59	3.55	2.94	2.58	2.34	3.14	2.61	2.30	2.09
Minimum average selling price to earn 10 % on investment	6.29	5.03	4.27	3.76	5.27	4.26	3.66	3.26	4.81	3.88	3.33	2.97	4.34	3.50	3.02	2.69

In the table figures are given for total yearly production equivalent to three, four, five and six months' full capacity. It must be understood that no plant will simply operate for these periods during the year and then shut down. Maximum capacity is usually required throughout July and August and possibly part of June and September, but the balance of the time will only demand from 50 per cent. down to a very small percentage of the full output. The total amount, however, will in almost every instance equal the full capacity for from three to six months. A plant operating with a yearly load factor of 25 per cent. or the equivalent of three months' production may be

taken as a bad case, although this is entirely possible in certain localities, particularly in the Northern states or where ice is only required for a summer population. The equivalent of six months' production would, on the other hand, represent good conditions for an average plant.

The estimated costs of installation are fairly liberal, but can be depended upon to be very nearly correct. The building for the 5-ton plant is assumed to be of very cheap construction on account of the small size of the plant. The larger plants would need stronger and more elaborate structures, which explains why these figures are not exactly proportional.

In the 5-ton plant the engineers can fire the boiler and also pull the ice. For the 10-ton plant, one tankman is figured on to harvest all the ice during the day, as the operation can be arranged in this manner by providing a distilled water-storage tank to collect the condensate during the night. Four men are necessary for both the 15- and 20-ton plants.

During the cooler months the labor expense can be reduced somewhat and in the tabulation, this item is calculated at 50 per cent. of the full expense, which can be assumed as a fair basis. On the mechanical equipment, 5 per cent. depreciation is conservative, because it does not cover any repairs or upkeep. If this 5 per cent. is actually set aside as a sinking fund and invested to earn a normal rate of interest, it would equal the total value of the equipment in about 15 years. In addition, 5 per cent. on the entire investment is calculated for yearly repairs and incidentals.

The cost of property has not been considered and would depend upon the location. This part of the investment really need not be required to earn anything, as generally it will increase in value during the life of a plant. It can therefore be considered apart.

The last two lines are of particular interest. It may be noted that the total cost per ton of ice per year ranges from $2.09 in the 20-ton plant with a 50 per cent. load factor to $4.62 in the 5-ton plant with a 25 per cent. load factor. Furthermore, the selling price necessary to obtain to earn 10 per cent. on the investment ranges from $2.69 to $6.29 per ton. These figures may be astonishing to those who only figure on the reguar operating expense without regard to the other items, but they are nevertheless perfectly true and represent the only proper basis on which to figure profits.

This article is not intended to discourage investors for there are hundreds of very profitable plants now in operation that indicate the success of the business in general. The gradual decline of the natural ice trade occasioned by the condemning of questionable sources of supply and the growing preference for the purer artificial product, promises a still better outlook in the future. A few failures, however, are often considered to the disadvantage of the entire business and a more universal knowledge of the true circumstances should help the situation.

CORRESPONDENCE

Ammonia Absorption Refrigerating System

In the Sept. 24 issue of POWER, Henry Torrance, Jr., published a short discussion on my article on this subject in the issue of Apr. 23, so that it is desirable to add a few words here to complete the data originally presented.

The article was not written to present actual figures of the performance of each apparatus of the system, but to explain in detail their functions, and the figures served to describe them more clearly. As stated in the article, the amount of heat eliminated in the rectifier was determined under the assumption that the partial pressure of the water vapor in the gases leaving the analyzer corresponded to the water-vapor pressure at the temperature of the gases. Lacking more exact information at the time, this method of figuring was deemed sufficiently accurate for the purposes of the article. In view of Mr. Torrance's statement additional data may be of interest.

The partial pressure of the water and ammonia vapors in the mixture of gases evolved from aqueous ammonia solutions were investigated by Perman, who used for his experiments solutions of varying strength ranging from 4 per cent. to a maximum of 24 per cent. of ammonia dissolved in the water. The writer substituted the values found by Perman in the following formula to determine the molecular rise in the boiling point of aqueous ammonia solutions for the concentration used in the calculations.

$$E = \frac{\Delta t M}{m}$$

Where E is the molecular rise in the boiling point of the solution, Δt the observed rise of the boiling point, M the molecular weight of the dissolved substance and m the weight of the substance dissolved in 100 grams of the solvent. It was found, as was to be expected, that E increased with the concentration from 1.5 for a 3 per cent. solution to 4.5 for a 24 per cent. solution, that is to say, the stronger the solution the smaller will be the proportion of the solvent vaporized on boiling. For the calculations used in the text to illustrate my article, the strength of the liquor varied between 18 and 35 per cent. Taking the average concentrations of the strong and weak liquors as a basis, the average rise in the boiling points of the various solutions can be found.

The hot vapors ascend to the analyzer in which they are cooled by the rich liquor coming from the exchanger, and a change in the proportion of ammonia in the vapor takes place. From the temperatures of the vapor leaving the analyzer and leaving the rectifier, the quantity of water vapor that is condensed and of the ammonia vapor that is reabsorbed, can be determined. These figures used in connection with the latent heat of steam and heat of absorption of ammonia for the temperature and pressure conditions existing will give the quantity of heat rejected in the rectifier.

Recalculating the heat rejected in the rectifier by using the above quoted formula as a basis of final figures, of the steam consumption in the generator per hour per ton of useful refrigerating effect, the following results were obtained:

Test No. 1	Test No. 2	Test No. 3	Test No. 4
41	52	35	26.4

These results, although lower than those given in the article, still show the same change in the steam consumption for varying conditions of operation, the steam consumption increasing when the temperature in the refrigerator decreases.

The water consumption was obtained from the actual tests of the particular plant described, and while it may be higher than that of some other plants differently arranged, due consideration was given to this fact in the calculations. The use of large quantities of water for cooling purposes is an advantage when it can be secured cheaply.

It is not proper to compare either the steam consumption or the water used by an absorption refrigerating machine with that of a compression refrigerating machine operated by a simple Corliss steam engine, because the quantities of heat carried off by the water are different in these apparatus. If the actual amounts of steam and cooling water necessary for the proper operation under the same conditions of an absorption refrigerating system and a compression refrigerating system operated by a compound condensing Corliss engine are compared, the results obtained will be of some value.

In view of the above, the final conclusions drawn from the quantative analysis of the absorption system, as given in my article, still stand.

FRED OPHÜLS.

New York City.

Another Diesel engine motor-propelled ship is to be built for service on the Great Lakes for the Montreal Transportation Co. *The Engineer*, London, says the vessel is to have two sets of Diesel engines, each of 300 hp., running at 400 r.p.m. Each engine will drive an alternating-current generator and the current will be delivered to an induction motor on the propeller shaft running at 80 r.p.m. The vessel will have a capacity of 2400 tons on a draft of 14 ft.

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Sworn to and subscribed before me this 30th day of September, 1912.

Wm. W. Mayer,
[Seal.] Notary Public
(My commission expires Mar. 30, 1913)

For the contents of this issue, refer to the Outline of Topics, page 62.

POWER

Defective Boilers

Regarding boiler explosions, their cause and prevention, the equivalent of several large volumes has been published in POWER in recent years. Engineers and firemen have been advised as to their care and operation, but in spite of all that has been written and notwithstanding that boilers are better built and cared for than formerly, between thirteen and fourteen hundred serious boiler accidents occur in the United States every year. Of these, according to William H. Boehm, in a recent lecture, abstracted on page 548, between three and four hundred are violent explosions, in which between four and five hundred persons are killed and from seven to eight hundred more are injured, and more than half a million dollars worth of property is destroyed.

During the past ten years 1,101,140 boiler examinations have been made by one boiler-insurance company, out of which 140,989 were reported defective, many of which were considered dangerous, and one boiler out of every eight examined contained defects of such nature as to be reported by the inspector.

These figures are of such magnitude that those responsible for the upkeep of steam boilers should exert greater vigilance in their examination and know, insofar as it is possible, that defects do not exist.

If one boiler out of every eight examined is defective in one way or another, then the ratio is greater, when the total number of boilers in daily operation is considered. Thousands of them are never examined by other than the men in charge, who, in many cases, are not qualified to intelligently determine whether a boiler is defective or not.

It goes without saying that many such boilers are operated with more or less serious defects existing which would be detected and ordered repaired if proper inspection were made.

If a boiler is used in a manufacturing establishment, the workmen are exposed to a hidden danger of which they are ignorant and against which they cannot guard. The engineer, therefore, has a double responsibility, because he not only is expected to safeguard the boiler, but the workmen around it.

If he is incompetent the responsibility should, but seldom does, fall upon the owner. If, for the sake of saving a few dollars a year in wages, he hires a man incompetent to intelligently care for the boiler, he should be held criminally negligent if an explosion does occur.

Boilers are dangerous enough at best, the men operating them are often inclined to be careless and allow defects to remain, and the owners, judging from the boilers found in operation, are indifferent as to their condition.

It is about time for everybody concerned to wake up to the gravity of the situation.

The Engineer's Duties

Speaking about putting worn apparatus in working condition, an engineer made the remark that he did not "fool" with anything of the kind because the company he worked for did not appreciate it. They only paid him for operating and not for the time spent in "fixing things."

Is that the right attitude for an engineer to assume, even if the management does not seem to notice those little extra things he does about his engine and boilers, etc.? By doing willingly such little things he shows an interest in the upkeep of this end of the plant that is much to his credit. It does not speak well for an engineer if he allows "junk" to accumulate, and when an injector, valve or anything of like nature is disconnected, because some part is worn or broken, he should put it in shape, or have it done as quickly as circumstances will admit, when it can be laid away for future emergencies.

Such a rule, if adopted by all engineers, would materially assist them in their work, at times, as well as save time and money for their employers.

This is not to say that any engineer should take the responsibility, or more properly speaking, the burden upon his

shoulders of doing all the repairing, whatever the size or nature, required in the plant, but that he should give attention to little things he can accomplish unhampered while on duty, and leave such work as is justifiable to a machinist.

An engineer should not do heavy repairing, simply for the reason that he is not paid to do it, taking into consideration the general wage received by engineers, his duty ends with operating, and seeing that things are always kept in the best operating condition, but this may require that he do many little jobs without the aid of an outsider. In so doing he develops his own resourcefulness and shows his ability to advantage.

The foregoing applies only to the average plant. In the larger ones things are carried out more systematically.

While there is a limit to the amount and nature of repairs that should be done by an engineer, he should not be too ready in "drawing the line," as an old chief puts it. It requires judgment to know where to begin and stop such work.

Scarcity of Elevator Literature

Nearly all, if not all, the various branches of power-plant engineering are well represented in engineering literature except that pertaining to elevators. By this statement we do not mean to infer that the few books devoted to this subject are unworthy of commendation. We do deplore the lack of such literature.

In modern city life the elevator is an extremely important factor. It is said that more people travel on elevators in New York City in one year than in all the subway, surface and elevated railroads combined. The problem of safely carrying so many passengers is serious.

In large office buildings the care of the elevators is only a part of the engineer's duties. True, the watch engineers in the very large office building plants have little to do directly with this work as it is entrusted to the care of a foreman and crew who do nothing else. But, the chief engineer is responsible to the management and the public for his foreman, and watch engineers are future chiefs.

If it were not for the work of trained inspectors of the insurance companies the number of fatalities due to wrecked elevators would be startling. This is no reflection upon makers of elevators or the men caring for them. The latter do well —the wonder is that they are so successful. In seeking to learn more about their work these men look for books covering the subject, and are astonished to find that there is no complete work available. The elevator makers are none too generous or responsive to requests for information concerning their machines. There is one commendably complete book on hydraulic elevators, but, to the best of our knowledge, none thoroughly treating electric machines. Some engineering books make a pretense of treating them, but the information given is far too meager to be of much value.

We were fortunate to obtain and publish during the years of 1908 and 1911 inclusive, a series of articles on electric elevators by the late William Baxter, Jr. This series made up the most thorough and practical treatise on the subject ever published. But this is not enough. A treatise on elevators embracing all kinds is sorely needed.

A Plan Worth Adopting

With the approaching season many engineering organizations will begin some sort of an educational program for the year. Here is a plan that all such bodies might advantageously adopt with confidence that much benefit will be realized.

Number Three Association (Newark, N. J.) of the National Association of Stationary Engineers has delegated two of its members to make plans and specifications for modern power plants. One is to cover the requirements of a high-class office building which, among other essentials of equipment, will have a refrigeration plant, coal and ash conveyors, vacuum-cleaning apparatus and the highest type of air purification and ventilating system. The other will design a first-class factory power plant. Each man may ask the collaboration of any others, but the aim is to make the work solely that of this local association.

Designing power plants is splendid training for operating engineers as it brings them to understand the logic of many things they had come to regard as matter of fact from long familiarity. There is every reason why a first-class operating engineer should be able to design a plant that would prove thoroughly practical and economical in service. His experience in operating power-plant equipment is very helpful to him as a designer. Making the specifications would give him a training in detail that should be of inestimable value in his regular work.

It is to be hoped that other associations having to do with power-plant practice will take some suggestions from this plan and incorporate them in their winter's educational program. That the benefits from it may be participated in by all the members, it would seem a good suggestion to have lantern slides made showing the work as it progresses and keep notes of the changes in the plans and the reasons for making them. Those in charge of the work could then give illustrated talks before all the members and thus insure the fullest realization of the purpose. It would make the plan still more fruitful of good results if a number of similar associations would engage in a friendly competition to see which could produce the best plans and specifications. Judges could be appointed to pass on the entries in the competition and award prizes for the best work similarly as in intercollegiate debate or athletic meets.

Smoke Abatement

The monthly meeting of the New York section, American Society of Mechanical Engineers, on Oct. 8, is to be devoted to a discussion of smoke abatement, and representatives from other cities will report on what measures are being taken to handle this problem in their localities.

While the smoke nuisance in New York has not become as grave as in some other cities because of the general use of anthracite, still considerable bituminous coal is burned and without regard to smokeless combustion. In proof of this, one has only to survey the city from the top of a high office building on a humid day. As already mentioned in these columns, New York City has no separate smoke-inspection department, this duty being performed by the Health Department, but it seems to have difficulty in obtaining the conviction of offenders.

Heretofore, there seems to have been a tendency to side-step the issue on the grounds that the situation was not acute enough to warrant agitation. The fact remains, however, that with proper handling, bituminous coal can be burned without obnoxious smoke, and it is the duty of the community to see that such measures are enforced. In taking the initiative the local committee of the American Society is to be commended.

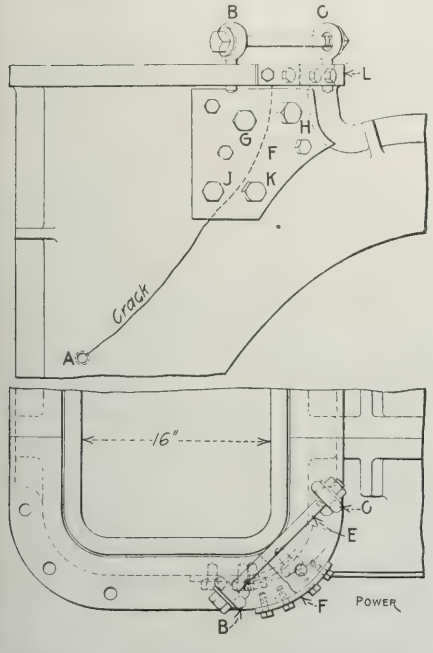
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Patching a Pump Case

Owing to the failure of a wharf crane, a new pump case was dropped several feet and cracked badly at the throat, as shown. It was a bad spot for a crack, as the case joints were overhung; that is, the joint was inside the bolt circle and screwing up on the bolts tended to open the crack still further. To be of any use the patch had to be air-tight. The repair was made as follows:

The full extent of the crack was determined by first rubbing the surface of the casting with waste soaked in kerosene and then wiping it clean and rubbing



How the Pump Case Was Patched

chalk over the face. The crack appeared plainly under the chalk. The lower end was drilled for a ⅝-in. screwed plug *A*.

As the top of the crack had sprung open $\frac{3}{32}$ in. and the case was 1⅜ in. thick it was a large crack. A couple of eye-bolts *B* and *C* were forged and the shanks turned 1¼ in. diameter to just drive lightly into the holes in the case flange. A 1-in. bolt *E* was passed through the eyes and washers and tightened until the crack closed.

After being bent well to shape and served with red lead paint, a plate *F* ½ in. thick was put on with capscrews, as shown. The holes *G, H, J* and *K* on each side of the crack were countersunk on the outside and the screws used on them were taper-headed at the bottom of the hexagon to fit. A slight draw was allowed on these case holes so that the screws drew the crack shut when pulled tight. A supplementary capscrew patch *L* was put on the edge of the flange also. This completed the job. A larger patch would have been better, but the above was perfectly satisfactory under working conditions.

J. McA. Howden.

Melbourne, Australia.

Boiler Water Impurities

Much has been said about the way we should obtain a perfect analysis of our boiler scale, feed water and the manner of feeding compounds and the proper kind of compound for each individual case to be treated. It seems to me that the first thing to be considered is the proper removal of the scale from the boiler. The treatment of the water before it enters the system is important, but the purifying effects of the feed-water heater and the mingling of the returns with the feed water before entering the boiler should not be forgotten. The scale should first be removed before introducing a compound.

Let us trace the water taken from a running stream. It is pumped to a reservoir and the mud is settled, nearly all being left in the reservoir. It leaves the reservoir and enters the open heater and, by being heated to 200 or 210 deg., most of the foreign matter is caused to adhere to the interior of the heater. As the heating takes place, another change of great importance, often overlooked, is the addition of perfectly pure water in the form of condensation, which in some cases forms a large part of the entire feed water leaving the heater. Then the steam-trap returns are also very important. If there is a purifier (scum-cock) on the boiler it will remove about one-sixth to one-eighth of the impurities if properly used, after the heater has removed three-fourths. Thus, the water enters the boiler very nearly pure.

Now, if the boiler has a general deposit of old thick scale the thing to do is to remove a large thick piece and have it analyzed. It might be well also to send a gallon of the water from the suction of the feed pump. I never feed compounds into the water before it enters the heater, as I believe it interferes with the purifying effect of the heater.

Ray Gilbert.

Virginville, Va.

Care of Electric Elevator Brakes

This is a summary of my experience with the troubles and dangers to the brake of a common type of electric elevator.

When the starting lever in the car is moved to the up or down motion, it sends current through the elevator switchboard, through the circuit-breaker and the soloneid *B*, which draws up *C* to *D*, compressing the spring and slackening the brake bands as shown dotted. When the controller is in the "off" position all current is cut off from the solenoid. Then the tension of the spring causes the bands to grip the brake wheel as shown by the full lines.

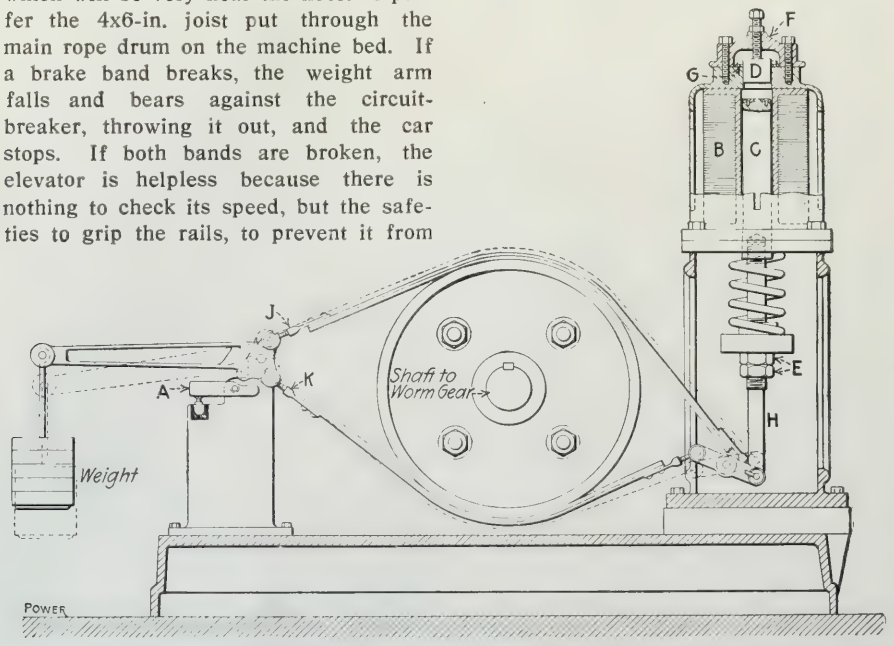
My greatest trouble with these elevators, which run 415 ft. per min., and in fact all high-speed electric elevators, is to make a good stop. The stop should not be too abrupt and yet the car should not slide, as both are equally dangerous. If the stop is too sudden it is likely to set the safeties, and if too much sliding occurs the car will go by the floors, causing a restarting which uses up more current than when running three stories. If the car slides excessively it is likely to go to the roof and against the limit switches, cutting off all current by throwing out the circuit-breaker *A*, and if the counterweights are slightly long, they will pound hard on the floor. When the stop is too abrupt, a piece of yellow soap held against the brake wheel is a good remedy; it will keep the car from sliding. Powdered rosin is good in extreme cases, but good judgment must be used with both. Elevator inspectors do not approve of using anything on the brake wheel.

With speeds of 250 ft. per min. rosin or soap is unnecessary, but with a speed of 415 ft. the brake band will get so hot it will smoke when the trips are frequent and the brake will slide as though greased.

When the foregoing remedies do not correct the trouble it is necessary to change the brake connections by screwing up on the nuts *E* under the spring, giving more brake friction, which in time, due to the leathers wearing, settles down and the plug *C* drops away from *D*, leaving too much space between them. The result is that the magnetizing effect is decreased and the brake bands will drag. The remedy is to take off the top *F*, raise the plug *D* and put thinner

liners in the place of those shown at *G*. This brings the plug *D* nearer plug *C* and increases the drawing power to lift *C*. Care should be taken not to get *D* and *C* too near together, as there will be no chance for *C* to lift because of coming in contact with *D* before the brake bands have lifted away from the brake wheel. The main switch should be pulled before attempting the foregoing.

Another bad condition is when the rod *H* strikes on the bottom bed, thus holding the brake bands loose. This may be caused by thin leathers, or the adjusting ends *J* and *K* being let out too far. The remedy is to take up *J* and *K* equally and renew the leathers. In most cases the rod *H* will have to be screwed into *C* to maintain the original space between *C* and *D*. Before doing this be sure to block the main rope drum by putting a 4x6-in. joist through the spokes. Then by letting up on the brake the car will not go flying to the roof, as it would do because of the action of the back drum counterweight. Another way is to run the car to the roof and pull in the safety, thus holding it. Still another, when the car is at the top, is to put a 6x6-in. block under the bottom counterweight, which will be very near the floor. I prefer the 4x6-in. joist put through the main rope drum on the machine bed. If a brake band breaks, the weight arm falls and bears against the circuit-breaker, throwing it out, and the car stops. If both bands are broken, the elevator is helpless because there is nothing to check its speed, but the safeties to grip the rails, to prevent it from racing to the roof with a light load, or racing to the bottom if heavily loaded.



Showing Construction of Typical Electric Elevator Brake

After changing the band connections at *J* and *K* there is probably either a sudden stop on the up motion and a sliding stop on the down motion. One would think that by tightening one connection the friction would be equal, but such is not the case. To loosen the connection to a solid stop on the up motion, the top brake band must be loosened by taking out the bolt and giving it a half turn and if sliding on the up motion occurs the top band should be tightened; if the down stop is too abrupt loosen the bottom band.

To put on brake leathers select a piece of fine grain leather about ¼ in. thick and of even thickness. Rivet one end to the band first, then press it down hard on the iron band to get the holes marked on the opposite end. The other holes can be bored by putting the drill through the holes in the iron brake band. The leather holes should be countersunk ⅛ in. for rivet heads.

Revere, Mass. A. C. Waldron.

Boiler Explosions

Permit me to commend the article under the above heading by S. F. Jeter in the Mar. 26 issue, and the editorial comment on the same in the Apr. 9 number which was also interesting. All should particularly read the two articles above mentioned, together with "Uniform Boiler Specifications" in the Apr. 2 issue.

I like a suggestion I found in the latter article, which read "Institute an educational campaign" relative to boiler construction and attachment of appurtenances, etc.

With the high pressures now employed I believe a more radical departure from old times should be made than is being done just now. What is needed most is a gradual reduction in diameters, and all flat surfaces eliminated if possible. In firebox boilers it is usually the crown-sheet which gives way, while in water-tube types it is the drum-heads.

By unequal expansion when under steam, it may be all this part of the boiler can do to withstand the pressure until finally a slight jar from water-hammer, etc., may cause it to let go when it is least expected.

I think the Jacobs-Schupert firebox an ideal departure from the old stay-bolt method of firebox boiler construction judging by the report of tests made on it, since it makes this part of the boiler practically as strong as the shell at all times.

To keep track of the age and firm turning out boilers, it would be well to require that an indestructible plate be riveted on boilers giving the year made, whom made by, strength of material, etc., or if objection should be made to this plate, the same might be stamped upon the boiler in some conspicuous place where it would not become obliterated, as prescribed by some state laws.

While I am upon the subject I wish to ask, if it would be of any benefit to "draw" boiler shells as seamless flues are drawn?

Lloyd V. Beets.

Nashville, Tenn.

[The contributor's question relative to the "drawing" of boiler shells is indeed interesting. We would be pleased to publish good, live comment on it in which should be stated the mechanical difficulties to be met with in doing this work and the probabilities of it as a commercial success.—Editor.]

Valve Guide in Pipe Damages Engine

A serious accident happened to the high-pressure cylinder of a Corliss engine I have under my supervision. The piston was broken into about 150 pieces and both cylinder heads seriously fractured. The damage was caused by a valve guide of a 4-in. globe valve, which is an auxiliary valve placed in the expansion loop above the engine. The valve guide had been broken off sometime previous to my taking charge of the plant and had lodged in the steam chest; otherwise it would have been noticed as I have had the line from the valve to the engine all apart repairing leaks.

The crank end head broke first and I was trying to throttle the engine, but before I could do so I had to jump to save myself from the flying fragments and escaping steam, and shut the stop valves on the boilers to save the siphoning of the water from the boilers to the air compressors and other engines then in operation. I hope that any engineer missing part of a valve or other fitting in a steam line will take heed and, before damage results, find where the part has located itself. Should this case have had that attention, I would not have had this experience.

I had considerable trouble with a dashpot on the high-pressure cylinder during my first week in charge of the plant. The plunger did not fit snug enough. I found by putting oil around the plunger it would work and with a very strong vac-

uum. I then tried graphite and cylinder oil, putting it on with a brush before starting each day, and this ended my dashpot troubles.

JOHN T. CHAMBERLAIN.

Irvington, N. J.

Jointing Materials

Conditions of use determine the choice of a jointing material for any particular purpose. The condition of flanged faces to be brought together to stand various pressures and the nature of the liquid or gas under pressure, together with temperature, are the chief factors determining what to use.

In early steam-plant practice when 5 to 10 lb. pressure was a maximum and when the engineer's chief difficulty was to prevent a vacuum in the boiler, oakum was the material largely used. A leak in a pipe line has many times been stopped with a piece of canvass and tarred yarn when the pressure did not exceed 10 lb.

Later, with faces perfectly scraped to surface plates, a wipe of tallow or boiled oil to prevent rust was sufficient to rigidly couple two cylinders together without leakage. Now the tendency is to consider the jointing material and finish surfaces accordingly.

Where cold water under pressure only has to be considered, ordinary hard woven paper millboard of suitable thickness makes excellent joints and is largely used. The surfaces in this case are left as machine finished with a coarse feed. Fire-engine makers use a more expensive material—leather—for covering joints, this, needless to say, makes first-rate joints, but is expensive.

My opinion, borne out by considerable experience, is that clearance volume in steam cylinders is frequently excessive with consequent loss of economy, due to bad jointing. I have seen cylinder covers in a big job with joints ¼ in. thick under them. A distinct improvement in economy can be made by looking to this point and substituting a thinner material. The designer as a rule does not take this fact into consideration when figuring the probable results of a new engine over the drawing board. With a triple-expansion engine of 2000 hp. a good many extra tons of coal can be wasted annually by thick cylinder-cover joints. Broad face cylinder-cover joints, where the pressure does not exceed 80 or 100 lb. with the surfaces in good condition can be efficiently made with thick brown paper steeped in boiled oil. Above this pressure the thinner asbestos millboard which will make a tight joint should be used, or one of the numerous patented jointings of the thinnest character. Much depends upon the closeness of pitch of the studs. It is better that these be smaller in diameter and closer together than of large size coarsely pitched.

With higher pressures like those used in freezing machines, it has been found advisable to use metal-to-metal joints, with flanges having grooves to take a solid unbrazed ring of annealed copper, which forms an ideal joint.

Corrugated gaskets and red lead make good joints, but care must be taken that the gasket is the full size of the flanges. If simply put inside the bolt circle the flanges suffer distortion and are a nuisance every time they are uncoupled and again have to be jointed.

Thick joints are only necessary where the surfaces are in bad condition. The worth of a jointing material depends upon its elasticity under the conditions of its temperature and use. As a rule cover joints of any description are not made to be permanent and where studs are used, excessive tightening down stresses them unfairly. Rather than attempt to squeeze too tightly when making cover joints with sheet material to hold steam, it is better to go over the bolts and tighten them after the joint is warmed. The least pressure possible to just stop leakage is the desideratum.

To satisfactorily discuss the various patented jointing materials and their claims would require considerable space and serve no useful end. They are legion and in the main serve the purpose for which they are made; but in some instances an ordinary material, at a fraction of the cost, would serve many purposes for which expensive jointings are used. Common sense is useful in this matter as in every part of engineering and I take it that jointing with brains is as useful as designing with brains; it is possible to save money both ways.

A. L. HAAS.

London, England.

Carelessness Causes Pump Accident

The chief cause of the accident herein described was due to the fact that "somebody forgot."

A small gasoline engine was used to drive a triplex power pump used to pump oil. Repairs were being made to it by the engineer and a helper. This helper, with all good intentions, threw in the clutch to turn the pump over to the most convenient position for disconnecting the connecting-rods from the cranks. After disconnecting the rods at the cranks only, he walked away leaving the clutch in.

Later the engineer, not knowing the rods were disconnected or that the clutch was in, flooded the vaporizer, threw in the switch and cranked the engine. At about the time he realized the clutch was in, the engine "kicked off." As the cranks revolved they struck the disconnected rods, causing serious damage to the pump.

It pays and pays well to look a machine over carefully before starting it.

J. P. COLTON.

Ohio City, Ohio.

Books for the Engineer

Just what books engineers should have is open to a lot of controversy.

Many older engineers do not believe in the book-engineer or his books, but this is not the case with the men that are coming into the business now. Here in Massachusetts, where the license law forces the engineer to study, a good book is always appreciated. I have personally invested more than $300 in technical books in the last 10 years, a great many of which have been useless to me. I have never, however, found any difficulty in disposing of these latter books, and the fact that a book, or for that matter anything, is of no value to me does not mean that someone else cannot find a use for it. I have found books also that I would not part with under any conditions.

The following comprises a library that from my viewpoint is invaluable to the operating engineer. If I was sure that I could not replace it, a week's pay would not tempt me to part with one of these books.

Each of the following books paid for itself before it had been in my possession six months:

Book	Author
Boilers	Power Hand-Book
Power Catechism	Power
Steam Heating and Ventilation	Technical Pub. Co.
Pipes and Piping	Power Hand-Book
American Engines	T. Hawley
Valve Setting	Collins
Elements of Steam Engineering	Spangler and Green
Steam Engine Indicator	Low
Shafting, Pulleys & Belting	Power Hand-Book
Steam Turbines	French
Hydraulic Elevators	Baxter
Pumps & Hydraulics (2 vol.)	Rogers
Condensers	Low
The Slide Rule	Ramsey and Hinkley
Practical Talks on Electricity (2 vol.)	Baxter
Motor Troubles	Raymond
American Machinist Hand-Book	Colvin and Stanley
Millwrighting	Hobart
Mechanical Drawing	Willard
Automobile Hand-Book	Homans
American Cyclopedia of Engineering (7 vol.)	Technical Pub. Co.

It will be noted that I have left out such an important subject as refrigeration. I have never found a book that to my mind thoroughly covered the subject. As regards electric elevation, I have Baxter's articles on this subject published in POWER some two years ago, and I have never seen a book that covered the subject as well. As the tendency seems to be toward a more general use of the traction type of electric elevators, Mr. Barter's articles made up in book form would be highly desirable.

As far as compressed air is concerned, the Encyclopedia of Engineering covers the subject well enough to supply my needs.

A. D. PALMER.

Roxbury, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Counterbalancing Elevators

The following may give the information asked by L. A. Trowbridge in the Sept. 10 issue on the counter-balancing of elevators and the care of electric elevators.

The function of counter-balancing is to minimize the power necessary to operate the car. As counter-balancing depends to a great degree upon the gearing used, this subject will be briefly considered.

The gearing of an elevator is essentially the same as the arrangement of the common rope and pulley block. A one-to-one gear shown in Fig. 1 allows the car *A* to travel the same distance only as that of the piston *B*. Such a gear is really of no practical value on account of the limited travel of the car compared to that of the piston. The height at which the stationary sheaves *C* must be placed is also a serious objection.

Fig. 2 shows the six-to-one gearing of an inverted plunger, high-pressure elevator commonly used in modern high buildings. Here the cylinder *E* is short and the plunger travel very small compared to the rise of the car. For a plunger travel of 5 ft. the car will travel 30 ft., the motion being multiplied by the passage of the cables over the movable sheaves *F*, *G* and *H* and over the stationary sheaves *J*, *K* and *L*. Gearing in practice runs from 2 to 1 to 8 to 1, according to the height of the lift. To find the gear of any elevator, add the number of cable lengths on both sides of the movable sheaves. Gears are made even to avoid an undesirable arrangement of the sheaves; the three-to-one gear being the only exception.

Most of the counter-balancing weights are located in the frame which guides the traveling sheaves and assist in raising the car. The empty car must weigh enough to pull up the plunger, sheaves, frame, etc., and also to overcome the friction of all the moving parts. Its weight when empty, is more than enough to do this, however, and, instead of making the car lighter, the weights *M* are loaded on the plunger.

It is here that the gearing determines the weight necessary to counter-balance the car. If it is a two-to-one gear, to balance 1 lb. of car will require a 2-lb. counterweigh at *M;* if a four-to-one gear, 4 lb.; a six-to-one gear, 6 lb., and so on. To minimize the weight necessary at *M*, a sheave *N* over which passes a cable connected to the frame of the car and on which hangs the independent counter-balance weight *O* is provided. As this weight is geared one to one with the car, 1 lb. of weight therefore balances 1 lb. of car. This weight is small compared to the weight loaded on the plunger. The reason for this is that a heavy weight at *O* would have momentum enough at high speeds to teeter the car up and down considerably and also put severe stress on the cables when the car was suddenly stopped. This does not apply to

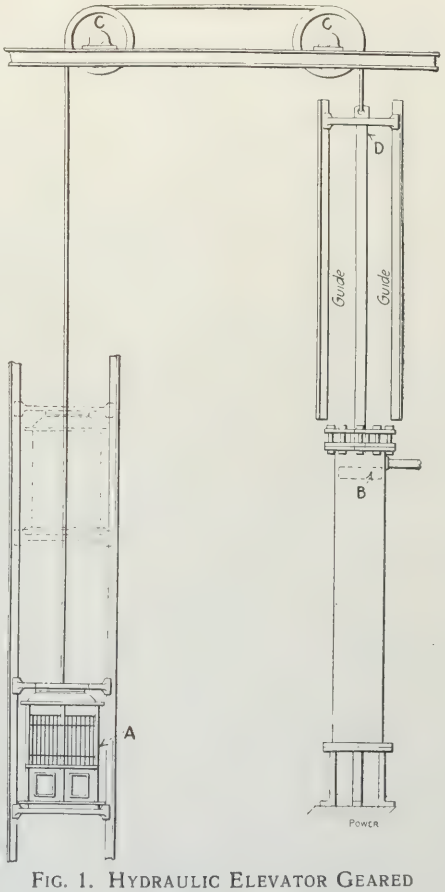


FIG. 1. HYDRAULIC ELEVATOR GEARED ONE TO ONE

the weights when placed at *M*, even though the weight is much more, because the force must act through the gearing before it affects the car. The cable carrying the independent weight *O* also aids in making the machine safer as this cable takes some strain from the other lifting cables. Some installations do not have the independent weight.

Besides the car, the cables in the shaft above the car must be counter-balanced. This may be clearly understood from the following:

Suppose the unbalanced weight of the car when it is at the top is 400 lb., then when it is at the bottom the weight of

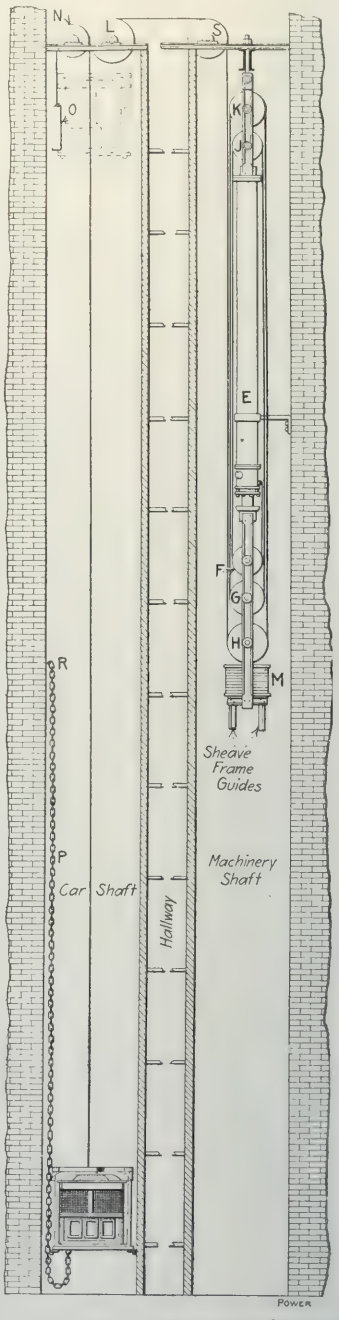


FIG. 2. HYDRAULIC ELEVATOR GEARED SIX TO ONE AND COMPLETELY COUNTER-BALANCED

the cables in the shaft must be added to this 400 lb. These cables are 5/8 in. in diameter and weigh approximately 0.8 lb. per ft. As there are usually four

or six of them, their weight becomes important, especially if the lift is high as it is in most city buildings.

It has often been suggested that the chain P, which is used to counter-balance the cables and which is attached to the bottom of the car, be connected at the other end to the independent weight O. This would reduce the length of chain, for when the car is at the top the weight would be at the bottom and *vice versa*. This is not wholly practical as there is danger of serious accident, due to the weight getting entangled in the equipment in the shaft. By attaching the chain a trifle above the middle of the car travel, as at R, it hangs from R when the car is at the top. The chain must weigh as much as the greatest length of cable in the shaft added to that of the cable length on the machinery side of the sheave S and from S to H when the car is at the top of its travel.

The care of electric elevators may be summed up briefly as follows:

Keep all contacts clean and all guides, sheaves, governor, worm-gear, etc., well oiled. Do not allow too much air gap in the magnets. Avoid all unnecessary and uncalled for sudden stops. Open the main switch or circuit-breaker if the car is to lay idle even over night; always do this when making repairs of any kind. Inspect the safeties and all contacts at the magnet control board frequently. Obtain wiring diagrams of the various circuits and learn them thoroughly. One of the best things to remember is not to pull the whole machine apart just because it will not start; inspect the contacts first as it is there trouble usually begins.

L. E. Tavor.

Arlington, N. J.

Mr, Trowbridge should particularly care for the commutator, knife switches and magnets on electric elevators as they require most attention. Where the motor-armature shaft is directly connected to the worm-gear shaft, the armature is reversed in rotation, which is severe on the commutator and brushes, and all troubles common to motors must be met. The magnetic brake on the armature must not be allowed to get out of order, the leather shoes should be renewed if the rivet heads show any sign of striking the wheel as they will cause the brake to slip. The oil in the gag-pot must not be so thick as to retard the action of the brake, nor so thin as to make the brake act suddenly, as this would cause unsteady running of the car. The density must be selected to meet the temperature of the room and in accordance with the conditions under which the machine is operated.

The starting rheostats and automatic or series magnetic controllers must all be kept clean, dry and not allowed to collect dirt or grease around their bases or they will become grounded and heat or burn out. No one special part or parts of these machines requires extra attention, but they must all be gone over regularly and sandpapered or oiled as the case may require. A general inspection of the safety devices on the cars each morning is the only precaution one can take to safeguard the public who must use the elevators.

R. A. Cultra.

Cambridge, Mass.

There are two general types of electric elevators, the worm-gear and the traction type. The traction type is divided into two classes: the full traction and the two-to-one type. In the full traction elevator there are two drums, one directly connected to the motor shaft and one just below it. The cables from the car pass over the motor drum, around the lower drum, up over the motor drum and down to the counterweights. In this type of elevator a small drum is used with a very slow-speed motor, running from 60 to 90 r.p.m. and as the cables only make one wrap around the drum the height of the building does not affect the size of it.

In the two-to-one type the cables are fastened at the top of the shaft and drop down and around a sheave on the top of the car, then up and around the motor drum and down to the counterweights. The speed of the car in this type is only half the speed of the cables. The worm-gear type is too common to need description.

The several ways of counter-balancing include that in which the car is fastened to one end of the cables and the weight to the other as in the traction type. The drum type has two weights, one connected by cables to counter-balance the drum while the cables from the other weight pass over the drum and connect with the car. The car weights should be placed above the drum weights. If they were placed below and the cables holding the drum weights should break the added weight might rush the car to the top of the shaft with serious results.

In high buildings, to overcome the weight of the cables and still keep the car balanced, chains are used which are connected to a point midway in the shaft and the other end to the counterweights, or the chain may be connected from the car to the counterweights. As the position of the car changes the length of chain hanging from it changes and thus makes up for the weight of cables lost by the car ascending. In some cases the chain is connected to the bottom of the car and the other end to the bottom of the shaft. As the car raises, the chain is picked up and having the same weight as the cables, keeps the car balanced at all times. The car is over-balanced to about the weight of the average load. Too much counter-balancing causes the car to jump when it is stopped or started. To prevent the rattling of the chains a light rope is threaded through the links, being cut into lengths of about 12 ft. to facilitate the work.

The controller is usually a magnetic type or one operated by a solenoid which makes connection by a carbon brush passing over copper contacts. The more frequent troubles with the latter type are dashpot troubles, the valve in the bottom of the pot being caught open, allow the arm to raise and cut out the starting resistance so quickly that a fuse is blown, the dogs set if on the down trip, and in any case the passengers are given a severe shaking. If the starting arm is caught up by rough contacts the same thing will happen. The dashpot, valve and contacts should be cleaned, and it should be seen that the arm works freely.

Another trouble occurs when the contacts that close to open the brake, if an electric one, do not make good connection and release the brake. They should be cleaned and the brake should open just after current begins to flow into the armature. If the brake opens too early it may, with a heavy load, allow the car to settle a little before the armature current is sufficient to raise the car. If the brake is set to open too late, it causes the resistance to heat and the car to start with a jerk. If the brake is mechanically operated from the car care must be used in setting it as with new leathers the brake may be late in opening.

In the magnetic type of controller when the first contact is made the circuit is closed through the lifting magnet of the second contact and so on. When a contact fails to close it is customary to look to the contact ahead of it. The circuits in a magnetic-type controller are usually divided and fuses placed in each circuit; these sometimes blow and cause that particular contact to fail. The most common trouble arises from poor contacts and is due to loose connections, burned contacts or dirt getting between the connecting surfaces. The limit switches in the shaft being seldom used may become stiff and refuse to close. The slow-down device on the top of the car may stick open and prevent the car moving. The small cables connecting the car-operating switch and controller may be broken and, as there are usually several dead conductors in the cable, one of them may be used to replace the broken one. The easiest way to find the ends if they are not marked is with a bell and battery.

Nearly all the motor troubles are caused by dirt getting on the commutator or around the brush-holders and care should be taken to keep them clean. Connections sometimes work loose and cause much trouble. The entire equipment should be gone over at least once a day as nothing will injure a building's reputation, especially if it is an office building, so quickly as poor elevator service. Care

should be taken that all grease cups on the sheaves are working and that there is sufficient oil in the worm case. (Castor oil is recommended.) An occasional coating of raw linseed oil and plumbago will lubricate the cables and keep them from rusting.

THOMAS H. WATSON.

Chicago, Ill.

[Mr. Waldron's letter on page 540 of this issue should be read in connection with the above letters.—EDITOR.]

Steam Ejectors on Shipboard

It is not very creditable, as intimated in POWER's editorial "The Inconsistency of the 'Titanic' Disaster," to Twentieth century engineering to have a large liner slowly sinking because the water could not be pumped out fast enough, while, at the same time, her huge engines were shut down, the fires banked, and boilers blowing off because there was no provision to utilize this enormous amount of energy for pumping purposes.

Separate reciprocating pumps would, on account of their huge size, be out of the question, and centrifugal pumps mounted on the main engine shaft would possibly be open to the same objection. There still remains, however, the lightest and simplest of all known pumping schemes—the steam jet. It takes up scarcely any room, has nothing to get out of order, can be instantly put in action and is self-priming. Any number of these ejectors can be connected to the same steam main up to its full steam-delivering capacity, and the pumping capacity varied by the number of ejectors in service.

The efficiency of the steam ejector is very low, but in recent years it has been increased, and there seems no reason why still better results could not be obtained by careful investigation and experiment. If by proper design an ejector working on a much larger scale than usual could be built that would develop 1 hp. on 50 lb. of steam, then we would have a very practical and efficient pump to use in such a contingency as occurred on the "Titanic."

A few approximate figures will give some idea of what could be accomplished along this line. The combined boilers of the "Titanic" ought to have been able to easily deliver 500,000 lb. of steam per hour, and this at 50 lb. per hp. for the ejectors would give 10,000 hp. as the pumping capacity. This is equivalent to $10{,}000 \times 550 = 5{,}500{,}000$ *ft.-lb. per sec.* If this was used in elevating the water 20 ft., then the weight of water lifted per second would be

$$5{,}500{,}000 \div 20 = 275{,}000 \text{ lb.}$$

If a cubic foot of salt water weighs 64 lb., then the discharge would be

$$275{,}000 \div 64 = 4297 \text{ cu.ft. per sec.}$$

Approximately what size hole in the side of the ship would it take to balance this discharge? Assume that the average center of depth of the hole is 20 ft. below the water line, and the rate of inflow at this depth to be equal to $0.8\sqrt{2g \times 20}$ which equals about 29 ft. per sec. This in all probability would be a higher velocity than actual, but errs on the safe side. As the inflow is equal to 4297 cu.ft. per sec., and as the inrushing water travels with an average velocity of 29 ft. per sec., then the area of the opening will be

$$4297 \div 29 = 148 \text{ sq.ft.}$$

say a hole 49 ft. long by 8 ft. wide. In the case of the "Titanic" this additional pumping capacity, combined with the water-tight compartments, would surely have floated the ship many more hours, if it had not actually kept it afloat and allowed collision mats to have been placed over the opening and checked the leakage until port was reached, or until a rescue ship arrived.

GEORGE P. PEARCE.

Exeter, N. H.

Engine Connected up Backwards

The article by J. H. Stratton, under this heading, in the Aug. 13 issue, I read with considerable interest.

There are some things, however, that do not look reasonable. I can understand how, by excessive compression, the pressure in the cylinder could rise above boiler pressure, but I fail to see how the engine could run when the pressure in the cylinder did not drop below boiler pressure at any time.

Perhaps Mr. Stratton can explain the phenomenon.

W. P. MARTIN.

Mt. Olive, Ill.

At first I doubted that the engine described by Mr. Stratton would run at all, but after drawing a sketch of a piston valve in several positions of stroke, I think it possible that the engine would turn over with the valve set just right. At least two statements in the article, however, need further explanation to make them clear to the average reader.

How could the cylinder show a higher pressure than the boilers except owing to compression; and if the pressure never dropped below 120 lb. (or boiler pressure), where did any power come from for compression, to say nothing of carrying the load?

If the engine ran at 650 r.p.m., it must have been a different quality of water than we have here that would flood that engine to a standstill and leave it in shape for a fresh start as soon as the water was gotten rid of. It would not seem likely that anything lighter than solid water would stop a machine that was piped up right if it was of sufficient power to run on with steam piped into the exhaust. The engines around here, especially piston-valve engines, have a way of making a bad wreck of themselves when water comes over from the boiler in large quantities.

L. JOHNSON.

Exeter, N. H.

Inward Swinging Fire Doors

The editorial, "Precautions Against Damage from Tube Failures," in the Aug. 20 issue, mentions fire-doors that swing inward as a preventive against scalding the fireman when a water-tube boiler tube blows out. This type of door is a great aid in this respect, and the editorial has not overstated its advantages.

I have experienced two cases of this kind. In one the fireman was standing directly in front of the boiler when the tube let go. The door was shut and when the excitement was over, it was found that nearly all the fire was banked up against the doors. Had the doors opened outward, the fireman would have been seriously or perhaps fatally scalded, as there was no warning and he could not have gotten away in time to save himself. The other case was nearly the same except that the fireman had just fired up and stepped to one side. The doors fitted quite tight and it was possible for the fireman to pass in front of the boiler to reach the ladder by which he reached the top of the boiler to shut off the stop valve. These boilers were not equipped with automatic stop valves as they should have been.

Although this type of door has the advantage of safety, it has some serious disadvantages (at least the kind I used had). They do not fit tight enough after the setting becomes old and warped out of shape, and to get them open wide the brick arch over the door must be set with very little spring and be higher above the opening to make room for the lining when the door is open. This leaves a part of the door frame exposed to the fire which soon cracks or warps the casing, causing the door to bind or leak air, and opening it against the arch causes the arch to become loose and fall down. I find that the life of the door and arch are not as long as that of the outward-swinging door.

It has another advantage, however, that the editorial did not mention. The fireman's side is not exposed to a red-hot door-lining while firing or cleaning fires. As they are counter-balanced, they are much easier to open than the ordinary door. In the above mentioned plant, boilers were in use, fitted with both kinds of doors, and the inward-swinging door boilers were preferred to those with the outward swing.

J. C. HAWKINS.

Hyattsville, Md.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Temperature of Superheatea Steam

What would be the temperature of steam at 150 lb. gage pressure superheated 250 deg.?

O. S.

The temperature of steam at 150 lb. gage pressure being 366 deg. F., superheated 250 deg. the temperature would be 366 + 250 = 616 deg. F.

Increasing Height of Chimney

What would be the advantage of increasing the height of a chimney from 125 to 200 ft.?

A. C. B.

The amount of draft would be increased about as the square root of the increased height, i.e., as 11.18 to 14.14, or an increase of about 26 per cent.

Liquid Fuel Injectors

In place of using compressed air for fuel injection in Diesel engines, why not use superheated steam generated from the heat of the exhaust?

E. R.

A pressure of 500 to 900 lb. per sq.in. is required, and as the temperature of the exhaust is only about 700 deg. F. it is unsuitable for generation of steam of pressure and temperature required for fuel injection.

Use of High Vacuum

Why is a higher vacuum than 26 in. less desirable in operating reciprocating condensing engines than in operating steam turbines?

B. G.

In most reciprocating engines, increased condensation due to higher vacuum would offset gain due to expansion, unless the steam were highly superheated. In the steam turbine such losses do not occur, as high- and low-pressure steam do not alternately come in contact with the metallic surfaces.

Laying Up a Boiler

What is a good method to prevent rusting of a boiler and iron smoke connections which are to be laid up out of use for several months?

J. R.

Wash out the boiler with clean water, disconnect the feed-water pipe and after draining off the water remove the safety valve or a manhole on top of the boiler. Then with a very light fire drive out all moisture and close the boiler up tight. All flues and smoke connections should be thoroughly cleaned, the smoke uptake should be disconnected from the stack and the boiler flues, smoke flues and stack connection should be stopped up to prevent circulation of air from depositing moisture.

Hold of Expanded Tube Ends

Can expanding the tube ends of a return-tubular boiler without special flaring or beading, be depended on for safely staying the heads, though the tubes might be expanded tightly enough to prevent leakage when the boiler is under cold-water hydrostatic test pressure?

B. E.

No, as the hold of expanding is almost entirely dependent upon friction, and a cold-water test pressure is no criterion of the holding power or safety of the tube ends when the boiler is under pressure, especially after it has been subjected to changes of temperature. Serious explosions under ordinary working pressures are directly tracable to failure of whole sets of tubes that have been only expanded by being drawn through the tube-sheet holes, due to bulging of one of the heads, though they have withstood 150 per cent. hydraulic test pressure, but failed without warning under ordinary working conditions.

Effect of Back Pressure

What effect will increase of back pressure for exhaust-steam heating have on the power and economy of an engine driving the same load?

W. F. B.

As the same mean effective pressure will be required, each pound increase of average back pressure will require a pound increase of the average forward pressure. The engine will be able to drive the same load, provided the initial pressure or length of cutoff can be increased so as to obtain the same mean effective pressure. In consequence of later cutoff or higher initial pressure being required for the same load, more steam will be needed per indicated hp. Under average conditions the percentage increase of steam required will be somewhat greater than the ratio of increase of average back pressure to the mean effective pressure, though for commercial considerations, the ratios may be considered as equal. If, for instance, the required mean effective pressure is 50 lb. and it is proposed to increase the average back pressure by 2 lb., then the same increase of steam will be required as though the change consisted in an increase to 52 lb. mean effective pressure. Knowing the hp. constant of the engine for 1 lb. mean effective pressure, it is readily ascertained to what per cent. increase of power the proposed increase of back pressure would be equivalent.

Starting a Long-Idle Plant

What should an engineer attend to if placed in charge of a power plant to operate it if the plant has not been operated for two years?

H. H.

Make hammer test and if necessary renew all defective boiler connections in each instance back to the first stop valve, and see that try-cocks and water-column connections are clean of any scale or sediment and all stop, check and safety valves are in good working order. Have the steam gage tested. Clean out furnaces and back connections and remove all soot from smoke uptakes and flues and from the base of the chimney. Make internal and external inspections of boilers and settings. Examine water supply and overhaul feed pumps and injectors. Examine whether blowoff connections are open to their place of discharge. After boiler and connections are cleaned and put in safe working order, fill boiler and with a very slow wood fire examine setting and stop any air leaks and repair them. Then slowly getting up steam, note any leaks and repair them, but not while under pressure. Make a light hammer test of heater and connections.

Remove engine-cylinder head, clean out grit and dirt from all steam passages by blowing steam through them and have cylinder perfectly clean before replacing the head. Repack stuffing-boxes and inspect valve setting and all governor connections.

Trace out all engine connections and examine valve setting. Uncover and clean all bearings and flush them with clean thin oil. In starting the engine, use plenty of lubricant. Before putting steam in the engine follow up all transmission connections to see that they are in safe running order, and in starting up run slow and have the load laid on gradually. The most important considerations are to safeguard life, limb and property and conditions of plant that will insure continuous operation.

Study Questions

This Week's Questions
Last Week's Answers

(106) The armature of a shunt-wound dynamo has a resistance from brush to brush of 0.03 ohm. The voltage across the brushes is 110 volts and a current of 150 amp. flows in the external circuit. The shunt field has a resistance of 55 ohms. What is the copper loss in the armature?

(107) A vessel (like a drinking glass) 9 in. in diameter at the top, 4½ in. in diameter at the bottom and 10 in. high, is one-quarter full of water. How large a ball can be put into it and be just covered by the water?

(108) If the pointer of a vacuum gage stands at 9, what is the approximate absolute pressure in the condenser?

(109) From an indicator card the steam consumption is computed to be 28.6 lb. per i.hp. If the mechanical efficiency is 87 per cent., what is the steam consumption per brake-horsepower?

(110) What is the thermal efficiency when the initial steam pressure is 85 lb. absolute and the exhaust pressure 2 lb. above vacuum?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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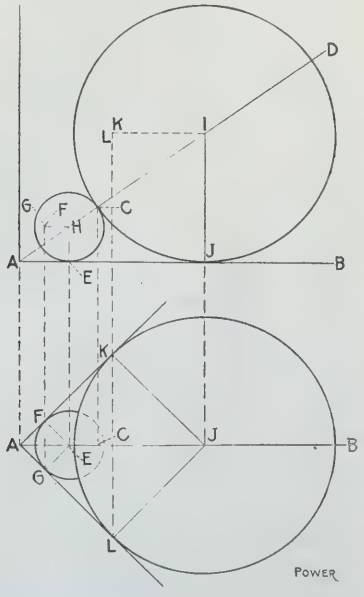
(101) The lower diagram is a top view of the two balls in the corner of the room. The line *AB* represents a vertical plane passing through the centers



of the two balls, their point of tangency *C* and the corner of the room. The upper diagram is a projection of this plane, or may be considered a vertical section through the centers of the balls; the point of contact between them *C* will lie on a line *AD* through the corner of the room and the ball centers. It is given that the diameter of the small ball is 12 in. Then its radius is 6 in. and this will be the distance from the center to the contact with either wall as *EF* and *EG*. Since *AFE* and *AGE* are right angles *AF* and *AG* will also be 6 in. long. The horizontal distance from the corner to the center of the ball *AE* will be

$$AE = \sqrt{AF^2 + FE^2} \qquad (1)$$

The actual distance will be *AH*, which is

$$AH = \sqrt{AE^2 + EH^2} \qquad (2)$$

EH also is the radius of the ball 6 in.; hence from (1) and (2)

$$AH = \sqrt{6^2 + 6^2 + 6^2} = \sqrt{108} = 10.392$$

The distance *AC* (upper diagram) is *AH* + *HC*, or

$$AC = 10.392 + 6 = 16.392 \text{ in.}$$

The radius of the larger ball is to be determined. Call it *x*. Then the distance from the corner to the center of the larger ball (*AC* + *CI*) is

$$AI = 16.392 + x \qquad (3)$$

AI can also be found, as was *AH*, by solving right-angled triangles.

$$KJ = x = AK$$

Then

$$AJ = \sqrt{KJ^2 + AK^2} = \sqrt{x^2 + x^2}$$

and

$$AI = \sqrt{AJ^2 + IJ^2} = \sqrt{x^2 + x^2 + x^2}$$
$$= \sqrt{3x^2} = x\sqrt{3} \qquad (4)$$

Equating (3) and (4)

$$16.392 + x = x\sqrt{3}$$
$$x\sqrt{3} - x = 16.392$$
$$x(\sqrt{3} - 1) = 16.392$$
$$x = \frac{16.392}{0.732} = 22.393$$

The diameter of the larger ball is therefore

$$22.393 \times 2 = 44.786$$

or very closely 44¾ in.

(102) The man's path will be the diameter of the circle; the automobile's, its circumference. The distance the man goes will therefore be as *D* and the distance the automobile goes as $\frac{3\pi D}{2}$ in the same time; hence the automobile must go

$$\frac{\frac{3\pi D}{2}}{D} = \frac{3\pi}{2} = 4.7124$$

times as fast as the man. If the man goes eight miles an hour the automobile must go

$$8 \times 4.7124 = 37.699$$

or approximately 37.7 miles an hour.

(103) J. B. Stanwood's formula as given in "Kent's Pocketbook" is

$$W = \frac{700{,}000\, d^2 s}{D^2 R^2}$$

where

d = Diameter of cylinder, in. = 18;
s = Stroke, in. = 36;
D = Diameter of flywheel, ft. = 15;
R = R.p.m. of flywheel = 90;
W = Weight of rim, lb. = ?

Substituting

$$W = \frac{700{,}000 \times 18^2 \times 36}{15^2 \times 90^2} =$$

$$\frac{700{,}000 \times 324 \times 36}{225 \times 8100} = 4480 \text{ lb.}$$

(104) The theoretical mean effective pressure will be the same as admission 110 lb. From the formula

$$I.hp. = \frac{plan}{33{,}000}$$

where

p = Mean effective pressure, lb. = 110;
l = Length of stroke, ft. = $\frac{30}{12}$ = 2.5 ft.;
a = Area of cylinder = 0.7854×18^2 = 254.47 sq.in.;
n = Strokes per min. = r.p.m. × 2 = 150.

$$I.hp. = \frac{110 \cdot 2.5 \cdot 254.47 \times 150}{33{,}000} =$$

$$\frac{10{,}496{,}887.5}{33{,}000} = 318.0875 \text{ hp.}$$

(105) The total heat in 1 lb. of steam at 11 lb. abs. (from steam tables) is 1144.9 B.t.u. Leaving at 120 deg. F. the 1 lb. of steam still contains 120 — 32 = 88 B.t.u. It has, therefore, lost

$$1144.9 - 88 = 1056.9 \text{ B.t.u.}$$

The cooling water has been raised

$$110 - 53 = 57 \text{ deg.}$$

Then each pound of it has absorbed that number of B.t.u. The quantity of heat lost by each pound of steam is therefore sufficient to raise

$$\frac{1056.9}{57} = 18.54 \text{ lb.}$$

of water through the range specified. Since the engine uses 26 lb. of steam per horsepower-hour and runs at 53 hp., the condenser receives

$$26 \times 53 = 1378 \text{ lb. per hr.}$$

The cooling water required per hour is therefore

$$18.54 \times 1378 = 25{,}548.12 \text{ lb.}$$

The formula for calculating the surface of condensers, as given by H. L. Hepburn in POWER, April, 1902, is

$$A = S \frac{H - N}{F - I} \times \log_e \frac{T_s - I}{T_s - F}$$

where

A = Area of surface, sq.ft.;
U = B.t.u. transmitted per sq.ft. per hour per deg. mean difference of temperature between the steam and water = 192;
S = Pounds of steam per hr. = 1378;
H = B.t.u. above 32 deg. F. in 1 lb. of steam = 1144.9;
N = B.t.u. in 1 lb. of condensed steam = 88;
F = Final temperature of the water = 110 deg. F.;

I = Initial temperature of the water = 53 deg. F.;

T_s = Temperature of the steam.

From steam tables T_s = 197.75 deg. F. Substituting in the formula

$$A \times 192 = 1378 \frac{1144.9 - 88}{110 - 53} \times log._e \frac{197.75 - 53}{197.75 - 110}$$

$$A \times 192 = 1378 \frac{1056.9}{57} \times log._e \frac{144.75}{87.75}$$

$$A \times 192 = 25{,}551.02 \times log._e 1.65$$

$$A = 133.06 \times log._e 1.65$$

$Log._e$ of 1.65 (from tables of hyperbolic logarithms) is 0.5008. Then

$A = 133.08 \times 0.5008 = 66.646$ *sq.ft.*

Free Electrical Lectures in Brooklyn

The Brooklyn Institute of Arts and Sciences announces a series of five lectures by Dr. A. E. Kennelly, of Harvard University, on the elements of hyperbolic functions and their applications to electrical engineering. The lectures, which will be illustrated by lantern slides, are free to the profession and will be given on the second Thursday evening of each month of the season in the physics lecture room of the Polytechnic Institute, Brooklyn, at 8:15. A working acquaintance with ordinary circular functions of plane trigonometry will be assumed, but no calculus or mathematics above algebra will be involved.

The dates and subjects of the lectures are as follows: Oct. 10, hyperbolic angles: their properties and applications; Nov. 14, the behavior of direct-current lines of uniform linear conductor resistance and dielectric conductance, in the steady state, as simplified by the use of hyperbolic functions; artificial lines and equivalent circuits; Dec. 12, the behavior of alternating-current lines of uniform linear conductor-impedance and dielectric admittance, in the steady state, and also in simple unsteady states, as simplified by the use of hyperbolic functions; Jan. 9, the application of hyperbolic functions to long alternating-current power-transmission lines; Feb. 13, the application of hyperbolic functions to telephone lines, loaded and unloaded, single and composite.

Standardizing Electricity in France

A bill for the protection of consumers against fraud is reported in preparation by the French Minister of Commerce. It will establish standards for electricity, heat and candlepower and the service companies will be compelled to conform to its standards. Meters and lamps will be subject to the same regulations as weights and measures.

Gas Engine Valve Pliers

A new tool, which seems to fill a want for users of four-cycle engines, is shown in Fig. 1. Fig. 2 shows its application

FIG. 1. VALVE PLIERS

to a poppet valve of a gas or gasoline engine. Valves of this class show very little wear if kept clean, and with this tool but a few moments is required to remove a valve and clean it.

The ring on the handle of the pliers is

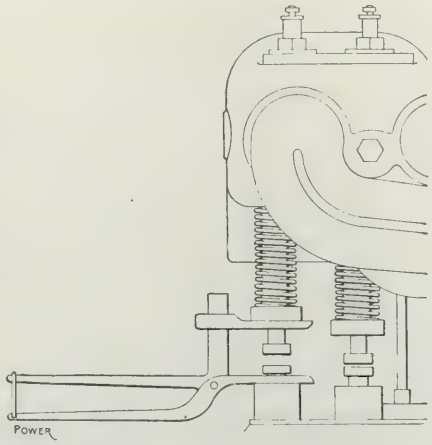


FIG. 2. APPLICATION OF PLIERS

for locking them when the spring is collapsed. The engaging jaws are forked to give them a wide range in valve stem size. The pliers are made of steel case-hardened drop forging by C. A. Cummings, P. O. Box 373, Canton, Ohio.

100,000,000 Gallon Turbine Driven Pump

A steam turbine-driven centrifugal pump of 100,000,000 gal. per day capacity is to be installed in the Ross pumping station of the Pittsburgh Water Works by the DeLaval Steam Turbine Co., of Trenton, N. J. This pump will be the largest steam turbine-driven centrifugal pump ever built in this country, the rated capacity of 100,000,000 gal. per day against a total head of 56 ft., amounting to over 980 water horsepower. The pump is guaranteed to show a duty of 115,000,000 ft.-lb. per 1000 lb. of dry steam, all steam used by the condensing equipment being charged to the main unit.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Blink Boffum, of the Waters power plant, opines that the Nifetown *Blade* is too cutting in remarking that he is "a rising young man." Says he was willing to stay down, but "ther dum ingineer weighted ther safety, an' when ther old biler riz," Blink had no choice.

"Anxious Subscriber" inclosed $2 and asked for the solution of a perpetual-motion problem. The best we could do to relieve his anxiety was to return the $2. As for perpetual motion, "there ain't no sich thing," as the farmer said the first time he saw a giraffe.

The following is an appreciation of our humble efforts to drive away dull care and make merry in our travail:

deer mister Spilway. Them 2 lines of yourn beets awl for testing my Mans sobriarty. he cum hoam 2 ours lait from the plant 1 nite, sum werce for ware. "i bin makin a test, Malviny," sez he. "Justus Jinkins," sez i, "ef you kin repeet them 2 lines from yure Power, 'just jests, jabs, joshes and jumbils', ile allow yure tung aint mixt ez bad as yure laigs be."

Justus aint agivin awl the datter of my test, but i heerd him tell Ezry Skidmore that he was rite het up ovar the konsidabul low pursentige of his fishency, whatevar them things is. hes goin ter vote along with the "drys" come this fall.

Yure obleeged friend,
MALVINA JINKINS.

At a Pittsburgh theater recently the orchestra played "Meet Me in the Shadows" when an accident in the lighting plant shut off the current and left the audience in darkness. American humor, when it is not too dark to see the joke, has often prevented a panic.

An ingenious Richmond is in the electrical field with a scheme for putting people to sleep by electricity. The lure of Morpheus is good enough for us at night-time. Keeping the wolf from the door makes us sufficiently alert and wakeful day-times.

This engineer made good, boasts a contemporary. Bully! With a few added details, the announcement would make an excellent construction note.

Looks like the New York police force was in need of a few lightning-arresters as some of them have been under heavy "charges" recently. A few might well be transformed and stepped down—and out.

Steam Boiler Explosions*

By William H. Boehm

In which it is pointed out that the principal causes of boiler explosions are due to faulty design and construction, false factor of safety, improper care and other faults due to ignorant operation.

*Excerpt from a lecture delivered at Cornell University before the state branch of the American Society of Mechanical Engineers.

Every year there occur in the United States between 1300 and 1400 serious boiler accidents, of which 300 to 400 are violent explosions. These accidents kill between 400 and 500 persons, injure 700 to 800 more, and destroy more than a half million dollars worth of property. These disasters have but scant respect for boiler types—for they occur with water-tube boilers, although with them violent explosions occur less frequently than with fire-tube boilers. They also occur with low-pressure boilers.

Design and Construction

It is of the utmost importance that boilers be carefully designed, that the stresses to which they are subjected be accurately computed, that suitable material be specified, that the material be critically examined for flaws or defects, that specimens of the material be tested to determine its strength, that no abuse of the material be allowed in the process of constructing the boiler, and that the completed boiler be subjected to a thorough inspection and a hydrostatic test before being put into service.

Stress in Girth and Side Seams

The stress in the girth seams of a boiler may be obtained by the formula

$$S = \frac{r p}{2 t}$$

in which

S = Stress per square inch to which the material is subjected;
r = Radius of the boiler;
p = Steam pressure carried;
t = Thickness of the shell.

The formula means that if we multiply the shell radius by the steam pressure, and divide the product by twice the shell thickness, the result will be the stress in pounds per square inch to which the material in the girth seams is subjected.

The stress in a side seam of a boiler may be obtained by the formula

$$S = \frac{r p}{t}$$

which means that if we multiply the shell radius by the steam pressure and then divide the product by the shell thickness, the result will be the stress in pounds per square inch to which the material in a side seam is subjected.

An inspection of these formulas show that the stress in a side seam is just twice as great as the stress in a girth seam. It is for this reason that the side seams are usually double riveted when the girth seams are only single riveted.

Bursting Pressure

The pressure required to rupture the shell of a cylindrical boiler is given by the formula

$$P = \frac{s t}{r} e$$

in which

P = Bursting pressure in pounds per square inch;
s = Tensile strength of the material in the boiler;
r = Radius of the shell;
e = Efficiency of the riveted side seam.

Stated in words the formula means that if we multiply together the tensile strength of the material, the thickness of the shell, and the efficiency of the riveted side seams, and then divide the product so obtained by the radius of the shell the result will be the steam pressure at which explosion will occur.

If it so happens that the efficiency of the girth seam is too low by reason of improper design, then the girth seam may fail instead of the side seam, in which case the bursting pressure is given by the formula

$$P = \frac{2 t s}{r} e'$$

which means that if we multiply together twice the thickness of the shell, the tensile strength of the material, and the efficiency (e') of the girth seam, the result will be the steam pressure at which explosion will occur. The girth seam is not likely to fail before the side seam, because to do so the efficiency of the girth seam would have to be less than half that of the side seam—a weakness that should not exist in a boiler of proper design and construction.

Factors of Safety

The formulas expressed herein give the pressure at which the boiler will explode and not the pressure at which it may be safely operated. It is usual in boiler practice to fix the allowable working pressure for a new boiler at one-fifth the computed bursting pressure and to decrease the pressure allowance as the age of the boiler increases.

This is equivalent to saying that the factor of safety applied to a new boiler is usually not less than five. The term "factor of safety," is often misunderstood and a better name would be "factor of ignorance," as it is as much a factor of ignorance as it is a factor of safety.

Take, for example, the case of a new boiler operated with the safety valve set at 100 lb. If the computed bursting pressure be 500 lb., then the assumed factor of safety is five. The assumption, however, is based upon the tensile strength stamped in the plate by the maker and this strength is only true of that particular part of the plate from which the test specimen was cut and not necessarily true of any other part.

As a matter of fact it is current practice to cut the test specimen from the outer edge of the plate and the strength there is almost invariably greater than the strength at the center of the plate. The reason is that after liquid steel is poured into a mold its solidification in forming an ingot proceeds in much the same manner as does the solidification of water in forming a block of ice. That is to say, the impurities and gases are driven toward the center.

Boiler plates made by the rolling of such an ingot will, therefore, have more impurities and less strength at the center of the plate than at the outer edges, and this variation in strength is very considerable. Then, too, the stress at which the elastic limit of the material is reached is little more than half the stress at which rupture occurs.

Besides our ignorance of the dependable strength in all parts of the plate, there is also our ignorance of the character of the workmanship in the boiler. We cannot be certain that all rivet holes come fair, or that incipient cracks have not been set up by an abuse of the material during the process of construction. It is seen, therefore, that factors of safety are really made up of two parts—one part a true factor of safety, the other a pure factor of ignorance.

Causes of Boiler Explosions

Boiler explosions may be attributed to improper construction, improper installation, or incompetent or careless operation.

Improper construction may consist: of unsuitable or inferior material; poor workmanship; abuse of material, as when unmatched rivet holes are drift-pinned to place, or uncylindrical shells are sledged to form; of employing the more dangerous lap joint for the side seams instead of the more safe and more sensible butt joint, etc.

The lap joint is dangerous because this form of construction promotes the formation of incipient cracks in the upper surface of the lower lap where they may

be impossible of detection. These cracks extend from rivet hole to rivet hole and gradually deepen with the continued raising and lowering of the steam pressure until the metal, no longer capable of resistance, gives way and causes a violent explosion.

The lap joint is given the preference over the butt joint solely, because, it appears at first thought to be cheaper. But there is no excuse for its existence, and its employment in the construction of new boilers should be prohibited by law in all states, as it now is in some.

Improper installation may consist of so supporting the boiler and its piping as to allow temperature changes to set up dangerous stresses in the material, of improperly attaching the usual appurtenances such as safety valves, steam and water gages, check, blowoff, stop valves, etc.

Incompetent or careless operation may consist in allowing the steam gage to get out of order, in allowing the water-gage connections to become so clogged as to indicate ample water when there is none in the boiler, in allowing the safety valve to become so stuck to its seat as to fail to blow at the pressure for which it was set, in allowing grease to enter or scale to accumulate in the boiler, in allowing large quantities of cold water to be impinged against hot plates, in allowing the water to be driven from the heated surfaces by forced firing, in allowing a large valve to be opened too suddenly, in allowing two boilers to be cut in on the same steam main when their pressures are unequal, and in allowing minor repairs to be neglected until they endanger the whole structure.

It is significant that many violent boiler explosions occur either just prior to the starting of the engines in the morning, or while they are idle at the noon hour, or shortly after they have been shut down for the day. One reason is that when steam is not being drawn from the boiler it accumulates rapidly; and if the safety valve fails to relieve the pressure, explosion soon follows.

The rapidity with which the bursting pressure is reached may be shown as follows:

Let

T = Time in minutes required to reach the bursting pressure;
W = Weight of water in the boiler;
t = Temperature of the steam at bursting pressure;
t' = Temperature of the steam at normal working pressure;
U = Number of heat units per minute supplied by the furnace and absorbed by the water.

The heat balance is then represented by the equation:

$$UT = W\ (t - t')$$

$$T = \frac{W}{U}(t - t')$$

which means that if we multiply the difference between the temperature of the steam at bursting pressure and at normal pressure by the weight of the water in the boiler, and then divide the product by the number of heat units supplied per minute by the furnace, the result will be the number of minutes that will elapse from the time the openings are all closed until explosion follows.

Take, for example, a 100-hp. boiler containing at normal level 5000 lb. of water and suppose it uses 50,000 heat units per minute when evaporating 50 lb. of water per minute. Then if the normal gage pressure be 85 lb., the corresponding temperature of the steam is 327 deg., and if the bursting gage pressure be 485 lb., the corresponding temperature of the steam is 467 deg.; and the time required to reach the bursting pressure with all steam openings closed and the safety valve stuck is:

$$T = \frac{5000}{50{,}000}(467 - 327) = 14\ min.$$

That is, with a stuck safety valve, only 14 min. would elapse from the time the engines were shut down until the explosion followed.

Explosive Energy in Heated Water

The temperature of the water in a boiler is approximately the same as the temperature of the steam with which it is in contact. If the fire be drawn when the openings are closed, ebullition ceases. If a valve be opened, ebullition starts again, even though there still be no fire under the boiler.

It is plain, therefore, that with the openings closed it is the pressure on the surface of the water that prevents further generation of steam. It is also plain that if a small rupture occurs below the water line a violent explosion may not ensue. But it ought to be evident that if a large outlet above the water line be suddenly opened, as, for example, when a steam pipe fails, then the sudden liberation of the pressure on the surface of the high-temperature water will allow it to flash suddenly into steam and cause a violent explosion and water-hammer that will disrupt the strongest possible construction.

Grease in Boilers

Grease does not dissolve or decompose in water, nor does it remain on the surface. Heat in the water and its violent ebullition causes the grease to form in sticky drops which adhere to and varnish the metal surfaces of the boiler. This varnish by preventing the water from coming into intimate contact with the metal, prevents the water from absorbing the heat, and this causes a blistering or burning of the plate that often results in a serious rupture, or a violent explosion.

Scale in Boilers

If scale is allowed to accumulate to any considerable thickness in a boiler, a bag or rupture of the shell is inevitable, unless the scale happens to be of a spongy formation, which is not often the case. Just why this is so, is shown by the following simple experiment.

Take an ordinary granite iron or tinned iron stewpan and firmly glue to its underside a postage stamp. Pour water into the pan and place it on a gas stove so that the postage stamp will be in direct contact with the flame. Leave the pan on the stove until the water has boiled violently and then examine the stamp. The stamp will not even be charred, much less burned, notwithstanding that it was on the underside of the pan and in direct contact with the hottest part of the flame.

Now put into the pan a mixture of water and portland cement half an inch thick. This, when set, will be the equivalent of half an inch of scale. Repeat the experiment made before and it will be found that the stamp will burn up very quickly.

The reason that the postage stamp is not charred by the flame when no scale is present is that the water, being in immediate contact with the thin bottom of the vessel, absorbs the heat as fast as it is put into the vessel by the flame. The result is that, no matter how hot the flame may be, the bottom of the vessel remains at practically the same temperature as the boiling water with which it is in contact. In an open vessel the temperature of boiling water is 212 deg. and this is not sufficiently high to char paper. When scale is present, the water cannot absorb the heat as fast as it is put into the vessel by the flame, and as a result the temperature becomes greater than 212 deg. and burns the postage stamp.

It is the same with steam boilers. If the water comes in direct contact with the thin plates, the heat is absorbed, the temperature of the plates remains practically the same as the water, and no harm is done. If there be a considerable thickness of impervious scale in the boiler, the water cannot absorb the heat as fast as it is put into the plates by the furnace, and so the plates become overheated, get red, become plastic, and finally give way to the force of steam pressure, causing a bag, or a rupture, or a violent explosion of the boiler.

Scale endangers the safety of boilers in other ways. It clogs the feed pipes, preventing the feed water from freely entering the boiler. It clogs the connections to the water gage, causing it to indicate ample water when it is at a low level in the boiler. Pieces get under valves and prevent their closure.

Scale in boilers is a serious matter, and in order to prevent its accumulation, it is good practice to eliminate the scale-

forming matter from the feed water before allowing it to enter the boiler. This can be accomplished either mechanically by means of separators, or chemically by treating the water in vats especially arranged for the purpose. If preferred, compound may be fed with the water into the boiler, but in such case the water should be analyzed, and the proper compound prescribed by a chemist making a specialty of such matters. Kerosene fed into the boiler has proved beneficial in many instances.

Inspection and Insurance

It is an almost universal custom for boiler owners to have their boilers insured and inspected. The insurance serves as a guarantee that the inspections will be intelligently and carefully made and the inspections lessen the chance of accident.

When boiler insurance is carried, an inspector visits the plant at regular intervals and critically examines the boilers, both internally and externally. During the past 10 years the company represented by Mr. Boehm made 1,101,140 examinations and reported 140,989 defects, many of which consisted of dangerous fractures in or near the riveted seams, and that one boiler out of every eight examined, contained defects serious enough to warrant their being reported.

Power Possibilities of British Columbia

The work of investigating the power possibilities of British Columbia, inaugurated last year, has been resumed under the direction of Arthur V. White, of the Canadian Commission of Conservation, who has charge of all work of this nature in western Canada. Three engineering parties are now engaged examining the Fraser River and tributaries between Lytton and Fort George. This embraces one of the largest drainage basins on the continent.

The recent report on the water powers of Canada drew special attention to the fact that the water powers available for development from an economic standpoint are fewer in number and potentiality than is popularly supposed. Their findings have an important bearing on the question of conserving these powers in the public interest in view of the new towns and cities springing into existence in British Columbia.

In the report the commission promised that special investigations would be carried on in the prairie provinces and British Columbia, because all the required data were not available at the time of publication. The water powers of the Kootenay River have already a profound economic influence in the Kootenays and Boundary district. Similarly the same is expected to be effected by the Adams River water power throughout the Shuswap Lake district and South Thompson River county, where irrigation is being resorted to.

Reported Boiler Explosion

According to a newspaper report a boiler explosion at the South Madison St. plant of the Bloomington, Ill., Railway, Electric & Heating Co., at 6:35 p.m., Sept. 21, partly wrecked the building, injuring one fireman, but not fatally. The entire service was disabled.

The No. 12 boiler, which is said to have exploded, had been out of service for six weeks for general repairs, which were done by the plant workmen. It had been in service but 20 min., when the accident occurred. No reason is given as to the probable cause.

Artificial Graphite

The manufacture of artificial graphite, according to a bulletin of the U. S. Geological Survey, is conducted by means of the electric furnace, an anthracite coal carrying small amounts of evenly distributed impurities being the material from which the ordinary grades are made.

For obtaining the purest grades of graphite, petroleum coke is substituted for anthracite. The process for the manufacture of graphite was patented in 1896 and its commercial development has been so rapid that at present the output of artificial graphite in the United States is greater than the whole domestic production of natural crystalline graphite.

Testing the British Yard Measure

The ceremony of testing the British yard measure and the British pound weight which occurs every 20 years is soon to be performed. A copy of each is buried in a wall of the House of Commons, and is compared at stated intervals with the originals to see if the latter have undergone any deterioration. They order these things better in France, where, by the skill of an American mechanic and the science of an American professor the length of the meter has been determined in terms of light waves so that it could at any time be recovered if the original standard were lost. This is more accurate than the original determination in terms of the circumference of the earth, which errs a little through defective geographical measurement.—*Popular Engineer.*

The Union Electric Light & Power Co., St .Louis, Mo., on Sept. 27 took a party of 150 manufacturers as its guests to Keokuk, Iowa, to inspect the hydro-electric plant under construction at that point by the Mississippi River Power Co., which will supply much of the power to be used in St. Louis.

NEW PUBLICATION

THE LOCOMOTIVE. Published by the Hartford Steam Boiler & Inspection Co., Hartford, Conn. 1910-1911. Size, 5½x9 in.; 254 pages; illustrated; complimentary.

This book is compiled from the quarterly bulletins published by the company. The volume is filled largely with data relating to boiler explosion and boiler inspection. There are also articles treating with the care of boilers.

One article in particular dealing with the management and care of steam boilers is of more than passing interest to those who are not thoroughly familiar with this phase of steam-plant operation. The instructions given therein are based on sound logic and common sense.

The list of boiler explosions during the two years covered by the book give food for thought to those who may feel secure in the belief that boiler explosions are not common.

There are also many short instructive and interesting articles too numerous to mention.

BOOKS RECEIVED

DIESEL ENGINES. By A. P. Chalkley. D. Van Nostrand Co., New York. Cloth; 226 pages, 5¼x8½ in.; illustrated; plates; tables. Price, $3.

THE PRINCIPLES OF PARALLEL PROJECTING-LINE DRAWING. By A. A. Adler. D. Van Nostrand Co., New York. Cloth; 66 pages, 5½x8¾ in.; 48 illustrations. Price, $1.

BOILER DRAUGHT. By H. K. Pratt. D. Van Nostrand Co., New York. Cloth; 138 pages 4¾x7½ in.; 29 illustrations; tables. Price, $1.25.

GAS AND OIL ENGINES. By Alfred Kirschke. D. Van Nostrand Co., New York. Cloth; 159 pages, 4½x7¼ in.; 55 illustrations; tables. Price, $1.25.

MATERIALS AND CONSTRUCTION. By James A. Pratt. Blakiston's Son & Co., Philadelphia, Penn. Cloth; 196 pages, 4¾x7 in.; 86 illustrations; tables. Price, 90c.

ENERGY AND VELOCITY DIAGRAMS OF LARGE GAS ENGINES. By Paul L. Joslyn. The Gas Engine Publishing Co., Cincinnati, Ohio. Cloth; 6x9 in.; 62 diagrams. Price, $2.

PERSONAL

Charles H. Clark, formerly chief engineer of the Connersville (Ind.) Light, Heat & Power Co., has been appointed manager of the Martinsville (Ind.) Light, Heat, Power & Water Co.

N. A. Carle has been recently appointed chief engineer of the Public Service Electric Co., with headquarters at Newark, N. J. Mr. Carle is a member of the American Society of Mechanical Engineers, the American Society of Civil Engineers and the American Institute of Electrical Engineers.

POWER

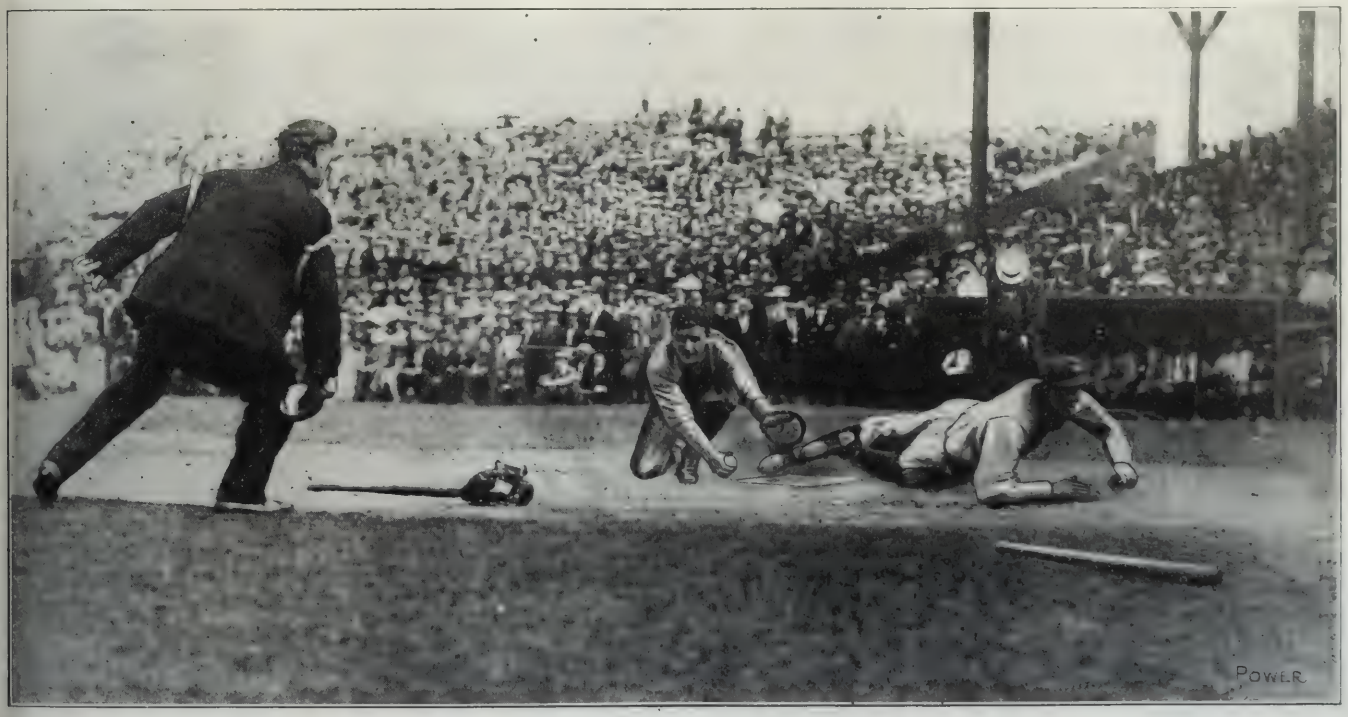
Vol. 36 NEW YORK, OCTOBER 15, 1912 No. 16

Playing the Game

AN oiler, fireman or assistant engineer is as necessary in a steam plant as a chief engineer. Neither may be as important, but each has a position to fill, either in a half-hearted way or energetically.

An oiler, for instance, may assume that his work does not count and only aims to hold his job; therefore why do any more than he has to?

One need not do more than he is compelled to, but that means a whole lot. There are two ways to work; by one it becomes drudgery and a thing detested; by the other, play, a thing enjoyed. The trouble is, most men do not know how to make their work enjoyable.



Suppose one man on a ball team lacks in interest, habitually lets chances slip without taking them. He would soon have to make way for someone who would *play the game*, a man who would work. It is each individual's work that makes the team work count.

The oiler may complain that he does not have a chance to adjust engines and handle them. True, but the successful ball player was not given a position on the ball nine untill he had shown that he was skilful. That came by practice. If the oiler wants to handle or adjust an engine, he will have to practice before he will be considered competent by the chief.

How? By lending a hand to the engineer who is doing the work, learning a little at a time. He will not have to do it, but it is *playing the game.*

No oiler plans to remain one all his life, and he will not if he *plays the game.* Steam engineering is an interesting study, and has no limit. Oiling a condenser pump is simple, but understanding its design, operation and what it accomplishes is a different proposition

A few instructions will enable one to oil an engine, but the action of the valves and of the steam in the cylinder are as greek to some men who have been accustomed to engines for years.

It is along these lines that the oiler, fireman and running engineer can make his work play. If he determines to know more about steam engineering a field of research will have been entered as interesting as it is remunerative.

He does not have to do it, and perhaps will not know that the effort is appreciated, but it is the only way to *play the game.*

Ljungström Radial Flow Turbine

Instead of the usual alternating stationary and rotating blades, moving blades only are employed. These are in circular rows upon the faces of oppositely revolving disks. The relative velocity is thus doubled and the number of stages quartered. Two generators are used which synchronize and run at the same speed without mechanical connection.

Several turbines have been proposed in which the guide blades, instead of being stationary, as in the Parsons and Rateau types, are attached to members moving in the opposite direction from the opposing blades. The complications involved by gearing these two motions to a common generator have discouraged the development of such a turbine; but the possibility of running an alternator from each of the oppositely rotating shafts and letting them synchronize, thus lock themselves together electrically, has been utilized by the brothers Birger and Frederic Ljungström, of Stockholm, who have built turbines of 500 and 1000 kw. capacity, and designed still larger units which are described in detail in *Engineering* of Apr. 12 and 19.

Allowing both sets of blades to rotate, and in opposite directions, doubles their relative velocity, which is particularly favorable to the reaction type of turbine. The Ljungströms, therefore, have made their turbine of this type. The simplest means of directing the steam upon oppositely moving blades is to use interlocking oppositely rotating disks, one connected to each motor, and this is the form which their turbine has taken. The steam enters at the center and flows radially, as shown in Fig. 1, which is a section, reproduced from *Engineering*, of the lower half of the turbine part of the 1000-kw. unit.

Steam enters through the conduits *AA*, and finds its way through the chambers *BB* and *CC* to the turbine, as indicated by the arrows. The top line of the engraving is the shaft center, and the steam, in flowing from *C* to the exhaust passage is flowing from the center toward the circumference. The section of the passage through the turbine is convergent-divergent and interrupted by rings of blading attached alternately to the disks *DD;* the first row is attached to the right-hand disk at *a*, the second to the left-hand disk at *b*, etc. The lines in the exhaust passage indicate the vanes of a diffusor by which it is claimed a lower pressure is maintained at the outlet of the last row of blading.

Fig. 2 is a section of two of the rings of blading. The blade proper is in the center at *A* and is attached to the disk *D* by the conical double bulb-headed ring, upon which the strengthening and holding rings *E* and *F* are spun. The ring *F* is held in the disk *D* by the soft iron calking strip *C*. Strips of very thin nickel *GG* run against the next ring, as shown, preventing lateral leakage.

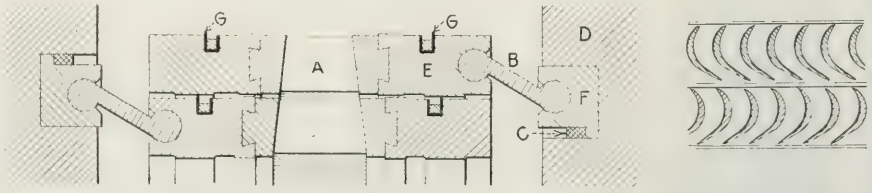


FIG. 2. SECTION THROUGH TWO ROWS OF BLADING

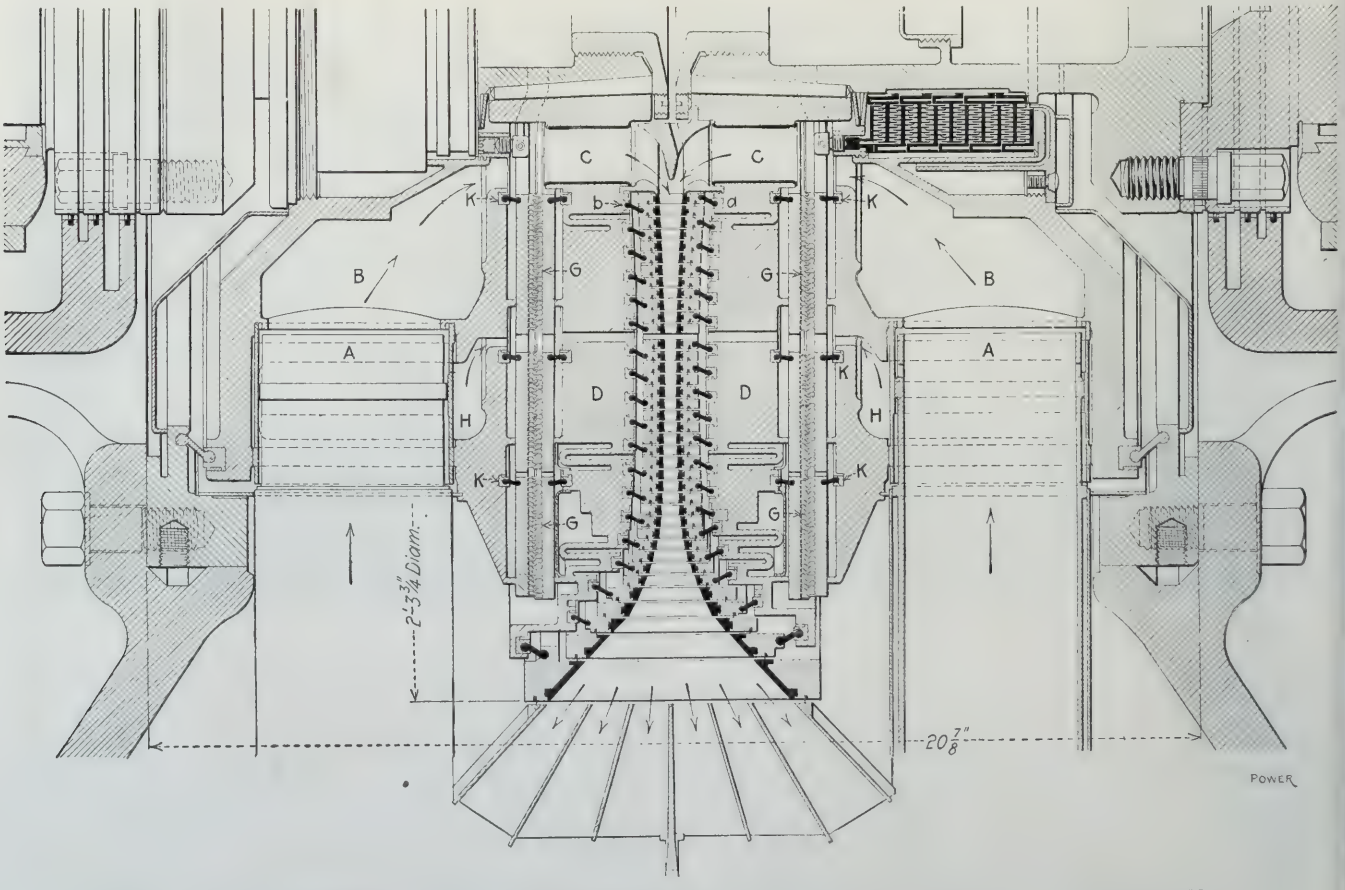


FIG. 1. THE LJUNGSTRÖM RADIAL-FLOW TURBINE. LONGITUDINAL SECTION, LOWER HALF OF 1000-KW. UNIT

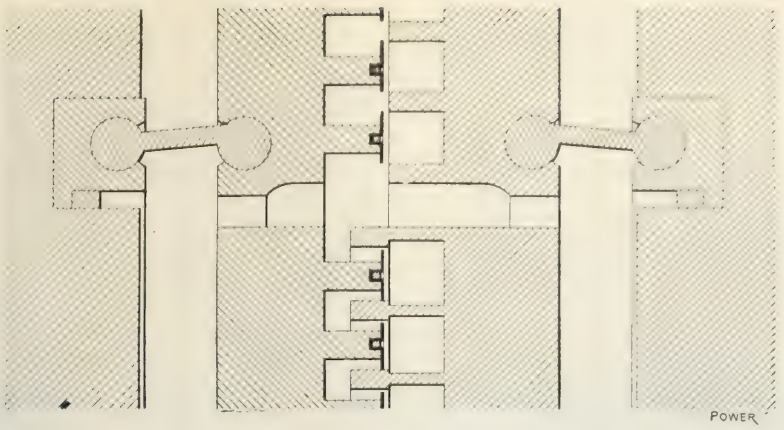


FIG. 3. DETAILS OF LABYRINTH PACKING

PARTICULARS OF TESTS OF 1000-KW. LJUNGSTRÖM STEAM TURBINE

Number of Test	1	2	3	4	5	6	7
Duration of Test in minutes	40	30	40	60	41.4	36.83	41.5
Revolutions per minute, both shafts	3000	3000	3010	3007	3018	3019	3019
Total load on generators, kw	972	788	480	1000.5	751	499.9	250.3
Total load on turbine, b.hp	1405	1150	723	1439.7	1093.2	746.2	403.7
Steam pressure before governor valve gage lb. per sq.in.	162	164	167	167	162	164	167
Steam pressure after governor valve gage, lb. per sq.in.	130	104	67	154	118	81	41
Steam temperature before governor valve, deg. F.	667	668	675	669	665	671	676
Steam temperature after governor valve, deg. F.	664	662	663	669	663	664	662
Superheat before governor valve, deg. F.	297	297	302	296	296	300	303
Superheat after governor valve, deg. F.	310	323	351	302	315	348	375
Absolute pressure in exhaust pipe, in Hg.	1.34	1.25	1.1	1.37	1.31	1.31	1.18
Absolute pressure condenser, in Hg	1.25	1.16	1.1	1.34	1.26	1.28	1.13
Temperature of condensed steam in measuring tanks, deg. F.	56	55.4	54	42.8	41.9	41.9	41.9
Temperature of cooling water, inlet, deg. F.				39.2	39.2	39.2	39.2
Temperature of cooling water, outlet, deg. F.				57.4	53.8	50.4	45.2
Temperature of oil leaving bearings, deg. F.				126	124	127	119
Steam condensed during trial, lb.	7429	4638	4123	11,550	6260	4010	2646
Steam escaping from shaft glands, lb.	45	28.6	26.4	106	61.6	42	26.8
Total steam used during trial lb.	7474	4666	4149	11,656	6321	4052	2763
Total steam used per kilowatt-hour	11.55	11.85	12.98	11.68	12.21	13.23	15.57
Total steam used per brake-horsepower-hour	8	8.13	8.61	8.1	8.39	8.85	9.6
Efficiency ratio, taking steam as in front of throttle valve	74.7	72.7	67.7	73.6	71.1	67.2	61.1
Efficiency ratio, taking steam as behind throttle valve	76.9	78.6	76.4	74.6	74.3	74.1	73.1

Bypass valves, either or both of which may be used, admit steam through the passages *H*, Fig. 1, for overloads. The pressure of the steam in the passage between the disks tends to force them apart and the end thrust thus produced is balanced by the dummy pistons *GG* attached to the disks *DD* by conical double bulb-headed rings similar to those

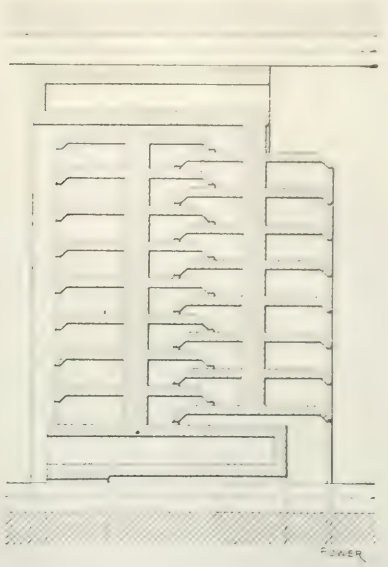


FIG. 4. LABYRINTH FOR PACKING SHAFT

used for fastening to the turbine casing at *KK*. Between them is a labyrinth packing, the details of which are shown in Fig. 3. The thin strips are of nickel, as in the case of the blades. Where the shaft passes through the casing, it has to be packed against the full initial pressure and a very effective labyrinth

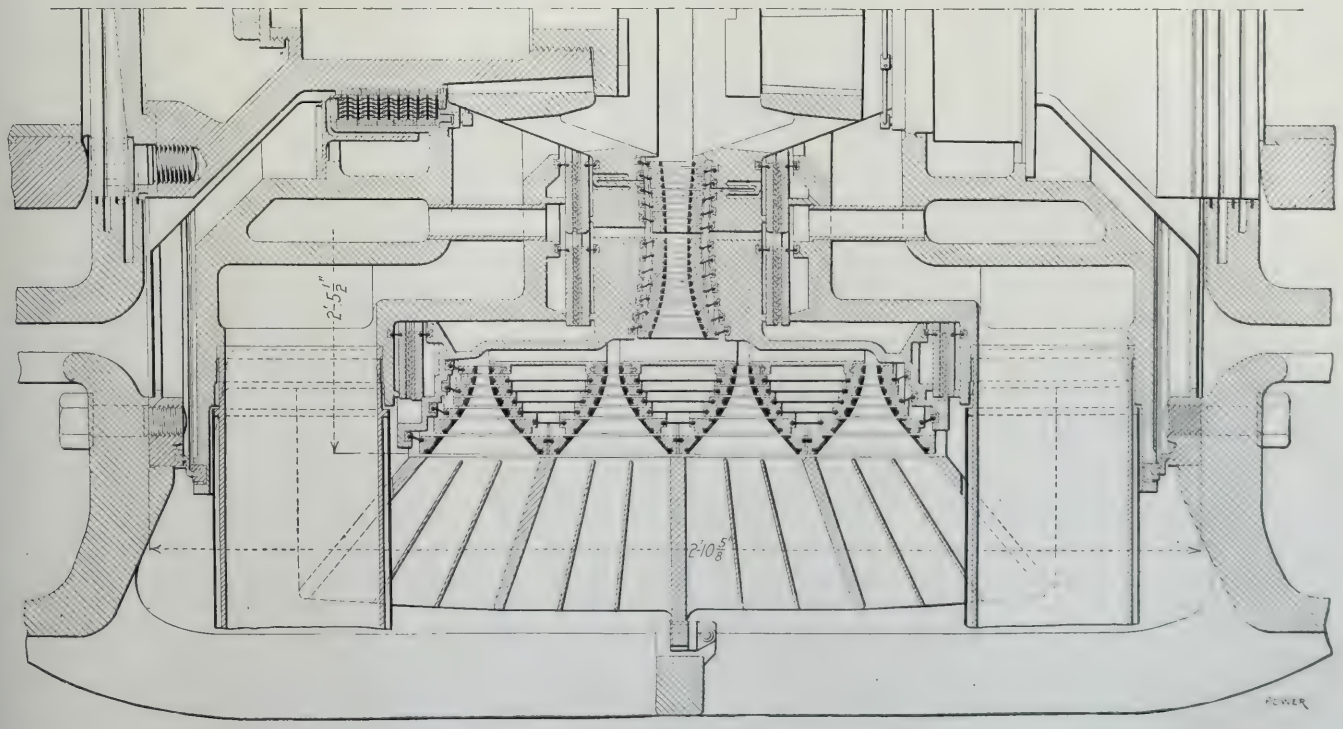


FIG. 5. SHOWING DIVISION OF LOW-PRESSURE STAGE

packing, made as in Fig. 4, is used. The rings are turned out of solid steel and designed to expand freely in all directions without distortion. The various rings are then spun over and made to just touch the rings which they embrace. A short run produces a working clearance. In 3¼ in. of shaft length they are able to get in 158 constrictions. Tests showed the leakage from both glands in the 1000-kw. machine to be but 110 lb. per hr. This leakage is led into a chamber in the side disks around the packing and thence to a feed-water heater or other point of application.

The 1000-kw. (at 3000 r.p.m.) machine is less than 17.5 ft. long, notwithstanding its two generators and exciter. Its greatest diameter is 4 ft. 2⅜ in. and its greatest height, to the top of the governor, 5 ft. 6 in. For large units the inventors propose to divide the low-pressure blading, as in Fig. 5, to get greater cross-sectional area in the low-pressure blading.

Tests of the 1000-kw. unit gave the results shown in the accompanying table. The first three were made on Oct. 10, 1911, by G. Oscar Olsson, M. I. N. A., and Axel F. Endström, of the Electriska Prüfungsanstatten, and the last four by Professors O. E. Lundström, of the Swedish Steam Boiler Association, and the Mr. Olsson above named.

Practical Features of Design

By J. E. Terman

In addition to embodying correct theoretical features a vessel should be of such design as to facilitate proper construction.

A few hints are given regarding tanks, flues and other work coming within the scope of the boiler shop.

When requiring special vessels to be built of boiler-type construction, it is best to go to a reputable boiler maker, who employs engineering talent, and let him know what is to be accomplished and allow him to get out details for the vessel. If this is not practicable or desirable, the designer should get up the design to the best of his ability and submit it to the manufacturer for criticism. In doing this, the manufacturer should be assured that his frank opinion is desired; for the experienced manufacturer invariably withholds adverse criticism of a design presented, to avoid offending a prospective customer.

Riveted joints are a part of the design which should not be attempted without practical knowledge. It may be simple to take some treatise on boiler calculations and ascertain how to proportion a joint for maximum strength, but all joints of maximum theoretical strength to resist rupture are not always practical. There should be certain relations between plate thicknesses, rivet diameters and the pitch of rivets, to produce joints that may be tightly calked and remain so in service. A design containing these qualities is imperative, regardless of the exact theoretical strength of the joint.

Openings in special vessels often cause entirely unnecessary expense in manufacture. When such openings need be only as large as an ordinary boiler manhole an 11x15-in. elliptical opening should be used; this is a standard, and reinforcements and cover plates for this shape are in nearly every boiler maker's stock. If the openings must be larger and retaining the simplicity of the manhole design is desired, the elliptical shape should be avoided, as patterns of that shape for both the opening reinforcement and the cover plate are expensive.

The shape shown in Fig. 1 has practically all of the advantages of an elliptical opening, and the circular ends simplify the pattern making. To make a pattern for the reinforcement of an opening, such as Fig. 1, a circular piece can be turned for the ends of the desired radius, and sawed in two, as indicated by the dotted lines, with straight pieces placed between these ends to get the desired length of opening. With a truly elliptical opening, the whole pattern would have to be worked out by hand at a great loss of time; the same applies to making the pattern for the cover.

When locating a connection to entirely drain a rectangular tank the connection should never be placed on the side if avoidable; a plain flat flange should be put on the bottom. Making a flange connection tight is very difficult when placed on the side in a position to completely drain the tank, as illustrated in Fig. 2. In the first place, fitting such a flange to the curve of the corner of the tank is difficult, and the rivets cannot tend to draw the flange tight against the sheet on both surfaces, which is necessary for good calking. The flange on the bottom is simpler, and it will generally be found better to sacrifice other features that might require the side connection rather than make a poor job.

A point always to be borne in mind when designing riveted vessels is that riveted joints must be calked to be made tight, and the design must be such as to make all joints accessible for calking, after the vessel has been constructed. This requirement is continually violated by the inexperienced designer in innumerable ways. Attaching a lug on top of an end seam, as illustrated in Fig. 3, is extremely bad practice, and if it is of cast iron, and cannot be calked down against the shell, making it tight is almost impossible; the head will interfere with any attempt to calk the rivet heads inside. A calking strip of plate material may be interposed between attachments of cast iron and the shell of the vessel, but this is not best, for it doubles the length of edge presented for leakage, and calking such strips tightly against both surfaces is also difficult.

If the lug is moved up, as in Fig. 4, and the supports are arranged to accommodate this position, the rivets need only be calked on the inside. Even here, having the lug of such material that it may be calked tight against the shell of the vessel, is better. Inside calking has to be done largely by guesswork, while if the external edges are calked, the effect can be noted under pressure and the leaks definitely located.

Fig. 5 shows a practically impossible construction as the girth seam under the lug cannot be calked. This seam could be made tight before attaching the lug, but there would be no certainty that attaching it would not disturb the calking edge. If the lug was broad this seam could not be reached for further calking after the lug had been attached. An almost impossible construction is represented in Fig. 6, because the longitudinal seam from *A* to *B* cannot be calked properly, due to the interference of the lug.

A designer should never specify exactly where joints should come in tank work; if they must not come at some places, it is best to designate these places and allow the manufacturer all the latitude possible. Very frequently vessels of this class are manufactured from stock material carried by the boiler shop, which is usually in standard-size sheets, and to make a tank with the seam locations specified generally requires a waste of plate, for which the purchaser has to pay.

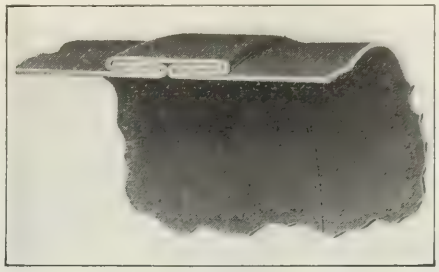
Often a vessel is designed for use as a receiver, such as shown in Fig. 7, with only small outlets for pipe connections. A good boiler maker can usually do many things that are seemingly impossible, but with a vessel of considerable size and small openings, riveting in one of the heads in a design, such as Fig. 7, is practically impossible. If there are reasons why an opening of suitable size to work through cannot be placed in such a vessel, one of the heads should be backed in as illustrated in Fig. 8.

The average designer of smoke flues who has not had shop experience usually attempts to produce a design which will theoretically offer the least resistance to the passage of the gases and be of proportionate area at every point to the volume of gas to be handled. This, from the standpoint of use alone, is commendable, but some consideration should be given to first cost and practical construction. A flue such as Fig. 9, although actually containing less metal than one like Fig. 10, is usually more costly, and the difference in the efficiency between the two would be very small. In rectangular flues, the thickness of stock and the spacing of the angle-stiffeners is important in insuring satisfactory performance in service, and only one who has had practical experience in such matters should be consulted to determine what is best.

Where two branches of a rectangular flue are brought together, as in Fig. 11, two pieces of sheet-iron baffle, bent and fastened in the flue at *CC*, afford an easy passage for the gases, prevent eddy currents and make a much stronger and cheaper flue to construct than that shown in Fig. 12. Straight lines should be followed as far as practicable in flue design, as every twist and change of shape presents more or less difficulty to the boiler maker and tends to prevent making a tight, strong structure which is essential to good service and can usually be constructed at less expense than some special shape.

Interlocking Spiral Pipe

The Standard Spiral Pipe Works, 25 North Dearborn St., Chicago, Ill., is placing on the market a reinforced galvanized spiral pipe, made of two steel strips of different widths and with a continuous interlocking seam. Both strips are rolled to shape on the edges and are interlocked and rolled down under very high pressure.



REINFORCED SPIRAL PIPE

The strips are first cut to the required width from long sheets and the ends are welded together as illustrated. The strip used for the reinforcing band comes in long lengths. These strips are fed into the machine at one end, rolled down and interlocked in the machine, coming out at the other end in the shape of pipe.

This pipe is smooth on the inside, with the reinforcing band running on the outside the entire length of the piece, and can be made endless. Ample folds are made at both strips so that a large bearing surface or contact is obtained at the seam. Where the reinforcing band passes around the pipe there are four thicknesses of metal. The pipe is supplied galvanized in any length up to 20 ft., and asphalted in any length up to 25 ft.

A slip-sleeve joint is made with a seamless sleeve welded into one end of the pipe. Lugs are also welded on the outside of both ends of the pipe. In making connections one end is slipped into the other and the pipes are then drawn together by wires over the lugs and a joint is obtained capable of withstanding a water pressure up to 60 lb.

Another style of connection is made with forged-steel flanges electrically welded on the ends of the pipe and furnished with bolts and gaskets complete. This pipe can also be supplied with cast-iron or forged-steel bolted joints. The pipe requires no soldering except where the ends are drawn together.

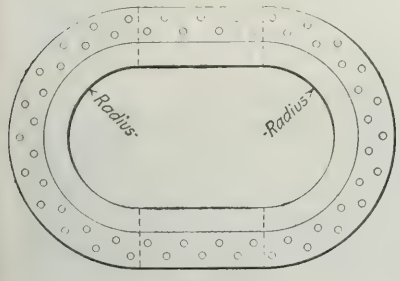


Fig. 1

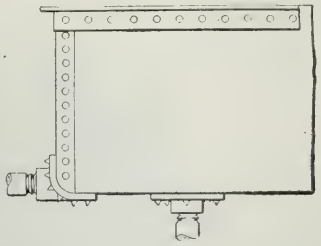


Fig. 2

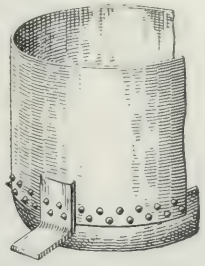


Fig. 3

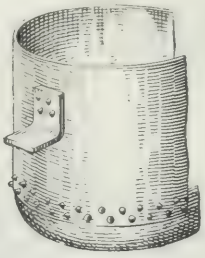


Fig. 4

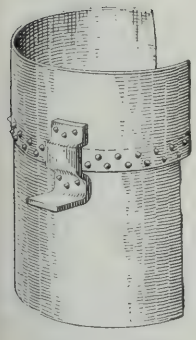


Fig. 5

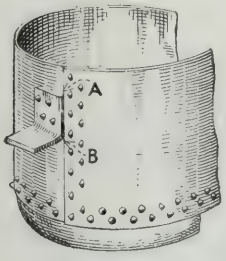


Fig. 6

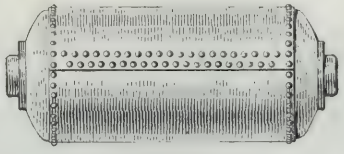


Fig. 7

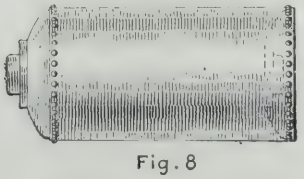


Fig. 8

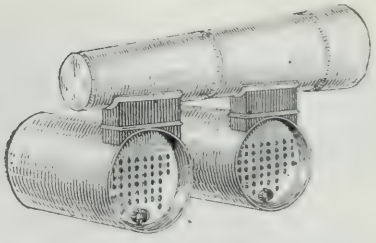
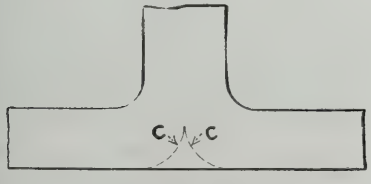


Fig. 9



Plan, Fig. 11

Plan, Fig. 12

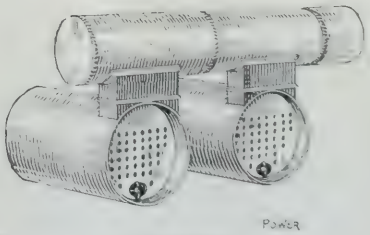


Fig. 10

Transporting a Boiler in the Tropics

By C. Carleton Semple

Describes the difficulties encountered in transporting an 80-hp. return-tubular boiler to a mine. Wooden walls, cement and fire brick were used for the boiler setting. Later a sheet iron casing replaced the wooden wall.

Several years ago the writer went to Nicaragua to take charge of a gold mine 90 miles from the coast and about one mile back from a small river, navigable only by shallow-draft dugout canoes or pit-pans. The river ran into the Caribbean at Prinzapolka, about 90 miles north of Bluefields, the nearest port of call of a line of fruit steamers.

The mine was in the midst of a dense forest, at low elevation, and in a part of the country where rains were heavy for nine months of the year. The road from the mine to the river bank was usually so muddy that one would sink to the knees in soft clay. Two rapids in the river added to transportation difficulties, and in places the water was so shallow that the canoes had to be dragged over smooth rock ledges in the river bed.

New Boiler Required

The mine steam plant consisted of a 30-hp. locomotive boiler and a slide-valve engine. Green wood fuel was used, because, due to the peculiar climatic conditions, if left to dry out and season the wood would rot. The firebox being too small to satisfactorily burn green fuel, and the tubes badly scaled, the boiler could not supply half the steam for which it was designed.

As it was intended to rebuild the mill and triple the capacity a new 80-hp. return-tubular boiler was ordered, which, with the old boiler retubed, would generate all the steam needed. A return-tubular boiler was desirable as it permitted enlarging the firebox to any desired extent, no skilled bricklayers are required for its setting and it is easily cleaned. The last feature was important as the water was bad.

An 80-hp., 60-in. by 16-ft. return-tubular boiler weighs over 11,000 lb., which was almost double the weight of the heaviest single piece of machinery that had theretofore been taken into that part of the country. The largest canoe on the river was about 45 ft. long, with an 8-ft. beam and capable of carrying 14,000 lb. Such a load had often been carried, but never in a single piece, and never with the center of gravity high above the gunwales. The transportation was considered feasible and, although the contractors would not guarantee safe delivery, the job was undertaken at the usual tariff of $2.20 per 100 lb. from Bluefields to the mines.

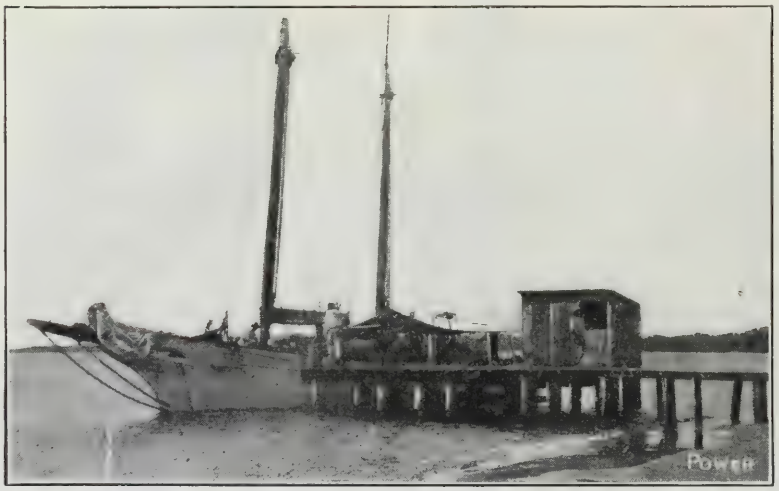


Fig. 1. Arrival of the Boiler at Prinzapolka

Fig. 2. At the River Bank, 25 Ft. High

Transporting the Boiler

When the boiler arrived at Bluefields it was loaded on one of the small gasoline auxiliary schooners of from 35 to 50 tons burden which ply the coast between Bluefields and Prinzapolka, and was placed athwartship just abaft the foremast, Fig. 1. After a calm night on the sea it arrived at Prinzapolka the following morning, where it took a day's hard work to remove the boiler from the schooner to the dock and from the dock into a canoe manned by 17 Mosquito indians, and 17 days later reached the mine. At Walpatara Rapids, several ropes were let out from the bow and on each side amidships, and the canoe was warped over the shallows, as shown in Fig. 3.

The river bank was fully 25 ft. high at the mine, Fig. 2. In hoisting the boiler up this bank, four heavy, wooden ship blocks, 100 ft. of 2-in. and 500 ft. of 1-in. manila rope were used. A jib crane was rigged up at the river bank, using a stout ironwood boom and a strong growing tree as a mast, Fig. 4. It was intended to sling the boiler with 2-in. rope and raise it by a ⅝-in. steel rope, using half a dozen yoke of oxen to hoist the boiler from the canoe to the level of the bank and then swing it in. This plan was changed when the steel rope "turned up missing" with only 1-in. rope to take its place. Four blocks and falls were rigged with this small rope and four teams of oxen of two yokes each were used, one team on each rope. They managed to lift the boiler out of the canoe, but could not be made to pull in unison and the scheme had to be abandoned. The 125 mine men were called to man the ropes and the boiler was gradually raised to the top of the bank after an entire morning's work.

More Difficulties

To haul a boiler of such weight over a mile of bush road, so muddy that the oxen sank to their knees, was a task. A wheeled vehicle was out of the question; skids alone could be used. The forest was searched for a tree that forked near the ground into two branches of equal diameter. When found, the small branches were trimmed off and the trunk cut 2 ft. below the fork, and the end of each branch at a distance of 18 ft. from the fork.

This gave a V-shaped sled, which, after trimming the front or trunk end upward, and smoothing the bottom side of the branches, readily slid along the muddy

road. The ends of the two branches were drawn together with a tie-rod, and cross-pieces spiked on the upper sides. The boiler was loaded on these cross-pieces and held in place by triangular blocks nailed to them. A snatch-block rigging and two yoke of oxen brought the boiler to the mine in about six hours.

Wood and Cement Boiler Setting

At this particular mine the cost of freight on bricks would have been 3c. per lb. The writer had once before obtained satisfactory results in setting a boiler in concrete, lining the walls with firebrick, and resolved to use the same setting here. The firebricks were laid up in a claywash, forming the inner wall of the setting. The outer form was built of 2-in. plank and concrete filled in between the two walls. If cement is shipped in galvanized-iron drums it can be used as in the temperate zones, but unfortunately the cement used for the boiler walls was shipped, contrary to instructions, in sacks and barrels. Cement traveling for several weeks through a damp, hot climate greatly deteriorates and often becomes practically useless.

Naturally the boiler walls did not last long, and in a few months began to crumble and fall away. Finally they were removed altogether and it became a question of what material could be used in place of brick or masonry. Hard stone suitable for stone walls was not to be found in the vicinity. It was, therefore, decided to build a wooden frame around the sides and back of the boiler and as high as the old concrete wall had been. Then, between the firebrick lining and this wooden wall powdered clay, that had been dried in the sun, was tamped, which made a fairly good backing for the firebrick lining, and as the space between the wooden walls and the lining was 8 or 10 in., the heat reaching the wooden walls was not sufficient to cause charring. This setting is shown in Fig. 5.

Had the clay been placed between the brick lining and the wooden walls while in a moist condition, trouble would soon have been experienced from cracking as the moist clay dried, which would have permitted the board walls to burn and would also have admitted so much air as to interfere seriously with the combustion. The wooden walls were used for two years before they had to be replaced with some old sheet-iron plates, which had no use. These were riveted together and used instead of the wooden walls. Some trouble was experienced with this setting because the heat that did penetrate the clay caused warping.

The cost of this boiler, as installed was about $2250. The boiler cost $900; the cost of transportation to the mine and setting made up the remainder. This is at the rate of approximately $28 per hp.

Fig. 3. Warping Over the Rapids

Fig. 4. Unloading the Boiler

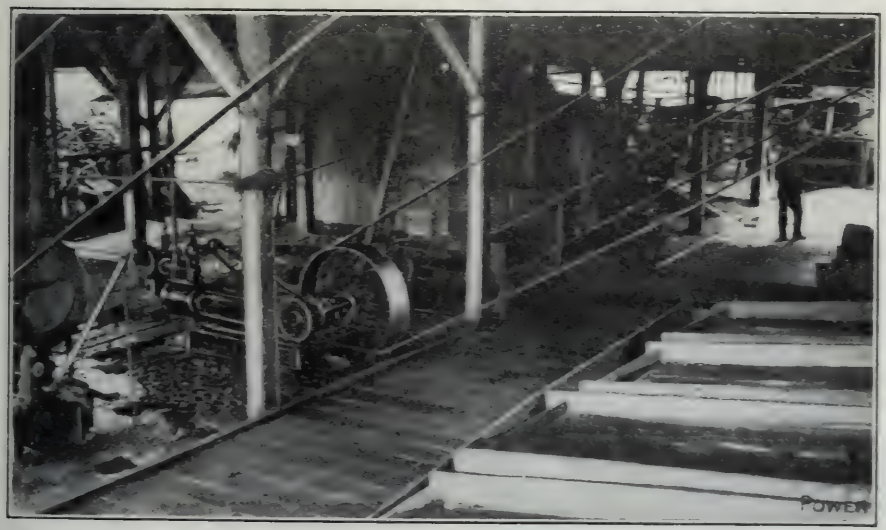


Fig. 5. View of the Steam Plant, the New Boiler and Wooden Setting Beyond the Engine

Electrical Disturbance on Shipboard

The growing use of electricity on board ships has had such a disturbing effect on the accurate working of chronometers that Capt. Cust, hydrographer to the British Navy, recently requested the assistance of the magnetic department at Greenwich Observatory in discovering how best to counteract the influence of a vessel's dynamos on its Greenwich time indicators. The experiments at Greenwich have been successful, and some remarkable variations in the magnetic effects have been discovered, according to the position in which the balance of a chronometer or watch is placed relative to the points of the compass. It was also found that a chronometer in the vicinity of a magnetic field is quite unaffected if its balance moves through a regular arc of 210 deg.

Mexico has granted a 99-year concession to an American company which permits taking water from the Santa Maria and Gallinas rivers for the generation of electricity.

U. S. Census Distribution of Power

By Davis H. Tuck

Ninety per cent. of the 18,760-686 hp. generated in the United States in 1909 was developed by manufacturers. Its distribution is given in the tables. Steam is the largest power source. The use of gas engines and electric power has strikingly increased.

In 1909, it is reported in the thirteenth United States census, 18,760,686 primary horsepower was generated or rented by manufacturing establishments in the United States. Table 2 shows the distribution.

Ninety per cent. of the horsepower in 1909 was that of engines or motors owned by the manufacturing establishments themselves, and 10 per cent. was rented power, mostly electric. Especially striking is the increased use of gas engines and of electric power, both that rented from outside concerns and that generated by the manufacturers. The total horsepower of electric motors in 1899, including those operated by purchased current and by current generated in the establishment, was 492,936; in 1909 it was 4,817,140, nearly ten times as great.

Renting electric power is rapidly becoming more common among small establishments, while there is a tendency for the large concerns to use electric motors for applying power which they themselves generate.

The largest amount of electric power is used by the steel works and rolling mills, the next largest being the foundries and machine shops. In the former the electric power is a little over one-third of the total amount of primary power and in the latter nearly three-fourths.

Table 1 shows, in geographic divisions,

TABLE 1

Division and State	Total horsepower (excluding duplication)	Owned by Establishments Reporting					Rented		Electric Motors	
		Steam engines	Gas engines	Water wheels	Water motors	Other	Electric motors	Other	Total [1]	Run by current generated by establishment
United States	18,680,776	14,202,137	754,083	1,807,144	15,449	29,293	1,749,031	123,639	4,817,140	3,068,109
Geographic divisions:										
New England	2,715,121	1,656,911	44,451	751,270	3,412	2,055	218,642	38,380	663,143	444,501
Middle Atlantic	5,531,502	4,151,662	274,274	466,541	3,947	11,736	568,723	54,619	1,737,236	1,168,513
East North Central	4,382,070	3,491,418	283,450	206,393	2,048	4,766	375,876	18,119	1,297,447	921,571
West North Central	1,101,990	838,988	57,434	82,791	3,539	939	115,002	3,297	266,534	151,532
South Atlantic	1,837,401	1,434,221	36,688	184,431	1,082	5,321	171,146	4,512	343,393	172,247
East South Central	1,036,560	953,511	12,270	29,040	275	1,690	38,580	1,194	108,409	69,829
West South Central	873,350	805,640	29,291	3,060	48	2,513	31,807	991	78,893	47,086
Mountain	400,766	306,786	4,188	21,345	198	224	66,956	1,069	113,984	47,028
Pacific	802,016	563,000	12,037	62,273	900	49	162,299	1,458	208,101	45,802
New England:										
Maine	459,599	168,595	6,583	253,830	1,912	179	27,203	1,297	54,266	27,063
New Hampshire	293,991	139,128	1,238	127,496	521	30	21,209	4,375	45,351	24,142
Vermont	159,445	64,252	2,160	78,881	181	415	12,917	639	21,233	8,316
Massachusetts	1,175,071	834,701	18,326	185,996	520	895	109,996	24,637	402,492	292,496
Rhode Island	226,740	175,293	3,300	31,376	41	39	13,697	2,994	42,130	28,433
Connecticut	400,275	274,942	12,844	73,697	237	497	33,620	4,438	97,671	64,051
Middle Atlantic:										
New York	1,997,662	1,080,877	99,899	394,221	1,397	3,583	389,945	27,740	689,976	300,031
New Jersey	612,293	529,668	20,867	18,558	1,118	180	33,157	8,745	182,475	149,318
Pennsylvania	2,921,547	2,541,117	153,508	53,762	1,432	7,973	145,621	18,134	864,785	719,164
East North Central:										
Ohio	1,583,155	1,362,134	103,801	15,777	330	1,586	93,592	5,935	417,844	324,252
Indiana	633,377	448,528	109,105	7,446	447	599	65,548	1,704	233,193	167,645
Illinois	1,013,071	838,199	37,025	12,178	513	1,433	117,007	6,716	398,621	281,614
Michigan	598,288	465,520	13,988	41,442	577	16	74,270	2,475	133,064	58,794
Wisconsin	554,179	377,037	19,531	129,550	181	1,132	25,459	1,289	114,725	89,266
West North Central:										
Minnesota	297,670	199,777	7,174	56,631	2,939	25	30,297	827	52,212	21,915
Iowa	155,384	121,882	8,025	6,326	85	147	18,463	456	40,736	22,273
Missouri	340,467	280,489	11,159	3,532	206	5	44,056	1,020	106,941	62,885
North Dakota	13,196	10,170	1,304	530	.....	.....	1,164	28	1,698	534
South Dakota	17,666	12,257	2,784	927	12	.....	1,683	3	2,084	401
Nebraska	64,466	44,806	4,408	7,361	75	76	7,530	210	15,942	8,412
Kansas	213,141	169,607	22,580	7,484	222	686	11,809	753	46,921	35,112
South Atlantic:										
Delaware	52,779	42,266	766	5,183	12	.....	4,502	50	17,910	13,408
Maryland	218,244	181,326	5,736	11,953	121	1,069	17,108	931	44,921	27,813
District of Columbia	16,563	12,169	1,073	775	.....	43	2,433	70	4,527	2,094
Virginia	283,928	221,303	3,664	45,122	33	38	13,356	412	42,043	28,687
West Virginia	222,896	187,389	16,952	12,901	71	.....	5,330	253	28,543	23,213
North Carolina	378,556	271,944	2,356	41,619	307	1,035	60,044	1,251	86,002	25,958
South Carolina	276,378	193,052	1,264	38,422	75	2,400	41,130	35	67,620	26,490
Georgia	298,241	240,264	3,380	28,288	460	536	23,890	1,423	44,264	20,374
Florida	89,816	84,508	1,497	168	3	200	3,353	87	7,563	4,210
East South Central:										
Kentucky	230,224	207,591	4,724	5,320	57	915	11,314	303	31,268	19,954
Tennessee	242,277	215,338	1,853	9,670	107	4	14,666	639	29,586	14,920
Alabama	357,837	328,275	4,616	13,812	111	732	10,104	187	39,928	29,824
Mississippi	206,222	202,307	1,077	238	.....	39	2,496	65	7,627	5,131
West South Central:										
Arkansas	173,088	168,152	1,374	639	35	52	2,581	255	7,417	4,836
Louisiana	346,652	331,370	3,496	65	10	2,401	9,077	233	27,139	18,062
Oklahoma	71,139	56,643	8,676	470	2	.....	5,281	67	7,887	2,606
Texas	282,471	249,475	15,745	1,886	1	60	14,868	436	36,450	21,582
Mountain:										
Montana	90,402	49,654	223	13,583	63	.....	26,504	375	27,301	797
Idaho	42,804	35,529	242	2,403	4	.....	4,606	20	8,409	3,803
Wyoming	7,628	6,467	182	456	9	.....	514	.....	801	287
Colorado	154,615	135,645	1,464	1,377	49	105	15,874	101	35,944	20,070
New Mexico	15,465	11,781	365	74	.....	.....	3,245	.....	4,586	1,341
Arizona	39,140	34,193	1,285	129	.....	19	3,314	200	15,100	11,786
Utah	42,947	28,984	226	2,926	71	100	10,592	48	15,402	4,810
Nevada	7,765	4,533	201	397	2	.....	2,307	325	6,441	4,134
Pacific:										
Washington	297,897	257,230	1,494	7,842	223	19	30,951	138	43,615	12,664
Oregon	175,019	112,244	428	47,041	397	.....	14,811	98	20,802	5,991
California	329,100	193,526	10,115	7,390	280	30	116,537	1,222	143,684	27,147

[1] Includes the horsepower of motors run by rented current and also of those run by current generated by the establishment.

the amount of each of the several kinds of power used in manufacturing industries in 1909. The relative total amount used is largely dependent upon the character of the industries predominant in each division or state, and the extent to which the different kinds are used depends upon the character of the industries and the situation of each state as to supplies of coal, petroleum and gas and the availability of water power.

In every state except Maine and Vermont, steam engines are the most important source of power. The proportion which gas-engine power represents of the total is larger in the East North Central division than in any other division, partly on account of the proximity of gas wells. The Middle Atlantic States rank next in the proportion of the total developed by gas engines.

For power obtained by waterwheels owned by the manufacturing establishments, New England ranks far ahead of the other divisions both in the absolute amount of power and in the proportion which water power represents of the total. More than two-fifths derived from waterwheels owned by manufacturing establishments is found in New England, and more than one-fourth of the power of the New England factories is derived from waterwheels. The Middle Atlantic division ranks next.

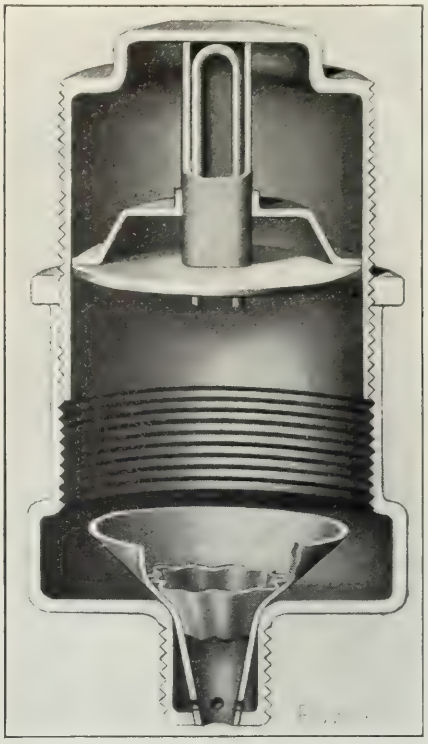
The largest absolute amounts of power of electric motors (including both those operated by current generated in the establishment and those operated by purchased current) are reported from the Middle Atlantic, the East North Central and New England divisions.

The leading individual states using gas engines to develop power are Pennsylvania, Indiana, Ohio, New York, Illinois, Kansas and New Jersey. In the use of waterwheels to develop power the states are New York, Massachusetts, Wisconsin, New Hamphsire, Vermont, Connecticut, Minnesota, Pennsylvania, Oregon, Virginia, North Carolina and Michigan.

In the absolute amount of electric power for manufacturing, Pennsylvania leads, followed by New York, Ohio, Massachusetts, Illinois, Indiana and New Jersey.

Lewis Compressed Air Grease Cup

This grease cup consists of a body with internal threads into which the top part of the cup screws. On the interior is a saucer-shaped disk, held firmly in



COMPRESSED AIR GREASE CUP

the top of the cup by an oval-shaped tube. In this tube a wire staple slides and is connected with and guides a movable disk. In the bottom of the cup is a funnel, corrugated where it comes in contact with the top of the shank. At the extreme bottom of this funnel, inside the shank, are a number of small holes. When the cup is full of grease, and the top is screwed into the body, the air is compressed in the top of the cup, which creates a pressure, or feeding force. The flat portion of the saucer-shaped cup prevents any grease passing the disk should the cup be screwed on too far. The funnel in the body directs the grease to the outlet.

After the top has been screwed down its full length and the disk rests on the funnel, the cup should be refilled, but if this is not done, the grease around the funnel is forced through the corrugations and small holes at the bottom by the air pressure passing around the disk and acting directly on the grease. This cup is manufactured by the Lewis Grease Cup Co., 1325 Arch St., Philadelphia, Penn.

The Swain Bearing Candle Lubricator

This lubricator consists of a metallic tube, open at one end, and fits into a hole in the bearing cap. The top end has a removable cap through which an indicating rod projects. This rod is fastened to a weight which rides on a lubricating candle and indicates the length of the candle in the tube. The lubricating candles are made in various degrees of hardness to suit different requirements.

The lubricator is manufactured by the Swain Lubricator Co., 328 West Lake St., Chicago, Ill.

"Twenty-nine college professors from 25 different colleges in 19 states carry their dinner pails and earn 20c. an hour in local industrial plants. They have forfeited their vacations that they might get more practical experience," says a Pittsburgh dispatch. These young "profs" have charge of the engineering departments of their respective schools. Three cheers for their spunk, and three times three for their common sense. Theory & Practice are doing business at the old stand.

TABLE 2

Power	Number of Engines or Motors			Horsepower			Per Cent. Distribution of Horsepower		
	1909	1904	1899	1909	1904	1899	1909	1904	1899
Primary power, total	408,472	231,363	168,143	18,680,776	13,487,707	10,097,893	100.0	100.0	100.0
Owned	209,163	169,774	168,143	16,808,106	12,854,805	9,778,418	90.0	95.3	96.8
Steam	153,482	127,267	130,710	14,202,137	10,825,348	8,139,579	76.0	80.3	80.6
Gas	34,352	21,515	14,334	754,083	289,423	134,742	4.0	2.1	1.3
Water wheels	20,126	19,595	23,099	1,807,144	1,641,949	1,454,112	9.7	12.2	14.4
Water motors	1,203	1,397	(1)	15,449	5,931	(2)	0.1	(2)	(1)
Other	.....	.....	.....	29,293	92,154	49,985	0.2	0.7	0.5
Rented	199,309	61,589	(1)	1,872,670	632,902	319,475	10.0	4.7	3.2
Electric	199,309	61,589	(1)	1,749,031	441,589	182,562	9.4	3.3	1.8
Other	.....	.....	.....	123,639	191,313	136,913	0.7	1.4	1.4
Electric motors	388,854	73,119	16,891	4,817,140	1,592,475	492,936	100.0	100.0	100.0
Run by current generated by establishment	189,545	73,119	16,891	3,068,109	1,150,886	310,374	63.7	72.3	63.0
Run by rented power	199,309	(1)	(1)	1,749,031	441,589	182,562	36.3	27.7	37.0

[1] Not reported.

[2] Less than one-tenth of 1 per cent.

Deane Motor Driven Triplex Mine Pump

Modern mining operations are carried on at such depth that the problem of unwatering the works is of interest, and special equipment has been developed to deal with its solution. The application of electrical energy in mine work simplifies the work where the cost of fuel is high and reduces the cost of operation.

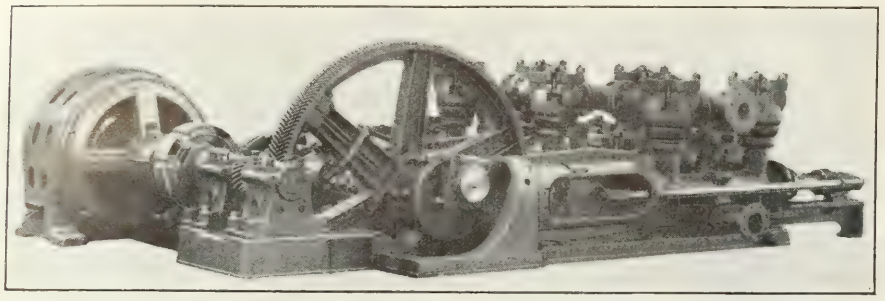


FIG. 1. MOTOR AND DRIVING GEAR

The accompanying illustrations represent recent development in large mine-pumping units. The pump shown is of the horizontal, triplex, double-plunger, pot-valve type. The machine is so designed as to permit its being lowered through a shaft 3 ft. 4 in. by 4 ft., but all parts subject to wear or breakage are so segregated as to facilitate replacements.

A single reduction of double-helical or herring-bone gears, Fig. 1, transmits power from the motor to the pump crankshaft with a speed reduction from 580 to 63 r.p.m. The motor is of the two-speed type to provide flexibility in the total capacity of the station; flexible couplings are used to secure smoothness of operation. This form of drive gives an estimated mechanical efficiency of over 85 per cent. of the energy applied at the motor.

The crankshaft and connecting-rods are steel forgings. The crankshaft bearings are of the chain-oiling type. All working parts are accessible for wear, but the adjustments do not require delicate attention.

The water end of the pump, Fig. 2, is made with deep-plunger stuffing-boxes. All valves are accessible from above and the handhole covers on the pots are held by swing bolts. The suction and discharge pots are inter-changeable. The flow of the fluid is as nearly direct as is possible, thereby permitting satisfactory operation at high rotative speeds. The glands and stuffing-boxes have renewable bronze bushings.

The principal data regarding this pump are:

Diameter of plunger, 4½ in.;
Length of stroke, 12 in.;
Speed of pump, 63 r.p.m.;
Capacity of pump, 300 gal. per min.;
Working head, 715 lb., or 1650 ft.;
Motor, 175 hp., 2200 volts, three-phase, 30-cycle;
Speed of motor, 600 and 300 r.p.m.

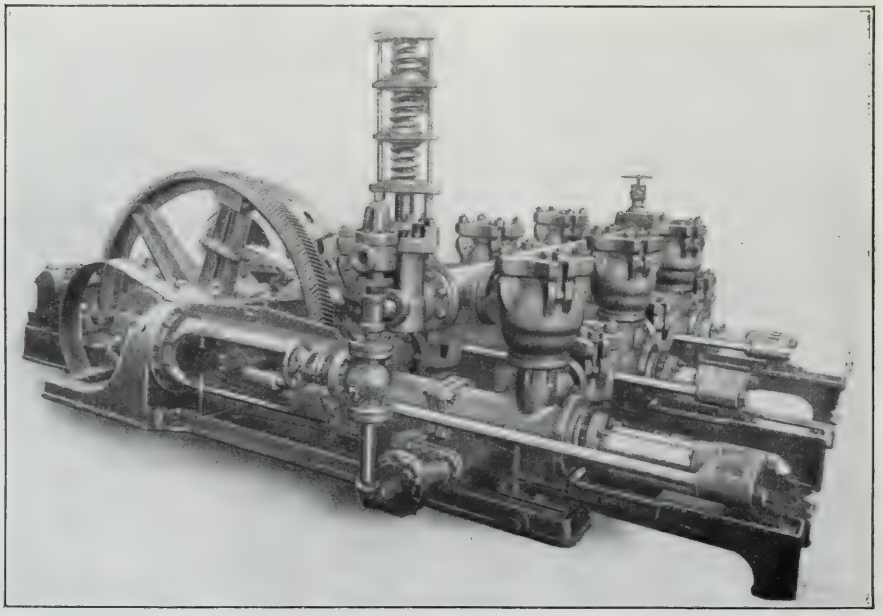


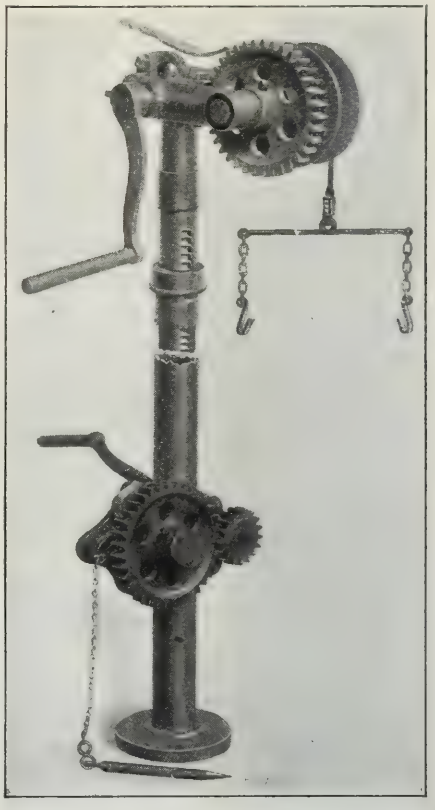
FIG. 2. WATER END OF PUMP

Two of these machines, built by the Deane Steam Pump Co., Holyoke, Mass., are being installed in a Colorado mine at the 1650-ft. level. Both machines deliver through a common column pipe to the surface.

G. & G. Ash Hoist

Loaded ash cans, barrels, boxes, etc., up to 500 lb. weight can be raised on the hoist herewith illustrated. It can be arranged so that the platform can be raised 5 or 6 ft. above the grade level, so as to empty ash cans directly into wagons without rehandling.

The hoist is adjustable, and no part of it shows above the sidewalk when it is not in use. Raising and lowering is made easy by the use of compound gearing, and a brake attachment permits the lowering of heavy loads without trouble. A safety ratchet device prevents the extension falling when being raised or lowered. Gillis & Geoghegan, 537 to 539 West Broadway, New York City, build this hoist.



G. & G. ASH HOIST

When you were a brash young fireman, did you ever scoop the other scooper's scoop when your own looked like the ragged edges of Despair and the other fellow's like a soft day's labor? A Western railway claims to get better service from its firemen with the individual scoop. How do you think such an arrangement would work out in a boiler room?

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Determining Neutral on Interpole Machines

By J. S. Kinney

To operate satisfactorily, an interpole motor or generator must have its brushes set exactly on neutral; if not, the commutation will be poor. A method for putting the brushes on neutral, which has recently come into favor, especially on large machines, is to separately excite the interpole field, and shift the brush-supporting ring until there is no voltage between adjacent brush arms.

The procedure is as follows: The spacing of the brushes is first checked by wrapping a narrow strip of paper around the commutator under the brushes and marking along the front of each brush. If there is more than one brush per arm, the spacing is checked for one brush on each arm, and the other brushes on the arm are checked to this one. After removing the paper, the distance between the marks are measured, and if not equal, they should be made so by adjusting the brush arms or holders. All the brushes are raised and in one holder on each arm is placed a brush which has had the contact edge beveled, care being taken to grind off the same amount on each side.

The machine is then driven at approximately its rated speed, by a motor, or other means.

With no field on the machine, a low reading voltmeter, say 15 volts full scale deflection, is connected across two adjacent brush arms to measure the voltage due to residual magnetism. This residual magnetism must be reduced to zero by separately exciting the shunt field through a high resistance and closing the field switch for an instant. The voltmeter should be removed from the circuit while this is being done, as the inductive kick would be likely to injure the meter. If, on replacing the voltmeter in circuit, the voltage due to residual magnetism is found to be greater than before, it is evident that the current through the field was in the wrong direction, and will have to be reversed.

After the residual field has been reduced to zero, the interpole field is separately excited with from 2 to 4 per cent. full-load current, and the reading of the voltmeter connected across the arms is noted. The interpole field is then removed and the residual magnetism, if there is any, is reduced again. The interpole field is again excited, with the same strength as before, and the brushes shifted until the voltmeter reading is practically zero. The residual magnetism must be kept down to nearly zero while the brushes are being adjusted. The final adjustment should be made with a voltmeter having a full-scale deflection of about 1½ volts.

The theory underlying this method is that, when the interpoles are excited, and the brushes on neutral, equal and opposite voltages are induced in the armature conductors under each pair of poles.

This method has the advantage over the "kick" method, in that an average neutral position for all the brush arms is found, while by the "kick" method, the neutral must be found for each arm separately, and a guess made as to the correct average position.

Edison Bipolar Machines as Railway Boosters

By L. J. Freed

Edison bipolar generators are being replaced by modern multipolar machines, and are fast going to the scrap pile. This has been their fate in the plant where the writer is employed, except four, which have been utilized as railway boosters. These are 60-kw., 125-volt, shunt-wound machines, but to meet the requirements of railway service a series field winding was added, consisting of 32 turns of No. 4/0 weatherproof wire, per pole; this winding was arranged with a single-pole double-throw switch to permit it being cut in and out of service. The four machines with a 300-kw. alternator, were belted to a 350-hp. engine, having a double-face flywheel, which carried the five belts, one running over the other.

To use the four machines as boosters for railway service, the series field winding is cut in and by referring to the diagram of connections, the booster circuit can be traced from the 550-volt busbar to the bottom throw of the switches in group *A* to the negative side of the booster, through the armatures and series fields (the switches of which are in the down positions) to the terminals 1, 2, 3 and 4 of group *A*. From here the circuit is carried to the corresponding terminals of switch group *B*, thence to the lines. When the switches of group *C* are in the up position the lines are fed direct from the 500-volt busbars, which is the case when no booster is required. When the switches of group *B* are in the down position the regular multipolar booster is in service.

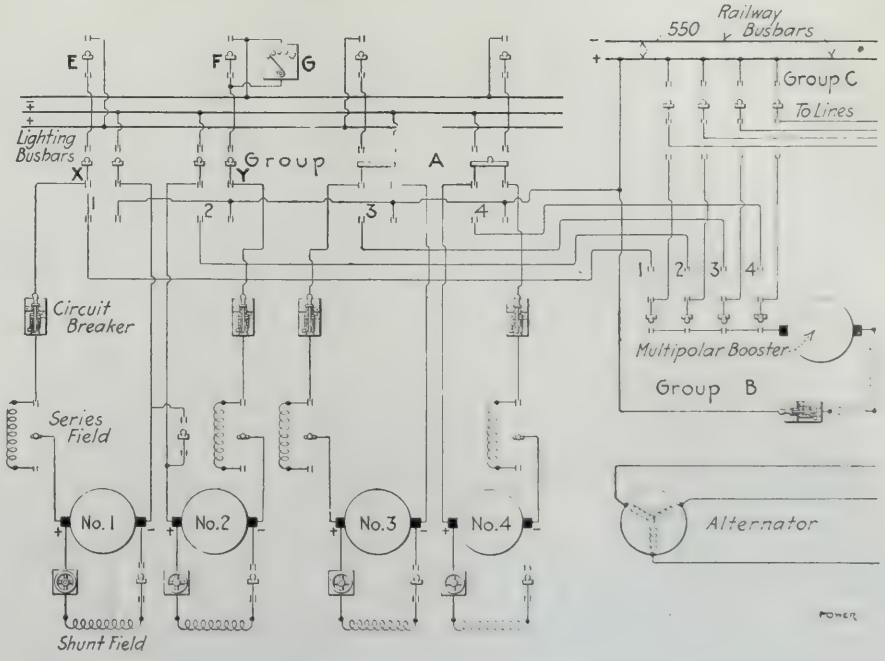


Diagram of Connections

To use the machines on the lighting system, which is an Edison 3-wire, 220-volt system, the series field is cut out by throwing its switch in the up posi-

tion, the shunt field switches are then closed and the machines are built up to the proper voltage. The switches of group *A* are then closed upward, connecting two machines in series. Next the switches *E* and *F* are closed, throwing the machines on the busbars. The same operation holds good for the other pair.

In summer it is customary to unhook the engine by taking off the connecting-rod and eccentrics, and drive the set by the alternator running as a synchronous motor. The set is started and run in the following manner:

Machines Nos. 1 and 2 are started as motors in series across the 220-volt busbars, the switches marked *X* and *Y* being closed in the up position, and the switch *D* is closed; this connects the machines in series. The shunt field resistance is all cut out and the positive switch *E* is closed. On the negative side a rheostat *G* is used in starting. It should be noted that these machines are belted to the alternator. The machines are brought up to speed (synchronous speed for the alternator) by the field rheostats and the alternator is then synchronized to the alternating-current busbars, not shown in the diagram. The alternator is now being run as a synchronous motor, driving the set. In winter the engine is connected again and drives the set.

In this way we have an emergency booster and lighting set besides a spare alternator for alternating-current purposes. This set has served for alternating-current lighting and railway boosting at the same time without any trouble, although not intended to do this.

CORRESPONDENCE

Repairing a Motor-Generator Set

In Fig. 1 is shown the plan view of a three-bearing motor-generator set, consisting of a three-phase induction motor directly connected to a direct-current generator. The motor has the bearings, one on each end frame, and the generator has one bearing at the commutator end. A short-circuit developed in the motor windings and the blaze that followed was fanned by the rotor about one-third of the circumference of the stator coils, damaging them beyond immediate repair.

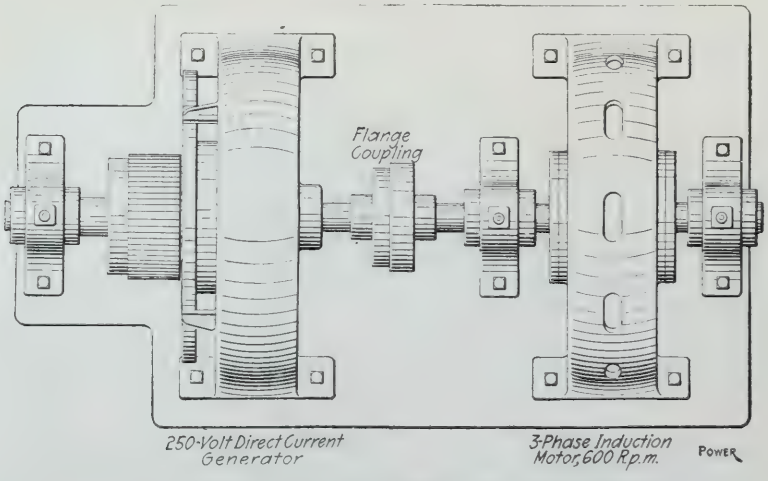


FIG. 1. PLAN OF MOTOR-GENERATOR SET

The problem was to repair the damaged coils and at the same time operate the generator. It was decided to drive the generator by a belt from a spare motor. Accordingly, the rotor of the motor with its shaft and coupling was removed, which left the armature of the generator unsupported at one end. As the rotor coupling had been shrunk on and was not easily removable, a new coupling and a short piece of shaft was made up, this shaft resting in the bearing of the motor after being bored out to suit the coupling only one pair of the original hub-clamping bolts remained. These were insufficient to hold or drive the pulley; therefore the driving was done by the arms indicated in the end view of Fig. 2. Extra holes were drilled through the couplings to take four bolts for the driving arms of 3x⅛-in. flat iron; the outer ends being fastened to the pulley rim by lag screws.

This pulley ran at a rim speed of over 5000 ft. per min. and carried a varying load up to 100 hp. This arrangement left the interior of the motor empty and permitted a workman to repair the damaged coils.

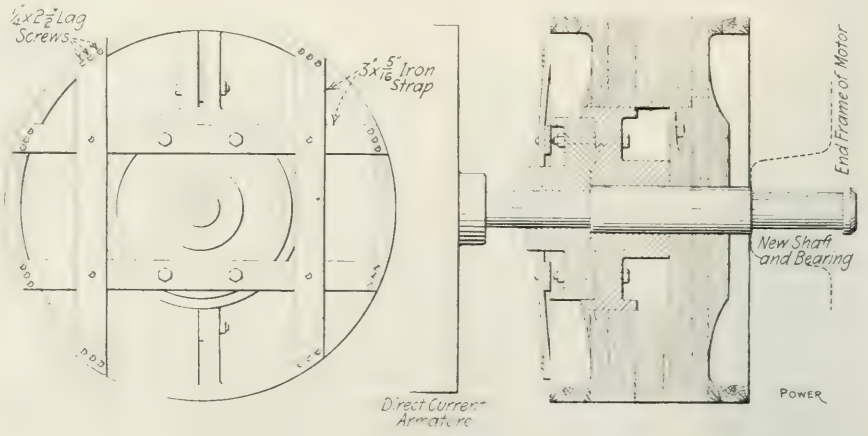


FIG. 2. METHOD OF ATTACHING COUPLING

GEORGE POWERS.

Canton, Ohio.

Generator Trouble

We have two 500-volt direct-current generators running in parallel, but which are about 4 miles apart; one is a 150-kw. and the other a 500-kw. machine. The larger generator frequently changes its polarity and causes trouble as the machines supply a trolley system. One side is grounded and where this machine changes over its positive side is connected to the rails. This does not appear to affect the load on the larger machine, but runs the load up on the smaller machine so that the circuit-breaker invariably goes out. We use sectional insulation with 6 in. of fiber or wood, and the current often jumps across.

When this happens we have to pull the section switches, a job that means loss of time as well as trouble. The 500-kw. generator is direct-connected to a large Corliss engine and the 150-kw. generator to a high-speed 240-hp. engine. I should be glad to receive suggestions from POWER readers as to where to look for the trouble.

W. F. LANDMESSER.

Herminie, Penn.

It is reported that a 150,000-volt transmission line is to be used in the Big Creek Development, a high head hydro-electric system which is being constructed in central California.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Some Details of the Cooper Gas Engine

Details, as a rule, play an important part in the successful operation of an engine, yet they are usually overlooked by the casual observer or are afforded but scant mention in technical descriptions.

During a recent visit to the works of the C. & G. Cooper Co., at Mt. Vernon, Ohio, the writer was very much interested in some of the details of their large twin tandem, double-acting, four-stroke-cycle gas engines. These are built in sizes from 200 hp. up, the one inspected being rated at 900 hp.

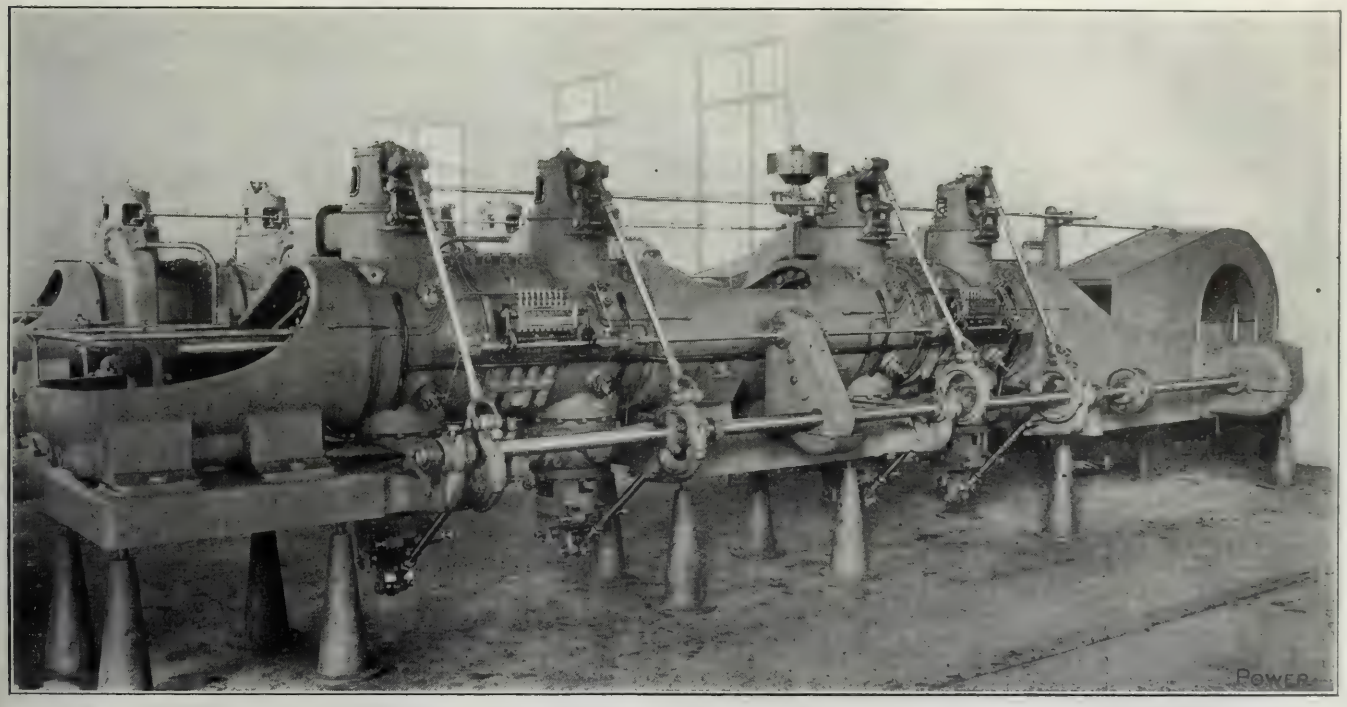


FIG. 1. TWIN TANDEM COOPER GAS ENGINE ON ERECTING FLOOR

Referring to Fig. 1, which shows one of these engines, one is first impressed with the symmetry of design. Considering the engine proper, everything is symmetrical about the middle and everything pertaining to one end of a cylinder is to be found at the other. This not only makes the parts interchangeable, but affords like gas and exhaust passages to the end of each cylinder insuring an equal charge to each. Furthermore, by driving the timing shaft from the middle the torsion effect is equalized.

INLET VALVE-GEAR

Of the various features, perhaps the inlet valve-gear is the most interesting. It is designed with the special object of affording uniform gas velocity and absolute control of the mixture at all loads. Referring to Fig. 2-a, which represents a section through the inlet-valve chamber, A is the valve held against the removable seat B by the spring C, the lower end of which presses against the split spider D; the latter also acts as the lower guide of the valve stem E. The upper part of the valve stem carries the guide nut F which is held at the top by the nut G. The bonnet cover H serves as the upper guide. A loose spring cap I, having a projecting collar is forced against the guide nut F by the spring, although sufficient play is provided between the flanges of this nut and the spring cap to permit the regulating valve to turn freely. A diametrical clearance on the collar avoids binding or sticking should the valve stem be out of alignment with the regulating valve.

One end of the multiplying lever K swings on the pin F' in the guide nut and the other is attached to the link L. A toe M is pivoted at one end to a lug H' on the bonnet cover and the other end is actuated by a pull rod driven through an eccentric from the layshaft. This is shown more clearly in Fig. 1. Hence, the inlet valve A and the regulating valve J to which it is rigidly attached open together against the action of the spring C.

Turning now to the means of regulating the mixture, Fig. 2-b shows a development of the sleeve N. This consists of two parts, an upper N' and a lower N''; the former is provided with fingers cut off at the bottom in steps and these fit into slots of unequal length between corresponding fingers on the lower section. In this way ports p' and p'' are formed for air and gas respectively. The lower part of the sleeve N'' is held fixed by dowel pins and the upper part is made adjustable vertically by a threaded ring Q, which is held in place by the lugs R. When this ring is turned by hand (see scale indicating width of port, Fig. 2-c), the ports p' and p'' can be proportional so as to give the proper mixture.

Surrounding this sleeve N are the air and gas chambers T and S, respectively, which have ports t and s corresponding with the ports in the sleeve.

Inside the sleeve the regulating valve J is provided with a double row of staggered ports j and j', shown in dotted lines at mid-position of the valve, Fig. 2-b. This regulating valve, besides reciprocating with the inlet valve A, also is made to rotate by means of the ring U and the arm W connected to the governor. This is ef-

fected by the ring having two blocks which slide in lugs on the valve; thus the valve *J* is allowed to slide up and down but is maintained in a fixed radial position determined by the governor and the desired amount of port opening is obtained.

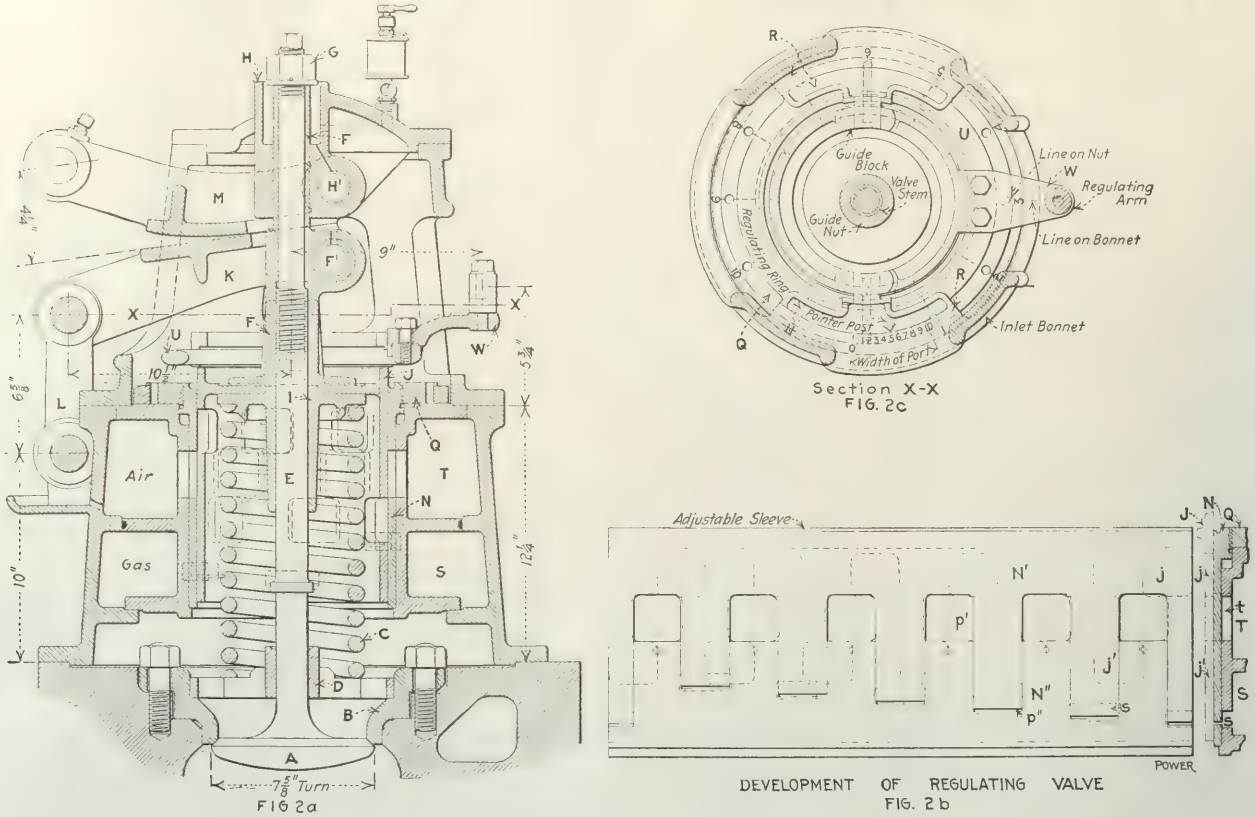


FIG. 2. SHOWING DETAILS OF INLET VALVE

AIR STARTING MECHANISM

As in practically all large gas engines, compressed air is employed for starting, their corresponding cylinders; hence, when the air is turned on all the valves but one automatically close and are opened in the proper order by the cam on the layshaft to admit air to the respective cylinders. In this way the incoming fuel mixture is not disturbed and the mechanism for this purpose being shown in Fig. 3. Mounted on the brackets which support the layshaft, are smaller brackets, each carrying a rod as shown. One end of this rod makes contact with a cam on the layshaft and to the other end actuates the timing valve. These valves are timed so as to open only on commencement of the power stroke in the check valve in each line prevents the starting air from entering the cylinder after the charge has been ignited. After the air supply has been shut off the timing valves and rods fall out of action.

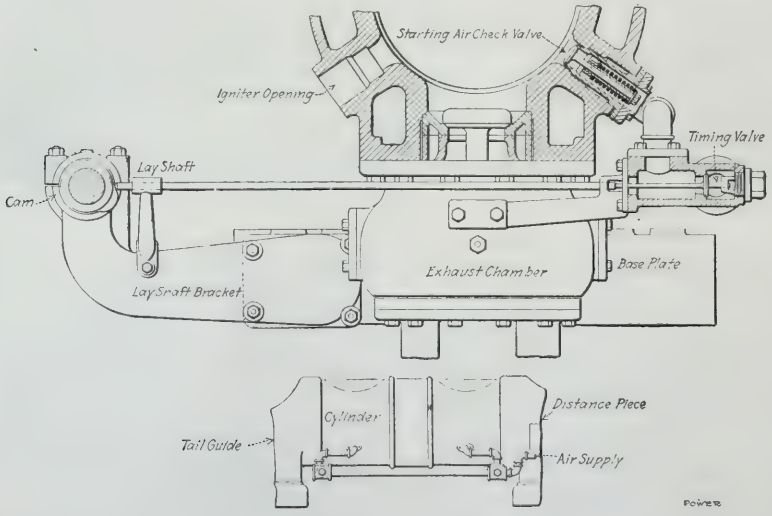


FIG. 3. AIR STARTING DEVICE

COOLING SYSTEM

Another feature worth mentioning in connection with the Cooper engine is the cooling-water system. As in most large gas engines, the pistons and piston rods are water cooled, but to prevent "sweating" of the rods, the water first passes through the cylinder heads and is then admitted to the rods and pistons. Thus it reaches the latter at a higher temperature. The intake into the rods is through a telescopic connection at the intermediate crosshead and the discharge, without reversal of flow, is at the main and tail guide crossheads.

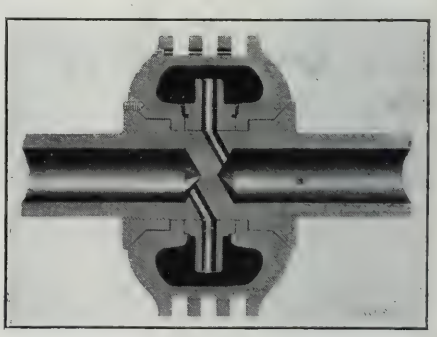
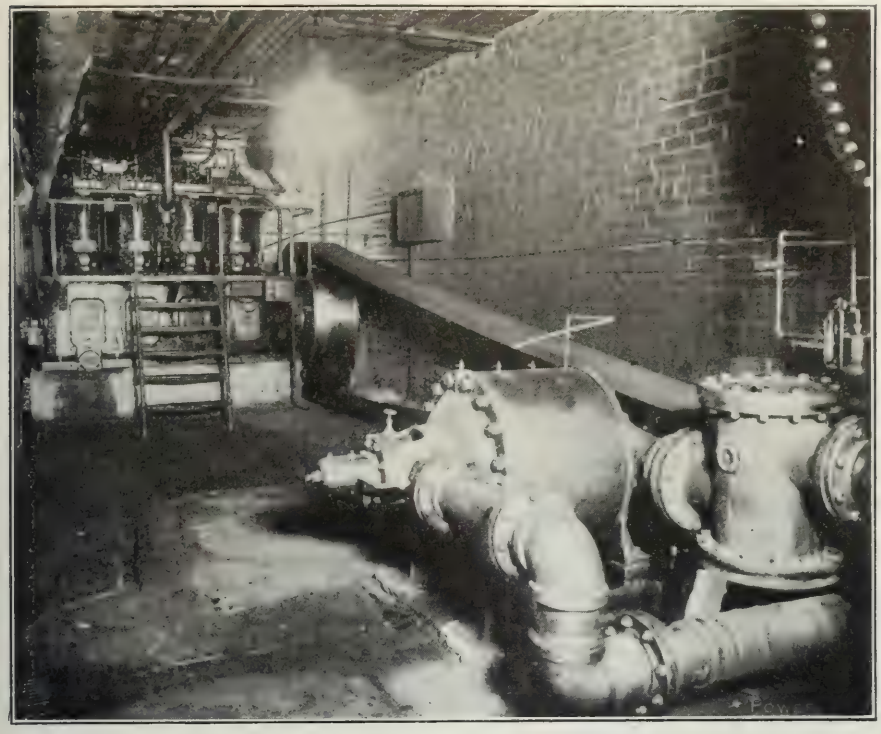


FIG. 4. SECTION THROUGH PISTON AND PISTON RODS

A section through one of the pistons and the rods is shown in Fig. 4, which also gives a clear idea of the method of attaching the pistons. The amount of circulating water required is from 6 to 10 gal. per b.hp.-hr., depending on the load.

Gas Engine for Elevator Service

The First National Bank building in Columbus, Ohio, contains a novel combination of centrifugal pumps driven by gas engines for hydraulic elevator service. The structure is a ten-story office building containing two hydraulic passenger elevators, each having a capacity of 5000 lb. and running at a speed of 500 ft. per minute. These are in almost constant operation during business hours and are in service from 6.30 a.m. to 9.30 p.m.



GAS ENGINE BELTED TO CENTRIFUGAL PUMP

As first installed, the elevators were served by one 9x12-in. triplex double-acting pump, driven by a 75-hp. motor and controlled by a Mason regulating valve which maintained a pressure of 140 lb. Power was purchased from the public-service company at 3c. per kw.-hr. and under these conditions the average cost of operation was $11.50 per day.

In addition to the high cost of operation, there was also serious objection to the noise of the pump gears, and it was decided to replace the electric motor with a gas engine and the triplex-plunger pump by a multi-stage centrifugal pump. The plant as so altered is herewith shown, and has been in constant service for 4 yr. at an average of 15 hr. per day.

The pump is a four-stage type built by the Alberger Condenser Co., and has a capacity of 600 gal. per min. against a head of 333 ft. A heavy multiple check valve eliminates any shock or hammer in closing.

The engine, built by the Bruce-Macbeth Co., is a four-cylinder, 100-hp. machine running at a normal speed of 250 r.p.m., and is controlled through a Mason bypass valve in combination with a special device on the engine-mixing valve. The action is such that when the pressure in the high-pressure tank rises above normal, the fuel and air supply is so reduced as to run the engine very slowly and at a small consumption of fuel. Natural gas is used for fuel.

The first cost of the gas engine and pump equipment was approximately $5000. The total operating cost under present conditions averages $62 per month, whereas with purchased current it was nearly $300 per month.

Preheating for Internal Combustion Engines

In its issue of July 28, *Elektrotechnik und Maschinenbau* publishes a review of an article contributed by A. Nougier to *Die Gasmotorentechnik* on the advantages of preheating the air in internal-combustion engines. The author says that with Diesel and similar engines the air must be compressed to from 425 to 500 lb. to reach a temperature above that of the ignition point of the fuel; in the case of crude petroleum, over 900 deg. F. This high compression is inconvenient as it necessitates increased weight of engine and requires a compressor for the air needed for fuel injection; it also presents further difficulties in maintaining tight pistons and valves. However, in the use of cheap fuels their high ignition temperatures must be reckoned with as these demand higher compression.

The method of securing a higher terminal temperature of compression by the injection of a small quantity of fuel shortly before the end of the stroke, its combustion so augmenting the temperature as to fire the later injected regular charge of fuel, has not proved justifiable, because the pressure still rose to over 560 lb. and the auxiliary fuel often failed to ignite. Also, the cylinder head was complicated by the two inlet valves.

An ample temperature rise can be attained without increasing the compression, by preheating the air drawn into the cylinder during the suction stroke whereby through a small initial heat increment the terminal temperature of compression is raised considerably even with a still lower terminal pressure. This preheating can be done by compelling the exhaust gases to pass through a form of heat reservoir over which the air passes on its way to the engine. Taking air at 60 deg. F. and atmospheric pressure it will be heated by the cylinder walls to say 100 deg. and through compression its temperature will be raised to about 1050 deg. This temperature, however, can be reached with a pressure of only 568 lb. if the air be heated to 176 deg., or with a pressure of 213 lb. by preheating to 227 deg.

In operating with the heavier oils or with tar oil it is necessary at first to run the engine with crude petroleum until the preheater becomes sufficiently warm.

The resultant lowering of the compression due to preheating of the air is of such advantage that the drop in thermal efficiency is negligible. On the other hand, the mechanical efficiency is considerably improved, and the process can at any time be applied to existent engines. It is not to be confounded with preheating of the air for the purpose of gasification, as of benzine, etc., nor with the lowering of the pressure at the end of compression by the injection of water.

Using crude petroleum, with a compression of 10 atmospheres and a terminal temperature of 1040 deg. F. the air would have to be preheated to 300 deg.

Such engines can very easily be built double-acting and working on the two-stroke cycle. The fuel control is simple and accurate and the average turning moment, and thus the duty for a determined speed or the speed for a given load, can be varied within sufficiently wide limits without changing gears. The carbureter and the ignition system disappear and in their place the air-injection arrangement, the fuel pump, and the preheater, the latter displacing the exhaust muffler, are all much less sensitive to derangement than are the parts eliminated.

Preheating, however, must not exceed a certain limit, as combustion would thus be impaired and other disadvantages might arise and it is therefore advisable to provide an air-temperature regulator.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilating Northwestern University Buildings*

By J. M. Stannard

The heating and ventilating of the Northwestern University buildings, at Evanston, Ill., are accomplished by hot-water under forced circulation, and direct and indirect steam plants under both vacuum and gravity methods. The accompanying map, Fig. 1, shows the various buildings with the method of heating in each, the location of the power plant and the size and location of the feed mains.

The boiler plant consists of six boilers, four for the generation of steam and two for the reheating of the hot water. Two are horizontal return-tubular steam boilers 72 in. by 18 ft., and rated at 150 hp. The other two are Stirling water-tube boilers, each rated at 250 hp. The two for hot-water heating are horizontal return-tubular boilers, 66 in. by 16 ft., and rated at 125 hp. The water-tube boilers have Green traveling chain grates and the four fire-tube type are hand fired. During the more moderate cold weather only one of the hot-water boilers is fired for the reheating of the water, and two of the boilers are in use for generating steam. Besides furnishing steam to the radiation, the steam boilers supply circulating and vacuum pumps and one fan engine, for the induced-draft system.

For circulating hot water there are two Marsh single-cylinder double-acting circulating pumps, 10x12x12 in., each having a theoretical capacity of 5.87 gal. per single stroke. These pumps have revolution counters so that an accurate record of the water pumped may be kept. One large closed heater having 396 sq.ft. of heating surface made up of brass tubes, is used in addition to the two hand-fired boilers for reheating the circulation water. This heater was installed to utilize the exhaust steam from the various pumps and stoker and fan engines, so that no exhaust steam is wasted.

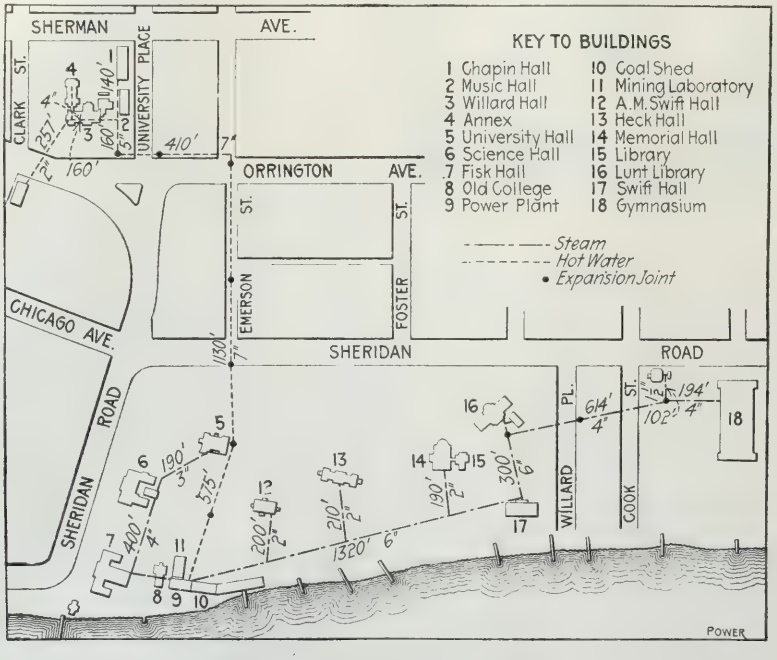


Fig. 1. Map of Buildings, Northwestern University, Evanston, Ill.

The hot-water system has a complete set of recording thermometers which show temperatures of outgoing and return water and outside temperature. In addition, regular thermometers are on the heater and the return line.

The steam-heating system consists principally of direct radiation supplemented with indirect in three of the larger buildings. The indirect radiation is for ventilating purposes, sufficient direct radiation being installed for heating only. Steam is generated at 80 lb. pressure and reduced in the power plant to 40 lb., for distribution to the 12 buildings. One to 5 lb. pressure is carried in the direct radiation and blast coils as weather conditions require, and is maintained by reducing valves in each individual building. Vacuum traps are applied to both direct and indirect radiation in two of the buildings.

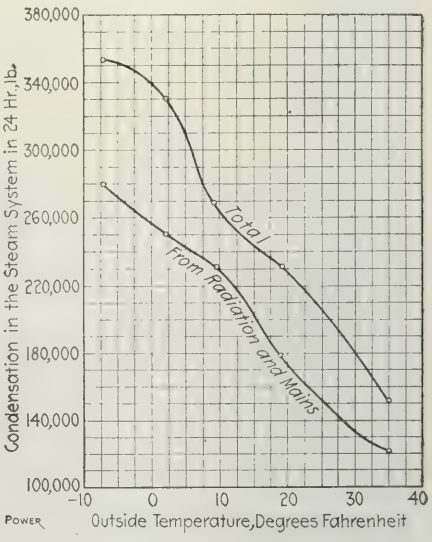


Fig. 2. Condensation in Steam System in 24 Hours

Three Marsh vacuum pumps, one 8x12x12 in., and two each 5x6½x10 in., are installed in the pump room and are used for holding vacuum on the steam-heating return lines and for handling the condensation from this system. The pumps maintain a vacuum averaging about 10 in., which drops to 2 to 3 in. where the returns leave the most distant buildings. The separate return lines are connected into two return mains which lead back to the power plant and are connected through settling chambers to the vacuum pumps in the usual way. The condensation is then delivered by the vacuum pumps to a receiving tank from

TABLE I. HOT-WATER TEMPERATURES FOR DIFFERENT OUTSIDE CONDITIONS

Outside Temp. Deg.	Ordinary Flow Deg.	Ordinary Return Deg.	High Wind Flow Deg.	High Wind Return Deg.
60	115	95	125	100
50	120	100	130	105
40	130	105	140	110
35	135	110	145	115
30	140	115	150	120
25	145	120	155	125
20	152	125	162	130
15	158	130	168	135
10	165	132	175	137
5	173	135	183	140
0	181	140	191	145
−5	190	143	200	148
−10	200	150	210	155
−15	210	155		

*Abstract of paper read before the American Society of Heating and Ventilating Engineers, at Detroit.

TABLE 2. DATA SHEET, HEATING PLANT, NORTHWESTERN UNIVERSITY AT EVANSTON, ILL.
HOT WATER SYSTEM

Date from 10 a.m. to 10 a.m.	Average Outside Temp. Deg.F.	Pump: Rev. of Circulating Pump per 24 hr.	Pump: Lb. of Water Pumped Calc.	Temperatures: Av. Temp. of Flow Deg.F.	Temperatures: Av. Temp. of Reheater Deg.F.	Temperatures: Av. Temp. of Return Deg.F.	Temperatures: Av. Temp. Added to Water by Reheater	Temperatures: Av. Temp. Added to Water by Boilers	Heat: Heat Added by Reheater B.t.u.	Heat: Heat Added by Boilers B.t.u.	Heat: Total Heat Added to Water B.t.u.	Heat: Heat Deliv. to Reheater (Calc. 1000 B.t.u. per lb. cond.)	Heat: Heat Deliv. to Boilers (Calc. 11,000 B.t.u. per lb. coal)	Effi-ciency of Reheater Per cent.	Effi-ciency of Boilers Per cent.	Lb. Condensation from Reheater metered	Coal burned in H.W. Boilers No. 5 Coal at $2.35 per ton Lb	Cost of Coal for Reheater	Cost of Coal for Hot Water Boilers	Total Cost for Heating Hot Water	Cost 1000 sq. ft. Direct Rad. per 24 hr.
Jan. 5-6, 1912	—7	24,965	2,134,807	206	172	155	17	34	36,291,719	72,583,438	108,875,157	37,000,000	112,365,000	98	64.6	37,000	10,215	$8.46	$12.00	$20.46	$1.20
Jan. 8-9, 1912	2	25,306	2,162,549	206	180.4	157.2	23.2	25.6	50,171,136	55,361,254	105,532,390	52,000,000	91,520,000	96.4	60.5	52,000	8,320	$12.05	$9.78	$21.83	$1.27
Jan. 2-3, 1912	9	22,456	1,930,632	180.4	149	137.5	11.5	31.4	22,202,268	60,621,844	82,824,112	23,000,000	97,350,000	96.5	62.5	23,000	8,850	$4.99	$10.40	$15.39	$0.91
Jan. 29-30, 1912	19	21,583	1,857,389	178	151	135	16	27	29,718,224	50,149,503	79,867,727	30,000,000	79,310,000	99	63.2	30,000	7,210	$7.18	$8.46	$15.64	$0.92
Dec. 22-23, 1911	35	19,073	1,649,661	144	126	117	9	18	14,846,949	29,693,898	44,540,847	15,000,000	52,030,000	99	57.1	15,000	4,730	$3.54	$5.56	$9.10	$0.53

which it flows by gravity to a feed-water heater and from there is taken by the boiler-feed pump and returned to the boilers.

The buildings on the forced circulation hot-water system are heated with direct radiation only. Temperatures and volume of circulation are regulated from the power plant according to Table 1.

The supply and return mains on the hot-water system are run in a special conduit, the bottom of which is made of book tile and covered with concrete. The pipes are covered with one thickness of high-pressure covering and a split sewer tile is cemented on the top. Expansion joints are placed at regular intervals. The expansion of water is taken care of by a tank in University Hall. The total hot-water radiation at present is 17,000 sq.ft.

TABLE 3. DATA SHEET, HEATING PLANT, NORTHWESTERN UNIVERSITY AT EVANSTON, ILL.
STEAM SYSTEM

Date from 10 a.m. to 10 a.m.	Average Outside Temp. Deg.F.	Condensation—Lb.: From Radiation Metered	From Hot Water Tank in Gym. Metered	From Reheater Metered	For Heating Swimming Pool Calc.	In Mains Calc.	For Heating Feed Water Calc.	Total Condensation	Coal Burned—Lb.: Heat Added by Boilers B.t.u.	No. 2 Coal at $2.85 per ton Weighed	No. 5 Coal at $2.35 per ton Weighed	Pocahontas Screenings at $3.00 per ton Weighed	Total Coal Burned	Temp. Swimming Pool Raised Deg.F.	Evap. in Boilers per lb Coal Calc.	Ave. Cond. per sq. ft. direct rad. per hour	Cost of Coal for Radiation and Mains	*Cost of Coal for 1000 sq. ft. direct rad. per 24 hr.
Jan. 5-6, 1912	—7	273,300	10,000	37,000	7,500	6,200	20,000	354,000	72,583,438	56,760	....	....	56,760	15	6.23	0.211	$63.93	$1.16
Jan. 8-9, 1912	2	245,600	2,000	52,000	6,000	6,200	18,700	330,500	55,361,254	53,795	....	....	53,795	12	6.14	0.19	$58.43	$1.06
Jan. 2-3, 1912	9	216,200	7,000	23,000	2,400	6,200	15,280	270,080	60,621,844	29,695	....	10,855	40,550	4	6.66	0.17	$48.20	$0.88
Jan. 29-30, 1912	19	172,200	9,000	30,000	3,000	6,200	13,200	233,600	50,149,503	29,250	11,375	....	40,625	6	5.7	0.14	$42.00	$0.76
Dec. 22-23, 1911	35	115,600	7,000	15,000	....	6,200	8,600	152,400	29,693,898	9,410	19,195	....	28,605	....	5.4	0.092	$28.72	$0.52

General Data.—Equivalent direct radiation in steam system, 55,080 sq.ft. Equivalent direct rad. in hot water system, 17,000 sq.ft.
*Includes condensation in mains.

For the steam system the distributing mains are: One 6-in. main for north buildings extending this size to between Swift Engineering Hall and Lunt Library through a 6-ft. concrete tunnel and 4 in. to Patton Gymnasium in a wood conduit. The total load on this north line is equivalent to 36,637 sq.ft. of direct radiation. In addition this line heats a hot-water tank for shower baths and the swimming pool, containing 66,000 gal. A 5-in. line in a brick tunnel carries steam to the south buildings, reducing at Science Hall to 3 in. for University Hall. The total load on this main is equivalent to 18,443 sq.ft. of direct radiation.

Branch lines are taken from these mains to individual buildings. The return lines are run parallel to the feed mains and are 3 in. in size at the power house. One of the interesting features

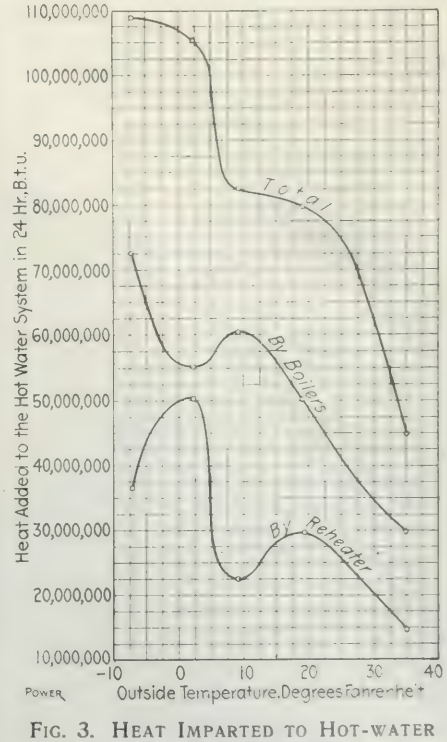


FIG. 3. HEAT IMPARTED TO HOT-WATER SYSTEM IN 24 HOURS

is the fact that the main return line at Patton Gymnasium is approximately 5 ft. below the high part of the tunnel near Swift Engineering Hall. The expansion of both supply and return mains is taken care of by slip expansion joints securely anchored at regular intervals.

Feed mains in the tunnel are covered with two thicknesses of 1 in. each of sponge felt covering. Return mains are covered with one thickness of low-pressure covering. Feed mains in the conduit are covered with one thickness of high-pressure covering, and packed together with uncovered return mains in oily shavings in a special wood boxing. The north supply main is dripped at Swift Engineering Hall, Lunt Library and Patton Gymnasium back into the return line through traps. The south main is sim-

ilarly dripped at Fisk and University halls.

All direct radiation in the buildings heated by steam is under the Johnson system of temperature control, and by-pass dampers in fan-ventilating systems are similarly controlled. For economical operation it was found that this temperature control must be installed.

Ventilation

Fisk Hall, Engineering Building and Patton Gymnasium have a fan system of ventilation, used intermittently as required. Science Hall, University Hall and Lunt Library have natural flue ventilation, and all other buildings have no provision for ventilation.

A system of records pertaining to the heating system is kept by the university. It was decided to select five days, the average temperatures of which would be as close as possible to the following: 10 below zero, zero, 10 above and 20 and 35 above. Readings and calculations cover coal burned, amount of condensation, temperatures, etc., for these days, all arranged on a comparative basis. The results are given in Tables 2 and 3.

All temperatures shown, except those pertaining to the swimming pool, are averages of 24 readings taken hourly between 10 a.m. and 10 a.m. The pump counters, meter and coal readings were taken once every 24 hr. at 10 a.m. Where no meter readings were to be had, as of the steam to the swimming pool, exhaust steam for heating feed water and the condensation in mains, the condensation to be charged to each was calculated as accurately as possible from the information at hand. The columns relating to heat, efficiencies and cost are calculated results based on actual records. In figuring the cost columns no account was taken of anything but coal burned. All curves are based on data taken directly from the figures and are submitted only to show graphically some of the more important items.

Steam vs. Hot Water Heating at Northwestern University

By Ira N. Evans*

The preceding article comparing the operating costs of hot water and steam heating at Northwestern University shows, on careful analysis, some curious facts. The different prices of fuel and boiler efficiencies so confuse the results that they are very misleading, when the economical generation of steam should have nothing to do with the relative costs of the two systems in operation.

The steam plant serves 55,080 sq.ft. of radiation at a maximum distance of 1620 ft. and comparatively little of the radiation at the end of the line. The hot-water plant serves 17,000 sq.ft. of radiation, most of it near the end of the line 2800 ft. from the power house. The maximum steam consumption of the steam system is approximately 14,000 lb. per hr., while that of the water system is about 4600 lb., or a little over one-third.

It is probable that the hot-water system was adopted on account of the extreme distance of the buildings from the power house and to utilize exhaust steam which would be wasted with a live-steam system operating at 40 lb. pressure.

The comparative distances and amount of insulated heating surface in the mains of the respective systems are given in Table 1. In round numbers there are 5500 sq.ft. of surface in the mains of the steam system, including the returns. The hot-water system has a total in mains

TABLE I. HEATING SURFACE IN MAINS

Dia. Main in.	Length Main ft.	Sq. ft. surface Per lineal ft.	Radiating Surface sq.ft.
Steam System			
2	600	0.62	372
3	190	0.92	175
4	1200	1.178	1413
6	1620	1.734	2809
Total	3610		4769
Assume including returns 5500 sq.ft.			
Hot Water System			
2	800	0.62	496
4	320	1.178	377
5	320	1.456	466
7	4230	2.00	8460
Total	5670		9799

of 9800 sq.ft. The steam system is operated at 40 lb. gage pressure (reduced from 80 lb.), or 286 deg. F., and assuming 1.8 B.t.u. per sq.ft. per hr. as the transmission and 40 deg. for the ground temperature the loss in radiation will be

$$(286 - 40) \times 1.8 \times 5500 = 2{,}435{,}400 \text{ } B.t.u. \text{ } per \text{ } hr.$$

The steam on this system is all generated at 80 lb. gage and the returns come back at 10 in. of vacuum, or 192 deg. F., so that 1025 heat units will be available per pound.

Dividing by 1025 will give the pounds of steam

$$\frac{2{,}435{,}400}{1025} = 2376 \text{ } lb. \text{ } per \text{ } hr.$$

If the covering is 89 per cent. efficient the condensation will be

$$2376 \times 0.11 = 261 \text{ } lb. \text{ } per \text{ } hr.$$

which amounts to 6264 lb. per 24 hr. or very close to that calculated in the previous article. The loss in the hot-water mains, calculated in the same way with 180 deg. as the average temperature of the water, would be

$$(180 - 40) \times 1.8 \times 9800 = 2{,}469{,}600 \text{ } B.t.u. \text{ } per \text{ } hr.$$

Dividing by the 1025 B.t.u. to compare the loss with that of the steam system gives

$$\frac{2{,}469{,}600}{1025} = 2410 \text{ } lb. \text{ } per \text{ } hr.$$

and using the same efficiency for the covering the condensation is

$$2410 \times 0.11 = 265 \text{ } lb. \text{ } per \text{ } hr.$$

The condensation is practically the same as for the steam system, but while the loss by radiation on the steam mains is nearly constant throughout the season, that on the water system decreases with the water temperature in milder weather. Upon investigation it may be found that 6 per cent. of the steam used at different outside temperatures will measure very nearly the loss on the water mains. See Tables 5 to 8.

The great drop in temperature, 51 deg., and the low average temperature in the hot-water system tend to make it inefficient and unresponsive to quick regulation at the power house of the average temperature of the radiation. The 7-in. main is much too large and the great volume of water in the mains to be changed in temperature and its low velocity cause a considerable loss as the heat is not radiated when and where needed. The pumpage, 1500 lb., or about 180 gal. per min., gives a velocity of only 1½ ft. per sec. A 4-in. main could be utilized to better advantage, increasing the head to 1.52 ft. per 100 ft. of length or about 40 ft. on the system. The radiation surfaces of the water mains would be reduced by more than one-third and the loss in heat would be less by a like amount to say nothing of the decrease in first cost.

If the pumpage were doubled, decreasing the drop by one-half, the greater velocity in the radiation would mean higher efficiency and would allow lower average water temperatures to be used with the same heating surfaces, or if the same temperatures were maintained less radiation would be necessary to accomplish the same result. Doubling the pumpage would require a 5-in. instead of a 4-in. main. As the water is heated by live steam, this could be done without materially increasing the cost of pumpage over the present arrangement, as the extra power required to circulate the water is largely returned in heat due to friction.

TABLE 2. COST OF STEAM PER 1000 LB.

Cost Coal Per ton	Cost per 100 lb.	Evap. per lb.	Fuel per 1000 lb. Steam	Cost per 1000 lb. steam
$2.85	$0.1425	6.23	160.51	0.22873
2.85	0.1425	6.14	162.86	0.23207
2.89	0.1445	6.66	150.15	0.21697
2.71	0.1355	5.7	175.44	0.23772
2.515	0.1257	5.4	185.18	0.23277

The method of estimating the pumpage is not accurate as the slip and weight of water are not the same for different temperatures, and this in turn tends to

*Consulting engineer, 156 Broadway, New York City.

give erroneous results when determining the total heat in the water.

Inasmuch as the distances and capacities of the two systems compared are so at variance, the loss in the mains should be separated from the comparison of the actual costs of heating. Various prices of coal are used with different evaporations, but as these factors have nothing to do with the relative efficiencies of the two systems, the most economical cost of generating the steam will be determined and used in the computations.

Table 2 gives the cost of steam per 1000 lb. and it is interesting to notice that the most expensive coal is the cheapest, costing $0.21697 per 1000 lb. of steam. Table 3 gives corresponding data on the hot-water boilers and shows that coal at $2.35 per ton gives better results than coal at $2.89 per ton on the steam boilers, the average cost being $0.1782 per 1000 lb. of steam.

The article states that the condensation is returned under 10 in. of vacuum, or 192 deg. F., by the pump. In all probability cold water has to be used for creating this vacuum, in which case some heat is wasted and the water overflowed if all condensation is returned to the plant. This is a charge against the steam system if the returns cannot be used for feed purposes. The steam for heating the feed water should be included in the steam-heating cost as presumably all exhaust steam is utilized and condensation may be returned at nearly 212 deg. F. from the water system.

All exhaust steam utilized in the reheater of the hot-water system will contain 970 B.t.u. instead of 1000 B.t.u. Except for a small loss by radiation from the heater shell, there is no loss as all the heat is either in the condensate or passed through the heater to the system. The efficiencies given for the reheater are probably accounted for in the differences in the actual and assumed values for latent heat.

In Table 5 the cost of the steam in the reheater is corrected so as to apply the price per 1000 lb. obtained with the steam boilers. The heat added by the reheater is given and dividing by 1025 will give the pounds of steam. The British thermal units in the coal used in the water boilers may be disregarded, as the weight and cost of coal is given and the number of heat units may be obtained from the records of the water system. The hot-water boilers show greater economy than the steam boilers, but their rated capacity is 250 hp., and the maximum load is 70 hp. and the minimum 36 hp., so it is not surprising, outside of the excessive estimate of water pumped.

The high-pressure boiler with a live-steam heater and gravity return for the condensation is decidedly more economical from an investment standpoint. The hot-water boiler is idle in summer and the fires are more difficult to handle economically for the wide ranges in outside temperature unless about three times the boiler power necessary is provided in the plant. Using one type of boiler for all purposes makes a more economical arrangement with a minimum of spare boiler power and a maximum overload capacity due to the interchangeability of the boilers. If this plant were properly arranged the entire load of both heating systems could be carried on the steam boilers and the hot-water boilers eliminated or used in the steam-power plant as spare boilers. If the same generosity in boiler power were shown for the generation of steam as evidently has been installed for the water system the lower priced coal might be used in the steam boilers with a more economical showing.

Tables 4 and 5 give the costs per 1000 sq.ft. of radiation with and without the mains for both the steam and water systems, using the lowest cost given in Table 2 for generating steam. The data shows that the two systems are about on a par when the mains are omitted as the cost per 1000 sq.ft. of radiation on the steam system averages $0.8656 against $0.87 for the water. With the mains the cost on the steam system is $0.89 against $0.925 for the water, a difference of less than 4 per cent. This is as it should be with automatic heat control on the steam

TABLE 3. COST FOR WATER BOILERS ON SAME BASIS

B.t.u. from Water Boilers per 24 hr.	B.t.u. per hour	Lb. of Steam (1025 B.t.u. per lb.)	Coal per 24 hr.	Coal per hr.	Equiv. Evap.	Coal per 1000 lb. Steam	Cost @ $2.35 per ton
72,583,438	3,024,310	2951	10215	426	6.92	144.3	$0.1695
55,361,254	2,306,719	2250	8320	347	6.48	154.2	0.1812
60,621,844	2,525,910	2464	8850	369	6.68	149.7	0.1759
50,149,503	2,089,563	2039	7210	300	6.79	147.1	0.1728
29,693,898	1,237,246	1207	4730	197	6.13	163.2	0.1918
						Average	0.1782

TABLE 4. COST OF STEAM HEATING

Outside Temp. deg.F.	Lb. steam per 24 hr. feed water and radiation	Cost at $0.21697 per 1000 lb.	Cost per 1000 sq.ft. per 24 hr. without mains	Cost per 1000 sq.ft. per 24 hr. with mains
—7	293,300	$63.64	$1.1554	$1.1798
2	264,300	57.35	1.0412	1.0656
9	231,480	50.22	0.9117	0.9361
19	185,400	40.23	0.7304	0.7548
35	124,200	26.95	0.4893	0.5137
		Average	$0.8656	$0.8900

Mains 6200 lb. at $0.21697 per 1000 lb. = $1.34 per day = $0.0244 per 1000 sq.ft. of radiation.

TABLE 5 COST OF HOT-WATER HEATING

Outside Temp. deg.F.	B.t.u. from Reheater	Lb. steam (1025 B. t.u. per lb.)	Cost at $0.21697 per 1000 lb.	Cost Coal for H.W. Boilers at $2.35 per ton	Total Cost Heating with mains	Total Cost without mains (6% deducted)	Cost per 1000 sq.ft. with mains	Cost per 1000 sq.ft. without mains
—7	36,291,719	35,406	$7.68	$12.00	$19.68	$18.50	$1.158	$1.09
2	50,171,136	48,947	10.62	9.78	20.40	19.18	1.200	1.13
9	22,202,268	21,661	4.70	10.40	15.10	14.19	0.888	0.835
19	29,718,224	28,993	6.29	8.47	14.76	13.87	0.868	0.816
35	14,846,949	14,485	3.14	5.56	8.70	8.18	0.512	0.481
						Average	$0.925	$0.870

TABLE 6. TRANSMISSION ON STEAM SYSTEM

Outside Temp. deg.F.	Total Steam lb. per 24 hr.	B.t.u. per sq.ft. per 24 hr.	B.t.u. per sq.ft per hr.	Average Temp. Steam-Room	B.t.u. per sq.ft. per hr. per deg. dif.	
					With Mains	Without Mains
—7	299,500	5574	232	215–70	1.60	1.57
2	270,500	5034	210	212–70	1.48	1.45
9	237,680	4423	184	212–70	1.30	1.24
19	191,600	3566	148	212–70	1.04	1.006
35	130,400	2427	101	212–70	0.71	0.677

TABLE 7. TRANSMISSION ON WATER SYSTEM

Outside Temp. Deg. F.	Total B.t.u. added to water per 24 hr.	B.t.u. per hour	B.t.u. per hr. without Mains (94%)	B.t.u. per sq.ft. per hr.		Average Temp. Water-Room	B.t.u. per sq.ft. per hr. per deg.dif.	
				With Mains	Without Mains		With Mains	Without Mains
—7	108,875,157	4,536,464	4,264,276	267	251	180.5–70	2.41	2.27
2	105,532,390	4,397,183	4,133,352	259	243	180.6–70	2.34	2.20
9	82,824,112	3,451,005	3,243,945	203	191	158.9–70	2.28	2.13
19	79,867,727	3,327,822	3,128,153	196	184	156.5–70	2.27	2.13
35	44,540,847	1,855,868	1,744,516	109	103	130.5–70	1.80	1.70

system, but the water system without heat control will be the cheaper to install. If the operating cost of the hot-water system were divided by the capacity in square feet of radiation which the 7-in. main could economically supply, or about 50,000 sq.ft. instead of 17,000, the cost would be considerably less than that of steam.

After all these costs have been deduced the water system uses more heat in proportion to the radiation than the steam system if the total radiation is given correctly in both cases.

In Tables 6 and 7 the total heat for both systems is given and the transmission rate per square foot of surface per degree per hour is determined under the temperature conditions given in the article. The data shows that the hot-water mains would have to be bare and added to the heating surface to transmit the amount of heat given at the rate for the steam system. At —7 deg. outside temperature all heating surface on the steam system should be in use and a rate of 1.6 B.t.u. only is shown, which is about right.

The water system with an average temperature of the medium about 30 deg. lower requires a rate of 2.2 to 2.4 B.t.u. to transmit the heat generated, which is impossible for the amount of surface given. If there are hot-water radiators that will give this transmission, it would be greatly to the credit of the hot-water system. These high values form a startling contrast to the efficiency of the steam radiators. The highest efficiency the writer has ever obtained is 1.8 B.t.u. with an average water temperature of 200 deg. and about 15 deg. drop on the job. Where the pumpage was furnishing one-half the heat in very moderate weather there was a seeming transmission of 2 B.t.u. per sq.ft., but this was due to the drop being made up by the friction of the water.

No doubt in moderate weather the low transmissions on the steam system are due to the heat control shutting off radiation, and this shows in the steam consumption. The steam job works out about right, but apparently something is wrong with the hot-water data; either the pumpage has been miscalculated or there is more heating surface. The article states that there is the equivalent of 36,888 or practically 37,000 sq.ft. of radiation on the steam system. The condensation is taken for the heating only and divided by 55 to obtain the cost per 1000 sq.ft. Is it possible that the capacity of the hot-water tanks and swimming pool is added to the system as radiation and only the steam consumption for the heating taken? If the cost of steam for the system were divided by 37 instead of 55 it would mean an increase per 1000 sq.ft. over the quantity given of 50 per cent., and would make the cost of steam heating relative to hot water much in favor of the latter.

Temperature Equivalents of Wind Velocities*

By W. H. Whitten

A comparison of records taken at the group of buildings of the Harvard Medical School on the total heat expended and average temperatures and average wind velocities showed that 1 mile of wind movement per hour required substantially the same amount of heat supply as 1 deg. change in temperature. A further study of similar records, however, has shown that there is a greater proportion of loss due to wind movement as the temperature drops.

This led the author to make investigations as to the impact effect of wind of the same velocity at different temperatures. It was found that there is a regular rate of increase in effective pressure as the temperature drops, although the wind velocity remains constant. This regular rate of increase of pressure is maintained only while the barometer readings are normal. A barometrical change caused changes in the impact pressure. The exact rate of this change was not determined, but the fact of such change was detected, the tendency being for an increase in pressure as the barometer rose and for a decrease as it fell.

It was estimated that, with the barometer and the wind constant, the increase in pressure is 0.4 per cent. per degree drop in temperature. It was also found that the nonpressure or suction on leeward sides of buildings increased in about the same proportion. The point at which heat loss from 1 mile of wind movement per hour and temperature were equal seemed to be between 36 and 39 deg. above zero. Above this temperature, the effect of wind became less important than the temperature changes, and below it, correspondingly more important.

For example: If, at 37 deg. plus, 1 mile of wind movement per hour is equal to 1 deg. drop in temperature, at zero 1 mile of wind movement per hour will equal

$$1 + (37 \times 0.004) = 1.148 \text{ deg.}$$

If the temperature increases to 50 deg. plus, then 1 mile of wind movement equals

$$1 - (13 \times 0.004) = 0.948 \text{ deg.}$$

As a rule for personal guidance, the author has adopted the following: From 40 to 15 deg. plus, 1 mile of wind movement per hour is equal to 1 deg. drop in temperature; from 15 deg. plus to 20 deg. minus, 1 mile of wind movement per hour is equal to 1.15 deg. drop in temperature. This is for buildings constructed in the ordinary manner, that is, without protected windows. Applied strictly to the glass surface, with leakage standardized, the loss from wind movement may be calculated as only three-sevenths of the loss under usual and ordinary conditions. This not only applies to the sides having the so-called greatest exposure, but, owing to the suction or nonpressure existing on the sheltered sides, should be applied to all sides of the building.

*Abstract of paper read before the American Society of Heating and Ventilating Engineers, at Detroit.

Location of Ventilating Fans

The improper placing of circulating fans often lowers the efficiency. Forcing or drawing the air around short turns means a waste of power or less air movement for the same amount of power applied to the fan. Fig. 1 shows an exhaust fan placed against the wall at *E*

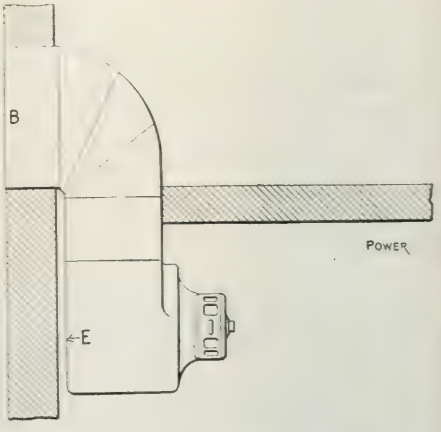


Fig. 1. Fan below Outlet

for removing hot air through the opening *B*. Originally the flue had a square corner, but with the long ell shown in the drawing the fan works more efficiently.

A better location for the fan is shown in Fig. 2 at *C*; the propeller is of the inclosed type directly connected to the motor. Air is pulled up through the opening *X* and forced out at *B*. The friction is reduced and the air is taken from

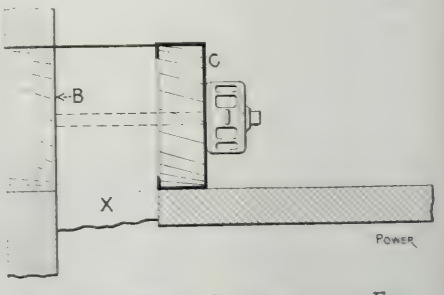


Fig. 2. Better Location for Fan

the hottest part of the room. The propeller could be set at the point *B* in Fig. 2 with the shaft carried back to the motor, as indicated by the dotted lines and better results obtained than with the installation illustrated in Fig. 1.

C. R. McGahey.

Baltimore, Md.

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POWER

Preparing for License Examinations

It is natural for the novice in steam engineering to be imbued with ambition and it is usual for him to take the shortest route to advancement. It is the license he is after with very little thought of anything farther. The advice often given the novice by those who have secured a license is sometimes misleading and frequently responsible for failure. Some advise the study of questions and answers in catechism form; others will give him questions, which the examiner of the district is in the habit of asking, and the answers which are thought to be satisfactory. Either of these "short cut" methods, while helpful in a way, is likely to be productive of disappointment, for the examiner as a rule will readily discern that the answers to his questions are "stocked," and a little divergence from his usual manner of questioning may cause confusion to the applicant.

In these columns a correspondent recently suggested changing around from plant to plant so that familiarity with various types of equipment might be had as an aid to securing a license. This is well enough as a part of the young man's training, but familiarity with the use of steam-plant equipment seldom teaches the principles of its action. It is this knowledge that the student of steam engineering should endeavor to possess, and it can be attained only by systematic and persistent study of the fundamental principles and the natural laws which affect the generation of power and its transmission.

Presuming that the beginner has a rudimentary education in common school studies, the logical procedure is to take up a course of study, beginning with physics, principles of mechanics, machine elements, and the kindred subjects to which his studies will progressively lead. The mere memorization of facts and theories, however valuable when properly used, should by no means be the main object of this study. There should be a constant aim to clearly understand what is read, to develop the faculties of observation and reasoning, and to rightly use what is learned. Only by using and adding to it can the student make it his own.

When sufficiently advanced in these studies, and with a reasonable experience in actual steam-plant work, the applicant for license will be able to go before the examiner and discuss the subjects on which the examination questions are based, giving answers that show an intelligent understanding of the principles involved, which invariably will impress the examiner favorably. The value of a thorough educational foundation does not end in the examination room. It will be found of far more value in the practical operation of the plant. For the problems which must be solved in actual work are readily mastered when the basic laws are understood.

The foregoing is neither new nor original, but is the basis of education as taught in colleges and technical schools, and is especially directed to those who, without preparation, have secured employment at the trade, and find it difficult to advance because they study at random and with uncertain result, when the same effort properly directed would lead to success.

Getting the News

Gratefully we acknowledge our appreciation of the interest shown by our readers in sending us newspaper clippings of boiler explosions, wrecks, flywheel accidents and other matters pertinent to POWER's columns.

But— (and this "but" looms large in this instance) early newspaper reports of such happenings are notoriously inaccurate, and necessarily so when one considers that they must be published a few hours after they occur. It is manifestly unfair to lay much blame upon the average reporter for his inaccuracies as his lack of technical training and mechanical unfitness excuse his not detailing the causes of these accidents.

Whenever it is possible, we ask that our readers who are located near the scene of the occurrence, and their occupation allows it, will endeavor to ascertain all the facts, the cause, how it might have been avoided, or at least try to verify the daily-press report. If you cannot get this news, ask someone who can. A photograph often helps to make the story clear.

As stated in our foreword in the issue of Mar. 12, "Don't let anything stop you. Be the *first* to send the news!" Remember, there are over thirty-one thousand readers in the POWER family, and we ask that each one make extra efforts to reflect additional credit on the family. We will bear all the necessary expense attendant upon securing all possible information of interest to our readers, and we will pay you the usual space rates besides.

Incomplete Data

A statement often seen in engineering articles is to the effect that the subject mentioned will necessitate more space than the author cares to give or thinks it deserves.

Unfortunately this frequently renders the article of much less value than if the author had covered a single subject thoroughly, rather than attempt to cover several subjects in the same space.

For instance, a book recently published contains less than ninety pages, yet the subjects it embraces include boilers, steam engines, turbines, and oil and other types of internal-combustion engines. Any one of these subjects deserves several volumes to adequately treat of it. As a result the volume is practically valueless to the purchaser, and is a disappointment in every way.

The idea is this. Suppose one is writing an article dealing with furnace combustion. A dozen pages or more may be devoted to giving directions for stopping up cracks in the setting, how to fire properly and control the draft, but if the author stops there the real meat of the article is missing. Engineers want to know the why of things; why air leaking into the furnace will adversely affect the economical combustion of the fuel; why it is not just as practicable to fire heavy as light, and what the draft has to do with the process of combustion.

Telling the reader that these conditions will produce a low CO_2 and then dropping that phase of the question, because the space available is insufficient, is disappointing. If the reader requires information as to how to handle his fire, he will also require information regarding flue gas and why faulty conditions affect a high percentage of CO_2.

It is one thing to know such and such is a fact, but it is more important to know why it is so. Stating facts by halves is much like eating partly cooked food, most unsatisfactory. Both should be given longer treatment and done to a turn.

Discussion vs. Comment

Our Sherlock Holmes thinks he has detected a disposition on the part of certain contributors to take the slightest pretext that any article affords to send us some comment to print. In his dyspeptic moments, he has gone so far as to declare that some evidently search the paper each week for something on which to hang an excuse for writing a letter, which, as a rule, simply reiterates what has already been said, or calls attention to the most obvious conclusions that anyone could reach from a mere reading of the original matter.

We sincerely hope that he is wrong in this supposition.

Discussion is wanted—not mere comment, and perhaps it may not be amiss for us to make a little explanation of just what we are after.

POWER aims to help the reader in every possible way and wants all the good practical information obtainable from those constantly meeting perplexing problems and solving them in ways that would interest others.

Not for a moment would we discourage anyone from calling attention to anything that may be wrong as described by another or that might be done in a better way. Such are particularly desirable. Whenever an article or a letter recalls to a reader an experience that throws a new light on a subject, we do indeed want to hear of it, but rambling comments that deal only in generalities and get nowhere with conclusions are no use.

Give the matter a little thought and you will not go wrong. When you write us, write not merely for the sake of writing, but because you have something really good to offer. In short, use POWER as it is meant to be used.

Being Contented

Now and again some well intentioned correspondent opens his tear-ducts and floods our editorial sanctum with a stream of grievances against his employer or chief. The more trivial the cause, the more inclined is he to hope that "some brother engineer" will take up the cudgels and provoke a long and useless discussion. Perhaps we should acknowledge our own culpability in that we have permitted too free discussion of grievances. If so, we have erred in our leniency.

Next to war and pestilence, the individual most to be shunned is he who is continually complaining. He dampens the ardor of his associates; he works unwillingly and halfheartedly and, sooner or later, he is let go as inefficient or otherwise undesirable.

A powerful factor in the betterment of almost any man's condition is contentment—at the least, patience. But as a celebrated divine has aptly said, when a man is perfectly satisfied with himself it is time for him to die. It is self-dissatisfaction that a man should cultivate; he will then have a clear insight in placing blame where it properly belongs. In a few words, contentment is born of self-dissatisfaction.

"If you do not like your boss, get a new one" is usually much easier said than done, but a rigid self-examination frequently reveals the fact that it is not always the other fellow in whom all the faults lie. The self-satisfied individual in his warped judgment makes mountains out of molehills, and scents a reproach in every order and a criticism in every look. The wise man, dissatisfied with himself, finds his real satisfaction in self-study.

Let us first to ourselves be true. If we discover that the mote is lodged in our own eye, our mental vision will not then distort the actions of those about us, and instead of grumbling and whining over adverse conditions, we will seek to ameliorate them. No one loves a "kicker." Let us cultivate contentment as a most valued possession. Then the daily task will be less difficult, we will more patiently bear with others, and the compensation will be satisfaction in our well-doing.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Cemented and Sewed Belt Joint

Much has been written pro and con on the subject of belt splices, but I have a very good splice that has not been mentioned and will describe it for the benefit of others. It takes more time to make the splice, but I consider it better than either wire lace or cement alone and it will last as long as the belt. The ends are scarfed and cemented with glue in the usual manner and then sewed with ¼-in. rawhide lace leather, as shown in Fig. 1. To do this a two-edged knife is used. It is seen from Fig. 1, that the holes for the stitches are cut at an angle to the side of the belt. The seams are placed about 2 in. apart and the lace is pulled in tight, the stitches being about

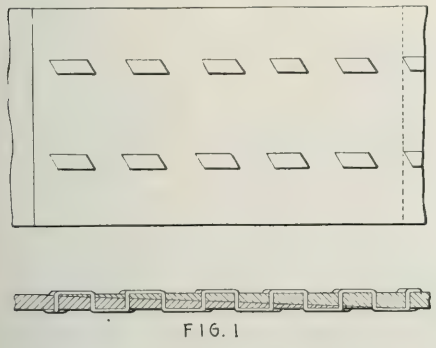


FIG. 1

CEMENTED AND SEWED BELT JOINT

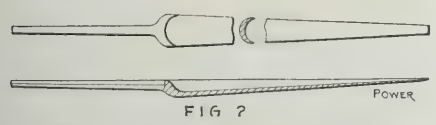


FIG. 2

TOOL FOR PUNCHING LACE HOLES

½ in. long. This makes a smooth, pliable joint, and one that cannot pull apart.

Of the numerous belt fasteners on the market nearly all make a stiff joint and will break the belt at the end of the fastener, as the belt is cut nearly one-third off in driving the fastener through, especially when used on small high-speed pulleys.

In making a splice with rawhide lace I begin in the center of the belt and work to the ends of the joint and back to the center. To make the lace holes I had a cutter made as in Fig. 2. This is forged from a piece of steel, like a half-round file hollowed out and filed to a sharp cutting edge. It is easy to push the sharp point through the belt and give it a half turn. If a larger hole is required, the cutter is pushed down and turned until the hole is the right size. It is better to bore from both sides because in a thick belt the hole will be tapered if bored from one side only.

In making a cement splice, I use a block plane to scarf the ends and common liquid glue as cement. I have a 2½-in. single-ply leather belt running a brine pump in a wet place. This belt has a 4-in. splice made with liquid glue, not sewed, that has been in constant use nearly one year without the splice starting to open. There is also a glued-and-sewed governor belt that has been in use over four years without any damage to the splice.

The mechanic that informed me about the sewed splice said that he had made several such in 36-in. belts, which gave good results and in which the rows of stitches were 1 in. from the edge and 2½ in. apart.

J. C. HAWKINS.

Hyattsville, Md.

Diagrams from Water End of Large Pump

The diagrams shown were taken from the low-pressure, intermediate- and high-pressure pump chambers of a vertical triple-expansion, pumping engine, 28 and 54 and 80 by 60 in., with 30-in. plungers and of 20,000,000 gal. capacity. The terms low-pressure, intermediate- and high-pressure are used to designate the pump chambers under the low-pressure, intermediate- and high-pressure cylinders, respectively.

The inlet and discharge pressures, as will be noticed, are practically the same on all three pumps. The pump receives water from a 48-in. supply pipe leading from a reservoir. The level of water in the reservoir is maintained at a height sufficient for water to flow into and fill the pump, as will be noticed by the atmospheric line, which runs parallel with the inlet line of the diagram, except for a slightly higher pressure in Fig. 1, from the low-pressure pump. This is due to the fact that the pump is set so that the low-pressure pump is nearest the supply pipe.

It will be interesting to follow the lines of the diagrams in the direction of the arrows and note the variation of pressure and the causes of the irregular lines. Starting at *A*, on the low-pressure diagram, Fig. 1, it will be noticed that as the plunger descends, the pressure rises considerably higher than that in the mains. This is due to the inertia of the valves and the resistance of the outlet caused by a heavy spring, intended to assist the valve in seating quickly.

The diagrams were taken when the pump was running at a discharge rate of 22,000,000 gal. per 24 hr., which gives a plunger speed of 270 ft. per min. With a 60-lb. spring it is found that the pressure raised to 115 lb., at point *B*. The pressure in the discharge line being 85 lb.

It will be observed that after valves

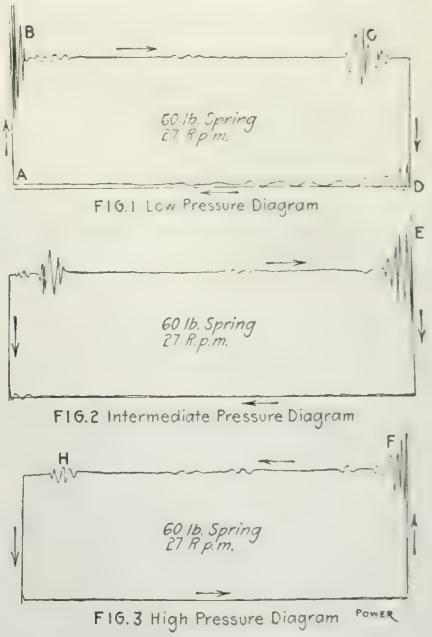


FIG. 1 Low Pressure Diagram

FIG. 2 Intermediate Pressure Diagram

FIG. 3 High Pressure Diagram

DIAGRAMS FROM WATER END OF LARGE PUMP

are fully opened the plunger displaces its volume without any variation in pressure until *C* is reached. The fluctuation at this point occurs when the plunger of the low-pressure pump has almost reached the end of its stroke and, so far as its displacement is concerned, there is no reason why there should be any variation until the plunger had reached the end of its stroke and started to rise.

By carefully watching the action of the pump we find that the intermediate plunger starts descending at this time, and the irregular line at *C*, Fig. 1, and at *E*, Fig. 2, are caused simultaneously, for when the low-pressure plunger is nearing the end of its stroke, the intermediate plunger starts descending, thereby adding volume and pressure to the discharge line. Fig. 3 is from the high-pressure pump chambers, the plunger commencing its downward stroke at *F*

and discharge takes place without any variation until the point *H* is reached, when the displacement of the low-pressure plunger causes a slight irregularity in the pressure line. As the low-pressure pump is a considerable distance from the high pressure, the effect is not so noticeable.

A singular thing is that when the plunger starts descending, and as the valves are forced slightly open, evidently at the instant when the pressure in the pump chamber is overcoming the pressure in the discharge line, water in the latter momentarily flows back to the pump chamber. I believe the irregular lines at the commencement of the plunger stroke prove this.

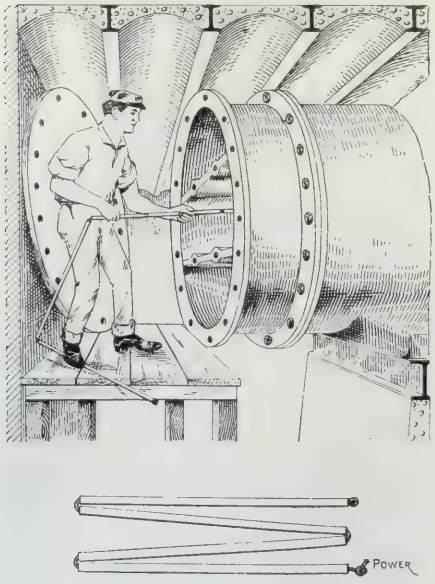
JOHN BURNES.

Columbus, Ohio.

Condenser Tube Cleaning Kink

A kink useful to many engineers in small plants where space is limited, is herewith shown.

The tubes on the intake side of the condenser which we have in service became plugged with small stones, twigs and mud, and how best to clean them puzzled us for a time. The head was taken off, and at first we tried to wash the tubes out, but found this of no avail. As we had but 4 ft. between the condenser and the wall, using a straight rod was out of the question, so we hit on the plan shown.



APPLICATION OF JOINTED CONDENSER TUBE CLEANER

A length of ¼-in. pipe was cut into 3-ft. lengths, a couple of strands from a steel-wire rope were passed through and made fast at one end by soldering, and a knot tied in the other end, leaving it a couple of inches longer than the pipe to allow for bending. This made a good form of jointed rod, easily handled in the close quarters, and good time was made in cleaning out all the tubes.

ARTHUR H. HODGES.

Hudson, N. Y.

Sand Submerged Suction Disables Condenser

While serving as troubleman for a large company, I was sent to a town to locate and remedy the trouble which a water-works concern was having with an independent jet condenser which my company had furnished. I was met at the station by the owner of the plant, a judge of two counties, by the way, and escorted post haste, over to the pumping station where the disabled condenser was located. I obtained a brief outline of the situation, while looking matters over, as to the behavior of the condenser and tried operating it myself; its action corroborated the engineer's remarks.

I concluded that the trouble lay in the suction pipe, so after a description of the layout of this line from the suction well, which was covered with about 10 ft. of snow, to the condenser, I took off the condensing cone and discovered a large stone about as large as a medium-sized cantaloupe lodged in the neck. This immediately aroused my suspicions and I made further inquiries about the suction well, being assured by owner and engineer, emphatically, that it was clean and clear and that the suction pipe was an ample distance above the bottom. I arranged to have one of the laborers clear the snow away from the top of the well while I made a still further examination of the interior of the condenser, finding everything in good condition.

I then went down to the well, which was uncovered by this time, and with the aid of a long pole on the end of which was a hook, discovered the end of the suction pipe to be submerged in sand to the extent of about 6 in. This sand the river had deposited during high-water periods.

The judge was with me when I made this discovery and it required no jury to help him form his opinion, as he immediately sentenced one of his men, the one who had been most positive that the well was clear, to put on hip boots and clear out the sand. After this was done, I started the condenser and raised and maintained the required vacuum without any trouble.

W. OLIVER.

Racine, Wis.

Suitable Form of Packing List

This copy of our packing list, and the manner of its arrangement, may prove useful to engineers in plants where there are many machines which require many sizes of packing.

In making up a list of packing and the machines in which it is used, it is usually best to arrange it as shown. The name and kind of packing should head the list of machines in which it is used. Then the names of the various machines and the location of their stuffing-boxes should follow. Opposite each stuffing-box item should be stated the size packing it requires.

PACKING LIST

Plant of . . . . . .

"A's" Steam Ring Packing

Engine No. 2
- Piston rod h.p.-cylinder . . . 1 1/4 x 2 1/4 in.
- Piston rod l.p.-cylinder . . . 2 1/2 x 4 in.
- Valve rod h.p.-cylinder . . . 3/4 x 1 1/2 in.
- Valve rod l.p.-cylinder . . . 1 1/8 x 1 7/8 in.

Engine No. 3
- Piston rod . . . 1 3/8 x 2 1/4 in.
- Valve rod . . . 3/4 x 1 3/8 in.

Engine No. 4
- Piston rod . . . 1 3/4 x 2 7/8 in.
- Valve rod . . . 7/8 x 1 1/2 in.

Fan engine
- Piston rod . . . 1 5/16 x 1 7/16 in.
- Valve rod . . . 5/8 x 15/16 in.

Fire pump
- Piston rod (both ends) . . . 1 3/4 x 2 3/4 in.
- Valve rod . . . 3/4 x 1 1/2 in.

House pump
- Piston rod (both ends) . . . 1 3/16 x 1 7/8 in.

High pressure water pump
- Piston rod . . . 2 1/2 x 3 1/2 in.

Boiler feed pump
- Piston rod (steam end) . . . 1 1/4 x 2 in.
- Valve rod . . . 1/2 x 1 in.

Drinking water pump
- Piston rod (both ends) . . . 7/8 x 1 3/8 in.
- Valve rod . . . 7/16 x 11/16 in.

No. 1 Vacuum pump
- Piston rod (both ends) . . . 1 1/2 x 2 1/4 in.
- Valve rod . . . 5/8 x 1 1/16 in.

No. 2 Vacuum pump
- Piston rod (both ends) . . . 7/8 x 1 3/8 in.

Air compressor
- Piston rod (both ends) . . . 1 1/16 x 1 1/4 in.

Brine pump
- Piston rod (both ends) . . . 1 1/2 x 2 3/4 in.
- Valve rod . . . 3/4 x 1 1/2 in.

Ammonia pump
- Piston rod (steam end) . . . 1 3/8 x 2 1/4 in.
- Valve rod . . . 3/4 x 15/16 in.

"B's" metallic packing

Engine No. 4
- Piston rod . . . 1 3/4 x 2 7/8 in.

"C's" metallic packing

Engine No. 1
- Piston rod . . . 3 x 4 5/8 in.
- Valve rod . . . 1 11/16 x 2 9/16 in.
- Expansion rings for above packing . . . 3 x 4 13/16 in.

Hemp packing
- Ammonia pump (ammonia end) . . . 5/8 in.
- Boiler feed pump (water end) . . . 3/4 in.

"A's" Ammonia packing-spiral

Ammonia pump
- Piston rod (ammonia end) . . . 1 3/8 x 2 5/8 in.

Boiler feed pump
- Piston rod (water end) . . . 4 x 5 in.

- "B's" spiral . . . 1/4 in. 3/16 in. 1/8 in. 1/16 in.
- "A's" spiral . . . 5/8 in. 1/2 in. 3/4 in.
- "A's" spiral-ammonia . . . 9/16 in. 3/32 in. 1/4 in. 3/32 in. 7/16 in. 5/32 in.

The list should be framed behind glass and hung in a place convenient to where the packing is kept. Another way is to paste the list on a board of the proper size and cover thoroughly with shellac, but the glass-paneled frame is preferable.

The date of receipt of each size and style of packing should be kept, also the date and amount used in each machine. This enables one to determine how much packing any particular machine uses in a given time, and also reveals the fitness of the packing for the work required of it.

It pays to keep packing in separated compartments, each labeled, so that a glance at the label will indicate the size of packing contained in the place or places in which it is used.

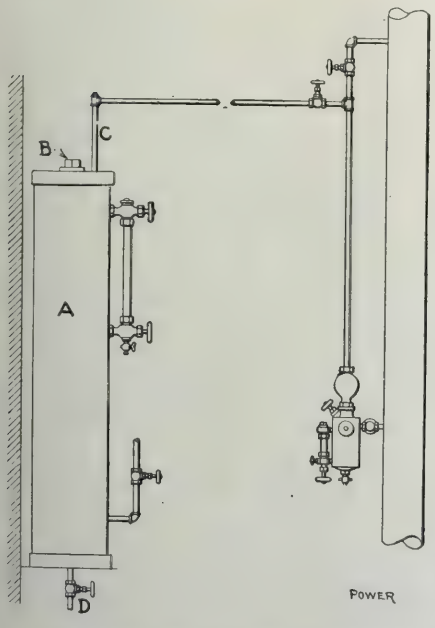
H. G. GIBSON.

Washington, D. C.

Filling Outfit for Lubricators

In a plant having many lubricators to be filled at comparatively short intervals the arrangement shown saves much time and trouble.

Alongside the pump or engine is located the oil reservoir *A*, made of a piece of 10-in. pipe of any length and fitted with caps at both ends. The top cap should be drilled and tapped to receive a plug *B* which can be removed for filling the tank. The oil outlet *C* to the lubricator should be ½- or ¼-in. pipe and connected to the top cap. The use of a glass gage is optional. A drain *D* should be provided for emptying the condensate.



OIL RESERVOIR FOR FILLING LUBRICATORS

If several lubricators are close together the filling outfit should be centrally located and the oil outlet piped to each lubricator. The manner of operation is evident from the illustration.

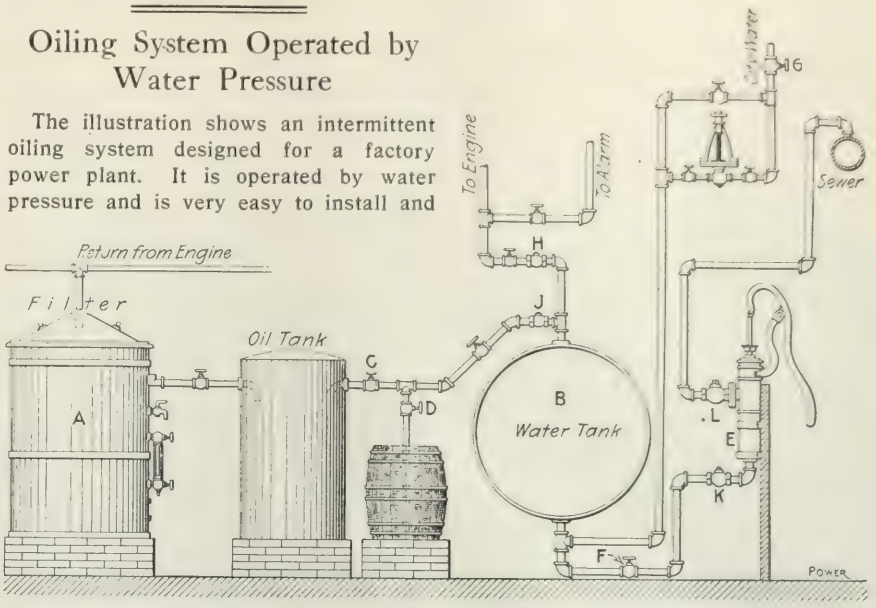
FRANK MCGOWAN.

Newark, N. J.

[The outfit might be further improved by connecting an oil line under gravity or forced feed to the filling hole of the reservoir.—EDITOR.]

Oiling System Operated by Water Pressure

The illustration shows an intermittent oiling system designed for a factory power plant. It is operated by water pressure and is very easy to install and use where it is possible to locate the filter *A* below the engines.



ARRANGEMENT OF OIL SYSTEM OPERATED BY WATER PRESSURE

With the storage tank *B* filled with water, and the water supply shut off, the valve *C* is opened to the receiving tank, or the valve *D* in the pipe shown between the two tanks, to a barrel of oil. The water is then pumped out of the storage tank by the hand pump *E* to the sewer and the oil flows into the storage tank as the water is pumped out. Next the valve *F* to the pump and the valve *C* to the receiving tank are closed and the valve *G* to the city water supply and pressure regulator is opened. The valve *H* is then opened to the engines and the oil, after passing through the bearings, flows by gravity to the filter, and from the filter to the receiving tank.

The engines are shut down at 5:30 or 6 p.m. and after all the oil has run from the filter to the receiving tank, the night engineer pumps the water out and the oil into the storage tank and the system is ready for operation next morning.

The receiving tank must be large enough to hold all the oil used during the day. Check valves *J*, *K* and *L* are fitted in the inlet and outlet pipes and also to the sewer pipe so that the oil cannot flow the wrong way or any be lost. If the water is turned on before the valve to the pump is closed, the water runs to the sewer and no harm can be done beyond the loss of the water. A gage-glass on the storage tank shows the amount of water and oil. Pressure and alarm gages and an alarm bell are located in the engine room. The pressure regulator is set to maintain a pressure of about 8 lb. more than is necessary to raise the oil to the engines, and the alarm gage is set so that the bell rings when the pressure falls to within 2 lb. of the pressure that will just raise the oil to the engines.

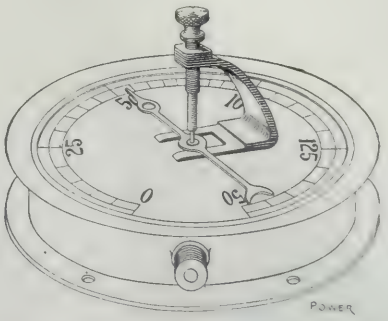
This system has been in constant service for over five years and has given no trouble.

JOHN HARPER.

Brooklyn, N. Y.

Removing Hands from Dial Gages

Removing the hands of most dial gages is often difficult. Unless great care is taken the tedious operation invariably results in bending the hand or defacing the dial and often bending the flanged front of the casing. To facilitate the work I use the device illustrated. It may be easily made of steel or brass. The illustration clearly shows its application to remove the hand.



GAGE HAND REMOVING DEVICE APPLIED

J. K. VOGLER.

Ellwood City, Penn.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Louisiana Eight-hour Law Criticized

In the Aug. 6 issue a news item covers the recent enactment of a law by the Louisiana legislature governing the hours of labor for engineers in that state. Your source of information was not trustworthy, because this law affects stationary firemen only, and but very few of them. This law seems to me to be the rankest kind of class legislation, and is in the opinion of high legal authority entirely unconstitutional.

The law as enacted applies to all businesses operating their power plants continuously night and day, except the petroleum industry, sawmills, cotton gins and presses, and sugar mills. Those familiar with the industries of Louisiana will perceive that practically only one business is affected by the law, i.e., public-service corporations. Anything not included in the above list of exceptions can probably without great inconvenience, shut down for at least one hour daily and so not violate the law.

If it is dangerous or undesirable, or even distasteful to the firemen to stand watches of 12 hours each in a plant that operates 24 hours daily, and if it requires an act of the legislature to rectify the "abuse," why does it still remain perfectly legal, to make one man stand a watch of 13 or 14 hours in a plant operating that long daily, but shut down the rest of the time? Furthermore, why not give the firemen employed in the excepted industries an opportunity to benefit by the law, also?

To understand my position in the matter one must realize that I have for three years been chief engineer for one of the corporations that "get it in the neck." I know of no fire room in Louisiana where white firemen are employed, except where oil is burned for fuel. I have paid my negro firemen, for hand firing coal, $2, $1.85, $1.75 and $1.50 daily for 12 hours, depending on the watch and the man. These wages and these hours may seem unreasonable to many in the North, but my plant has been considered a desirable one to work in because of the wages paid. Negro labor in the South is poor at best. The negroes are usually mentally deficient, irresponsible and they get "tired" and quit on a moment's notice.

Any man employing them has his hands full without any complications introduced by a paternal legislature. The law now says that I cannot work my negro firemen more than eight hours daily. Naturally, I figure that I cannot afford to pay them as much as formerly, particularly on the light watches, where about the only necessary qualification is the ability to remain awake, a natural difficulty with these firemen. But they will not "stand" for a wage reduction and instead, go to some plant where they can earn as much even if they have to work several hours more, for it is the high wages and not long hours they consider.

About the only effect of the law, as I see it, has been to shift a portion of the firemen's work and troubles onto the already overburdened back of the chief engineer, whose hours of labor, whose worry and responsibility are great enough without this new law to aggravate the situation.

JOSEPH POPE.

Meridian, Miss.

Substitute Piston Rings

L. M. Johnston's article under this heading in the Sept. 3 issue recalled an experience of mine. I was employed at a large hotel and, like most hotels, this one had a laundry in connection with it, which was busy every week day and until noon on Sundays. The laundry machinery was driven from the same engine that drove a 75-kw. generator, which was a 16x42-in. Corliss engine running at 100 r.p.m.

One Sunday after the engine had been acting badly, I removed the head and follower-plate and found one ring in seven pieces and about 3 in. gone from the other. We had no extra rings and as the laundry had to run on Monday, I substituted rings of ⅞-in. hydraulic packing.

When the engine was started on Monday morning plenty of oil was used in the cylinder and everything went along all right. I had expected to have a new set of rings made on Monday, and put in immediately, but the manager decided to put in a motor to run the laundry and have the engine overhauled. By the time the motor was installed, seven weeks had passed.

When the cylinder was opened, the packing was found in as good condition as when put in; it had worn smooth on the edge in contact with the cylinder.

ARTHUR V. LACY.

Julietta, Ind.

Mr. Johnston's article recalls my experience with substitute piston rings. Few engineers are acquainted with the old forms of pump packing, one of which is shown in Fig. 1. This was known as an improved hemp packing.

The piston had two broad packing rings made to the exact bore of the cylinder and had a follower-plate against them on each side which drew up the cones *AA* and set out the hemp filling, which expanded the rings. The piston nut could be tightened and a pin put through the rod to keep it from working off. Some of the heads had three or four screws to hold on these follower-plates. This kind of packing was much used formerly, and considered a big improvement over the old hemp packing which just filled the space between the flanges as it was easier to put in.

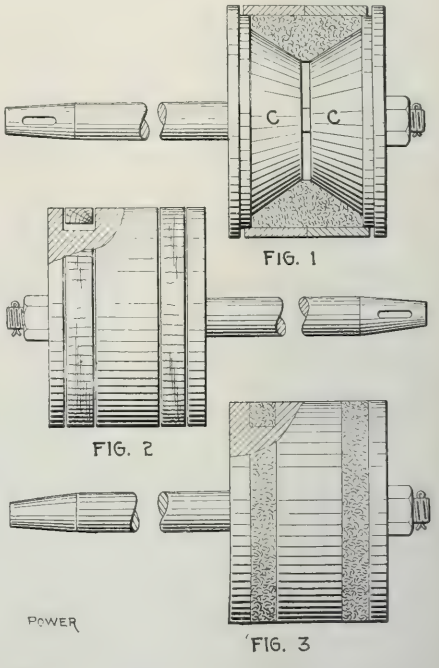


FIG. 1

FIG. 2

FIG. 3

PISTON RINGS OF HYDRAULIC PACKING AND HICKORY WOOD

Fig. 2 shows another form of substitute packing, which has worked successfully for several weeks on small engines. The ring was made of a hoop of hard-wood which filled the ring groove without binding against the sides. I have seen such rings made from hickory wood that gave long and successful service.

I have corrected piston troubles by using a packing such as shown in Fig. 3. The rings were made of hemp rope

driven in the grooves, but they will not last long unless the piston head fits the cylinder bore closely. On steam pumps having broad steam-packing rings badly worn and leaking, I have filled in the grooves with hemp packing, and the pump consumed much less steam, as this hemp filling stopped the free passage of steam past the piston head. By putting sheet springs beneath badly worn snap packing rings they will be made steam-tight and run for several months without leakage.

C. R. McGAHEY.

Baltimore, Md.

Losses in Uncovered Steam Pipes

The recent letter under this heading, by J. L. Kezer, will inspire others to effect similar economies in conveying steam or hot water. That pipe covering and other insulation bring a much larger return for every dollar expended thereon than many other things in the plant is not sufficiently appreciated. Every opportunity for preventing loss of heat should be studied and taken advantage of if painstaking calculations prove that the expenditure is warranted. In Mr. Kezer's case the loss due to uncovered pipes amounts to $3160.90 per year.

Mr. Kezer will not be able to reduce the radiation loss to quite 90 per cent., as expected, for the steam pipe cannot be insulated with sufficient effectiveness. But the economic advantage in utilizing covering continuously is actually in excess of 200 per cent. per year as shown by the following calculations:

Assuming for simplicity, that all the steam pipe to be covered is of 2½ in. inside diameter, then the 3626.1 sq.ft. of radiating surface represents

$$3626.1 \times 1.329 = 4819 \textit{ lineal feet}$$

In testing nine kinds of commercial steam-pipe covering in 1901, Prof. D. S. Jacobus, of Stevens Institute, found that asbestos air-cell covering, saved 74.4 per cent. of the heat lost from bare pipe, which means in Mr. Kezer's case, a saving of steam worth

$$\$3160.90 \times 0.744 = \$2351.71 \textit{ per year}$$

Using asbestos-sponge felted covering the saving was found to be 84.9 per cent., which represents a saving of

$$\$3160.90 \times 0.849 = \$2683.60 \textit{ per year}$$

From these estimated savings remain to be deducted the first cost and depreciation charge. The air-cell covering, 1 in. thick, applied, may cost $500, and the asbestos-sponge felted covering $1000, not counting cost of covering fittings and flanges. Adding 10 per cent. for depreciation, the total cost of the air-cell covering during 10 years will be

$$500 + [10 \times (500 \times 0.10)] = \$1000$$

making in 10 years a net saving of

$$\$23,517 - 1000 = \$22,517$$

or a saving of 453.4 per cent. each year on the original investment of $500.

Using asbestos-sponge felted covering the cost in 10 years equals

$$\$1000 + [10 \times (1000 \times 0.10)] = 2000$$

making a net saving of

$$\$26,836 - \$2000 = \$24,836$$

or a saving of 248.4 per cent. each year on the original investment of $1000.

From the comparative results it does not follow that the use of high-priced covering is never justified because a highly efficient covering will further reduce the amount of condensation in a steam pipe feeding an engine and thus contribute to safety in operation.

A factor which tends to diminish the above difference in advantage between the two kinds of covering is found in the insulation required for valves, fittings and flanges. If, to be conservative, the radiating surface of these parts is assumed to equal 25 per cent. of the surface of the pipe, or 900 sq.ft., the cost of covering with either kind at 25c. per sq.ft. would be $225. This covering, owing to the use of cement, is credited with an efficiency of only 50 per cent. and its saving of heat in 10 years will amount to

$$900 \times 0.50 \times \frac{\$31,609}{3626.1 \textit{ sq.ft.}} = \$3924$$

Added to $23,517 the total saving in 10 years is $27,441. The total cost is

$$\$450 + 1000 = \$1450$$

leaving a gain of $25,991, which represents, with air-cell covering, a profit saving each year of 358 per cent. on the original investment of $725.

Now adding the same saving, $3924, to $26,836, the saving with asbestos-sponge felted-pipe covering, brings the total in 10 years up to $30,760. Deducting the total, $450 + 2000 = 2450 from this saving, leaves a net gain in 10 years of $28,310. The saving during each of the 10 years of life of the covering is therefore

$$\frac{\$2833 \times 100}{\$225 + 1000} = 231 \textit{ per cent.}$$

on the original investment of $1225.

These propositions are certainly attractive, but it is important to keep in mind, that the fewer the number of hours the covering is made use of, the smaller becomes the return from the investment, and *vice versa*.

Mr. Kezer's method of comparing the saving in radiated heat with equivalent tons of coal as delivered to the trestle is unusual. Since the efficiency of boilers will always be less than 100 per cent., more heat than that radiated from the pipes must be generated in the furnace. If the boiler efficiency is 70 per cent., the actual fuel consumption with uncovered pipes will be

$$\frac{2,719,575 \textit{ B.t.u.}}{10,817 \textit{ B.t.u.} \times 0.70} = 359 \textit{ lb. coal per hour.}$$

or 4.61 short tons per day, which at $1.65 = $7.61 per day, × 365 = $2777.65 per year, or 53.7 per cent. more than by Mr. Kezer's method.

Since by taking the cost of steam at the rate of 11.63c. per 1000 lb., the above sum is increased only from $2777.65 to $3160.90 or by 13.8 per cent. This steam cost evidently includes no fixed charges. In other words, the full cost of the steam is probably 15 per cent. greater than the net cost of 11.63c., and the saving capacity of the pipe covering is therefore greater than the calculations show.

CHARLES H. HERTER.

New York City.

Mr. Spencer's Belt Troubles

Mr. Spencer does not give the size of the pulleys in his letter on belt trouble in the July 9 issue, but I venture to say that the diameters of the driving and driven pulleys on which the rider gives trouble differ considerably while they are nearly the same on the other set of which he speaks.

Assume the engine-wheel diameter as 12 ft. and the driven-pulley diameter 4 ft. Neglecting creep and slipping the driven pulley makes $\frac{12}{4} = 3$ revolutions for one of the engine wheel. For each turn of this wheel the rim will travel $12 \times 3.1416 = 37.6$ ft. and the rim travel of the driven pulley will be the same.

The three-ply rider belt used by Mr. Spencer must be about ⅝ in. thick, which will cause it to act as a lagging on both pulleys, increasing their diameters

$$2 \times \tfrac{5}{8} = 1\tfrac{1}{4} \textit{ in.}$$

The rim travel with this increased diameter for one turn of the engine wheel will be

$$\frac{(12 \times 12) + 1.25}{12} \times 3.1416 = 38.0259 \textit{ ft.}$$

But the circumference of the driven pulley will be increased to

$$\frac{(4 \times 12) + 1.25}{12} \times 3.1416 = 12.893 \textit{ ft.}$$

and to make three turns for one of the driver it travels

$$12.893 \times 3 = 38.6793 \textit{ ft.}$$

during one revolution of the engine wheel. This amount is 0.6534 ft. more than the travel of the rider belt (without slipping) on the engine wheel, the rider therefore is drawn on by the driven pulley with a slip of 0.6534 + ft. per revolution of the engine wheel or one-third that quantity for every revolution of the driven pulley. Mr. Spencer will get better results if he leaves the rider off.

A friend informed me that he once riveted a rider to the under belt and after a run was surprised to find the rivets had pulled out. If Mr. Spencer will make a mark on the two belts which run without trouble, he will find after a run that the marks are not in the same relative position.

I recently saw a horizontal belt drive

(driving pulley 28 in., driven 84 in.) the 24-in. belt of which slips on the large pulley. The only reason I could see for this was that the small pulley had a very high crown, in fact, I have never seen another pulley with so high a crown. The belt has taken the shape of this crown so that it has troughed. The bottom of this trough does not touch the large pulley, but is carried around on a cushion of air and for this reason the slip comes on the large pulley, which is iron, the small one being paper.

L. A. Fitts...

Leominster, Mass.

The condition which Mr. Spencer mentions is due to two causes: first, that the original belt slips or creeps on the driven pulley, even with the rider on, and, second, that the old belt stretches more than the rider. In placing the rider belt, he probably drew it up very tight, it therefore stretches but little when loaded. There is little or no slip between the belts; hence the rider, in addition to increasing the friction on the pulleys, prevents the old belt from slipping as much as it did formerly. At first it would seem that less work is done with the rider on, because the effective pull of the original belt is lessened by the amount of tension in the new one. However, the rider increases the pressure along a 10-in. strip between the pulley and the big belt, with the result that the slip is not as great. Hence more work is done because, with the same belt speed, the driven wheel turns faster than before.

About the only thing that can be done with the drive as it stands, is to thoroughly clean the old belt with a mixture of equal parts of naphtha and benzine or some other grease-dissolving agent. Then, when the leather is clean, give it a thin coating of good belt dressing, applied hot and frequently in small amounts until the belt is soft and pliable. It will be as good as ever and will transmit as much power as possible for the given speed, width and tension. I had a similar experience, several years ago, but with a smaller belt. The difficulty was overcome in the manner described.

W. G. Hawley.

Clarks Summit, Penn.

I am not in favor of such a drive as Mr. Spencer is using when both pulleys are not of the same diameter. If he will figure out the speed of the shaft from the base pulleys, then add the thickness of the 30-in. belt to both pulley diameters he will perceive that the 10-in. belt tends to drive the shaft somewhat slower; therefore deriving its power from the driven instead of the driving pulley; hence the slack and tight sides of the belt in the position he states. The crowning of the pulleys will also make a difference in a drive like this one. If the driven pulley is the more crowned, it will aggravate the trouble; if the driving pulley is the more crowned, it may counteract the effect; therefore making it appear to be a satisfactory drive.

James W. Wilde.

Pawtucket, R. I.

Centrifugal Pump Troubles

E. D. Gargnier's letter in the Sept. 3 issue under the above heading is good, but I would ask him what formula he would recommend to others who do not use a certain formula every day.

His formula

$$F = \frac{WRN^2}{35{,}196}$$

is all right if used frequently, but one who uses the formula for the centrifugal force only occasionally would spend more time hunting up this special one than by using the fundamental formula, and in this case the formula

$$F = Ma$$

will prove to be the shortest.

Since

$$M = \frac{W}{g}$$

and the acceleration a of the body toward the center is equal to $\frac{v^2}{R}$ it is evident that

$$Ma = \frac{Wv^2}{gR}$$

This is the formula which F. Webster used in the June 25 issue and which is easier remembered than the odd denominator in the special formula.

Franz Szabo.

Kewanee, Ill.

Phosphorus for Flue Gas Analysis

Allow me to call attention to Mr. Terman's articles on flue-gas analysis relative to the use of phosphorus as a reagent for determining oxygen. The phosphorus should be of the white stick form, and a pipette when filled with it looks very much like the pipette illustrated on page 330 of the Sept. 3 issue, the sticks of phosphorus occupying the positions shown for the glass tubes.

My reasons for preferring stick phosphorus to potassium pyrogallate are that the absorption of oxygen is quicker, that there is never the slightest doubt when a fresh supply of the reagent is needed, and that the pipette needs new reagent less often. One pound of the stick phosphorus is enough for several thousand analyses.

When the gas to be tested is admitted to the pipette, the sticks of phosphorus will begin to smoke. This will usually last only a minute or two and the reaction is over, and the gas should be run back into the measuring burette. The difference in volume is oxygen. A little time should be given to allow it to come back to the room temperature before measuring. No shaking of the pipette is required.

In handling the stick phosphorus precautions are necessary. If possible, handle only with pincers; if the fingers must be used, see that no particles adhere to them; give the phosphorus no chance to get under the fingernails, and keep the unused reagent under water. I have made several hundred analyses with both phosphorus and potassium pyrogallate, and I would not use the latter when the former was obtainable. White stick phosphorus must be used; the red will not do, nor will any other form than the sticks.

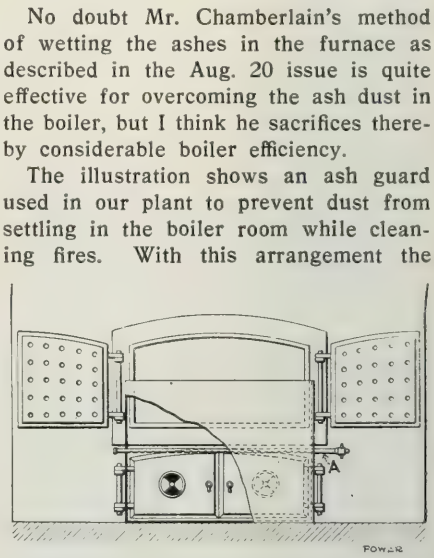
Donald M. Liddell.

Elizabeth, N. J.

Device for Wetting Ashes

No doubt Mr. Chamberlain's method of wetting the ashes in the furnace as described in the Aug. 20 issue is quite effective for overcoming the ash dust in the boiler, but I think he sacrifices thereby considerable boiler efficiency.

The illustration shows an ash guard used in our plant to prevent dust from settling in the boiler room while cleaning fires. With this arrangement the



Sheet Iron Box Containing Water Pipe Used in Wetting Ashes

ashes are moistened at the time they are drawn from the furnace, thereby preventing the furnace from cooling as much as it would by Mr. Chamberlain's method.

The guard is a sheet-iron box shaped to the width of the furnace doorway. The front of the box is open so that as the ashes are pulled out they fall into the box and between it and the ashpit doors. A perforated water pipe A is run through the box so that the ashes may be wet as they are drawn out. The top of the box should be enough below the top of the furnace door to allow using the cleaning tools. If the box is made too deep, trouble will also be experienced in manipulating the tools when pulling ashes from the dead-plate. It should also set flush with the front of the boiler.

A. R. Kummerer.

Hazelton, Penn.

Air Compressor Efficiencies

By E. M. Ivens*

Even when accompanied by a statement of terminal pressure and inlet temperature and pressure volumetric efficiency is nothing more than a guarantee of clearance volume.

Overall efficiency gives something definite and is a more satisfactory guide.

*Ingersoll-Rand Co., New Orleans, La.

Extending use of compressed air has brought into prominence such terms as "mechanical efficiency," "compression efficiency," "volumetric efficiency," and surprisingly few engineers actually know their real meaning.

"Volumetric efficiency" has become a high sounding catch word with compressor salesmen and manufacturers, and a high guarantee inserted in the specifications is regarded by many as a good and sufficient assurance of superior economy of the compressor. As a matter of fact, so many variables and limitations are embodied that a mere statement of the percentage of efficiency is not only misleading, but absolutely worthless when made without qualifications.

The simple formulas, derivations and discussions hereinafter given may be followed easily by keeping in mind the fundamental laws of expansion and compression of sensibly perfect gases.

Mechanical Efficiency

The energy in the steam admitted to the steam cylinder of an air compressor is expended:

1. To compress the air.
2. To heat the air during compression.

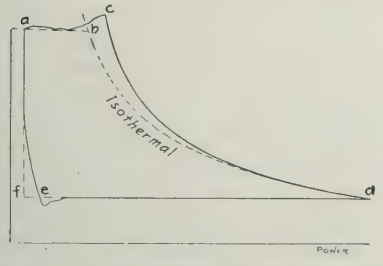


Fig. 1. Determining Compression Efficiency

3. To heat the jacket water.
4. To overcome the friction of the machine.

The horsepower required by 1 and 2 may be computed from the air-indicator diagram; 3 may be found by observing the temperatures of entering and leaving jacket water together with its weight and computing the heat units therefrom and the equivalent horsepower; and 4 is found by subtracting the sum of the first three from the indicated horsepower in the steam cylinder.

The mechanical efficiency of a steam-driven air compressor, then, is equal to the air horsepower plus the jacket horsepower divided by the indicated horsepower, or

$$E_m = \frac{A.hp. + jkt.hp.}{I.hp.} \qquad (1)$$

and the mechanical efficiency of a power-driven machine is expressed by

$$E_m = \frac{A.hp. + jkt.hp.}{B.hp.\ delivered\ to\ compressor\ shaft} \qquad (2)$$

This efficiency depends on the mechanical construction of the machine and the lubrication. It will be found to vary from 75 in poorly designed machines up to 92 per cent. in the best designs.

Compression Efficiency

Compression (or compressor) efficiency is the ratio of the theoretical horsepower required to compress an amount of air, to that actually required, or

$$E_c = \frac{Theoretical\ hp.}{A.hp.} \qquad (3)$$

The adiabatic and isothermal horsepowers are both theoretical, but since the term efficiency expresses how nearly perfect a machine or appliance is, the latter is proper to use (referring to no clearance base) in the formulas. This efficiency depends upon the design of the water jacket and cooling appliances, and it is principally to increase compression efficiency that multistage compression is employed.

To determine the compression efficiency, the isothermal curve is plotted on the air card, starting at the beginning of the stroke, and ending at the theoretical delivery line, or terminal pressure line. The area (Fig. 1) *abdfa* thus inclosed, divided by the area *acdea* of the actual card is the compression efficiency.

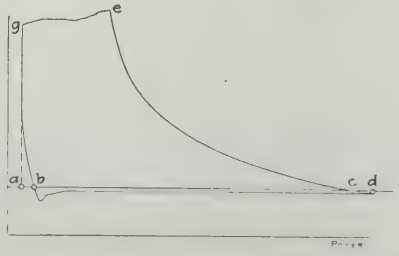


Fig. 2. Typical Compressor Diagram

Indicator cards that show a very high compression efficiency are open to suspicion as investigation will invariably show that either the suction valves leak, or air is escaping from the compression side of the moving piston to the suction side, due to a scored cylinder or leaky piston. Actual compression curves will follow the adiabatic quite closely as the water jacket has little effect other than to facilitate lubrication.

Volumetric Efficiency

Volumetric efficiency is the ratio of the actual cubic feet of free air compressed per unit of time to the cubic feet of piston displacement during that time, or

$$E_v = \frac{Actual\ cu.ft.\ free\ air\ per\ minute}{Cu.ft.\ piston\ displacement\ per\ minute} \qquad (4)$$

On the indicator diagram, Fig. 2, the observed volumetric efficiency is *ad* ÷ *bd*.

Volumetric efficiency depends, first, on the clearance volume in the air cylinder: With no clearance between cylinder heads and piston at the end of the stroke, and no lost space in and around the valves, the volumetric efficiency (referring to atmospheric air) would always be 100 per cent. The greater the clearance volume, the greater will be the volume of the cylinder occupied by the expanded clearance air. This is self-evident.

Volumetric Efficiency depends, second, on the terminal pressures: The higher the terminal pressure of air in any given cylinder, the greater will be the volume occupied by the expanded air of the clearance spaces. This means that as the terminal pressure is increased, the volumetric efficiency decreases. To show this graphically, three superimposed diagrams are reproduced in Fig. 3, taken from the same cylinder at the different pressures shown. The increase in volumetric efficiency as the terminal pressure decreases is plainly evident.

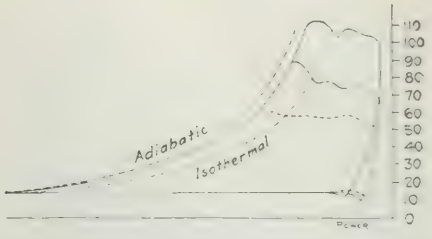


Fig. 3. Volumetric Efficiency Depends on Terminal Pressure

Volumetric efficiency depends, third, on the temperature and pressure of the intake air: Since, by definition, volumetric efficiency refers to free air, or air at 14.7 lb. pressure and 60 deg. F., every change of temperature and pressure of intake air has its effect directly on the volumetric efficiency. For instance, assume a room temperature of 60 deg. F. and atmospheric pressure at 14.7 lb., or conditions where actual free air is available and drawn without heating into the cylinder. In the room is a compressor whose capacity is 135 cu.ft. of this air per minute, whose piston displacement is 150 cu.ft. per min.; and the terminal pres-

sure 100 lb. The volumetric efficiency under these conditions is

$$E_v = \frac{135}{150} = 90 \text{ per cent.} \qquad (5)$$

Now, suppose that from some cause, the temperature of intake air were raised from 60 to 65 deg. F., the atmospheric and terminal pressures remaining as before. The compressor will still draw in 135 cu.ft. of air per minute, but owing to the higher temperature, a lesser weight or mass of air will be withdrawn from

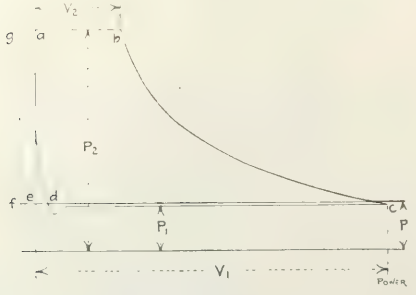


FIG. 4. VOLUMETRIC EFFICIENCY FROM INDICATOR DIAGRAM

the atmosphere. According to the law of Charles, 135 cu.ft. of air at 65 deg. F. and 14.7 lb. pressure is equivalent to 133.71 cu.ft. of air at 60 deg. F. and 14.7 lb. pressure. Under these conditions the expression for volumetric efficiency is

$$E_v = \frac{133.71}{150} = 89 \text{ per cent.} \qquad (6)$$

This shows that for every 5 deg. F. rise in intake air temperature, a loss of practically 1 per cent. in volumetric efficiency occurs.

Likewise, volumetric efficiency is affected by change of atmosphere or intake pressure—the temperature remaining constant. To show this, suppose the compressor removed to a higher altitude where intake air of 13.7 lb. pressure and 60 deg. F. is available. Now, 135 cu.ft. of this air is equivalent to 125.81 cu.ft. of free air and the third expression for volumetric efficiency is

$$E_v = \frac{125.81}{150} = 83.87 \text{ per cent.} \qquad (7)$$

Therefore, for every 0.163 lb. decrease of intake pressure, a loss of 1 per cent. in volumetric efficiency occurs.

FORMULAS AND MEASUREMENT

The most popular and convenient method of determining the volumetric efficiency of an air cylinder is from the indicator diagram. Referring to Fig. 4, the ratio $\frac{dc}{ce}$ might be called the observed volumetric efficiency. On the actual diagram, these distances are measured with some convenient scale, and the computations made and the results so obtained corrected for inlet temperatures and pressure. Volumetric efficiency found after these corrections are made is the true or real efficiency.

Expressions for real and observed volumetric efficiencies may be derived from the diagram as follows: Remembering that

$$PV^n = P_1V_1^n = \text{constant}$$

from Fig. 4 is obtained

$$\frac{df}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} \qquad (8)$$

$$df = de + ag$$

Substituting in (8)

$$\frac{de + ag}{ag} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

whence

$$de = ag\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (9)$$

As ag is the clearance volume and may be determined by actual measurement by other approved methods, it is known, and designating it as V_c and substituting in (9), gives

$$de = V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (10)$$

Now

$$\text{Observed } E_v = \frac{dc}{ce} = \frac{ce + de}{ce} \qquad (11)$$

where ce = piston displacement = P_d

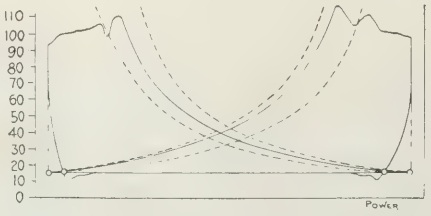


FIG. 5. EFFECT OF LEAKY VALVES

Substituting de as given by (10) in (11) gives

$$\text{Observed } E_v = \frac{P_d - V_c\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]}{P_d}$$

$$= 1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right] \qquad (12)$$

Obviously the ratio $\frac{V_c}{P_d}$ is the percentage of the cylinder volume given up to clearance. Equation 12 shows that in scaling the diagram for volumetric efficiency the effect of clearance has been taken into consideration, and also terminal pressures, but not that of the initial temperature and pressure. To provide for this, (12) must be multiplied by $\frac{T_1}{T}$ where T is the absolute temperature of the air at the instant that compression begins, and T_1 is 60 deg. + 460.6 deg.

Also, the pressure at the beginning of compression is nearly always less than that of the atmosphere, due to frictional losses, valve-spring resistance, etc. To provide for this, equation (12) must be multiplied by $\frac{P}{P_1}$ where P is absolute pressure shown by the intake line on the diagram, and P_1 is 14.7 lb.

The expression for the true or real volumetric efficiency therefore is

$$\text{True } E_v = \frac{T_1P}{TP_1}\left\{1 - \frac{V_c}{P_d}\left[\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}} - 1\right]\right\} \qquad (13)$$

Under some circumstances, this method of volumetric efficiency determination is to be avoided, for results then obtained will be misleading and consequently worse than worthless. Leaky suction valves or stuffing-boxes, and a cylinder scored at or near the end of the stroke will produce an almost perpendicular re-expansion curve.

These defects may be detected on the diagram by plotting the theoretical curves and comparing with the actual. Fig. 5 shows the typical case of leaky valves on one end and their effect on the volumetric efficiency and comparison curve.

A better way to determine the volumetric efficiency of a compressor under all conditions of cylinder, is to actually measure the air delivered and divide by the piston displacement. The air may be measured by a standard orifice or by a system of inclosed tanks. The former method has been frequently described. The latter is as follows:

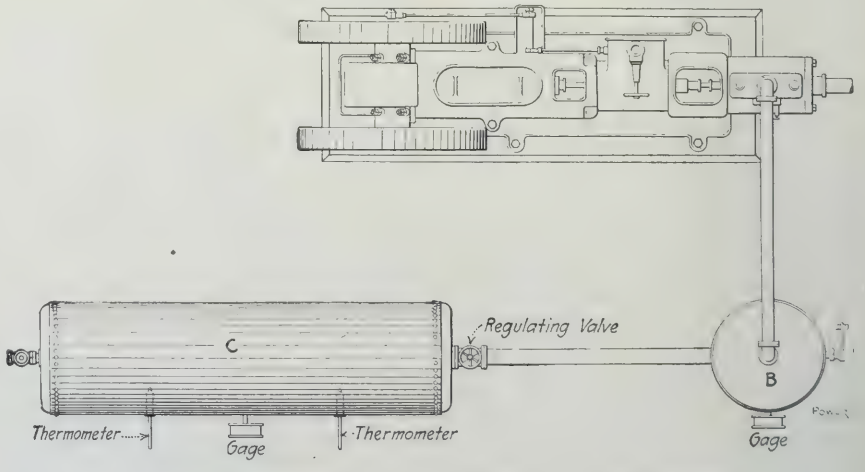


FIG. 6. MEASURING AIR DELIVERED TO COMPRESSOR

Connect the air compressor to two inclosed tanks, B and C, as in Fig. 6, with a regulating valve between B and C, air gages and thermometers as shown.

Through the regulating valve, the air pressure in B may be maintained at the desired pressure for which the volumetric efficiency is to be determined. The tank C then may be pumped up from pressure P_1 to P_2, both lower than the pressure in B. It is always advisable to begin the test at an initial pressure higher than atmospheric. Shut off tank C and start the compressor. Watch the gage on C until the needle reaches a suitable point, say 15 lb., and from then on, until pressure P_2 is reached, count the revolutions of the compressor and observe temperatures and the time of the run. A number of runs should be made and the mean of the results found and substituted in the formula following:

Let

P = Atmospheric pressure = 14.7 lb.;

P_1 = Initial absolute pressure in tank C;

P_2 = Final absolute pressure in tank C;

T = Absolute room temperature in degrees F.;

T_1 = Absolute initial temperature of air in tank C;

T_2 = Absolute final temperature of air in tank C;

V = Volume in cubic feet of tank C;

V_1 = Free air equivalent of air in tank at beginning of test;

V_2 = Free air equivalent of air in tank at end of test;

V_3 = Actual amount of air pumped into tank;

$R_1 = \frac{P_1}{P}$ atmospheres beginning of test;

$R_2 = \frac{P_2}{P}$ atmospheres end of test.

Disregarding temperature,

$$V_3 = V_2 - V_1 = \textit{Volume of air compressed}$$

$$V_1 = V\frac{P_1}{P} \textit{ and } V_2 = V\frac{P_2}{P}$$

$$V_3 = V\left(\frac{P_2}{P} - \frac{P_1}{P}\right)$$

$$V_3 = V\ (R_2 - R_1) \qquad (14)$$

The theoretical quantity of air pumped is equal to the piston displacement of the compressor, or $l \times a \times n$, where l = length of stroke in feet, a = area of the piston in square feet, and n = number of strokes. The observed volumetric efficiency is then represented by the formula

$$\textit{Observed } E_v = \frac{V\ (R_2 - R_1)}{lan} \qquad (15)$$

Correcting for temperatures, (14) becomes

$$V_3 = VT\left(\frac{R_2}{T_2} - \frac{R_1}{T_1}\right) \qquad (16)$$

If the barometer pressure is other than 29.92 in. of mercury, the formula should be corrected and made to read as follows:

$$V_3 = \frac{VT}{B}\left(\frac{29.92\ R_2}{T_2} - \frac{29.92\ R_1}{T_1}\right) \qquad (17)$$

The expression for real volumetric efficiency then becomes

$$\textit{Real } E_v = \frac{\frac{29.92}{B}\left(\frac{R_2}{T_2} - \frac{R_1}{T_1}\right)}{lan} \qquad (18)$$

The definition and formula derivation have shown conclusively that unless a guarantee of volumetric efficiency is accompanied by a statement of the terminal pressure and inlet temperature and pressure, it is meaningless. If these conditions are stated, the guarantee is nothing more nor less than a guarantee of clearance volume. So, really, what excuse is there for the existence of the term "volumetric efficiency?"

Overall Efficiency

This efficiency is the most important of all to the user, for it refers directly to the cost of operation. It means the cost in fuel, upkeep, supplies, interest and depreciation of air delivered to the receiver or pipe line, and is consequently a combined statement of mechanical compression and volumetric efficiencies as well as reliability.

The following is a general expression of overall efficiency referred to the isothermal no clearance base.

$$E_o = \frac{\left(\begin{matrix}\textit{Boiler hp. per 100 cu.ft. free air per}\\ \textit{min. actually delivered}\end{matrix}\right)}{\left(\begin{matrix}\textit{Isothermal hp. per 100 cu.ft. free}\\ \textit{air per minute}\end{matrix}\right)}$$

This expression does not take into consideration all the factors that affect overall efficiency, but even as it is, it is something definite and a much more satisfactory guide than any yet given. Elaborate tests over long periods of time are necessary to determine the true overall efficiency of any machine and no accurate formula can be derived that will take into consideration even one of the variables, such as upkeep or depreciation. The most reliable information on the subject is a series of tests made by Richard L. Webb on a number of air compressors in the Canadian mining district, published in "Compressed Air Plant" by Prof. Robert Peele.

Economy Essentials

The foregoing discussion shows that the economy of an air-compressor unit, depends:

(1) On the mechanical construction, that is, the size and proportion of bearings, and wearing surfaces; lubricating system and general design of parts.

(2) On the length and volume of ports in the air cylinder. Long and tortuous ports and air passages increase the losses by heating the incoming air before compression begins.

(3) On the cooling devices and water jackets and temperature of cooling water.

(4) On the surrounding conditions, that is, altitude at which the machine is being operated and atmospheric temperature.

(5) On the clearance spaces.

(6) On economy of power end.

Weight of Mercury

By C. N. Cross

Mercury is very frequently used for measuring pressures varying from 0.1 to 4 or 5 lb. per sq.in. and occasionally as high as 150 to 200 lb. in steam-gage calibration. It is used almost exclusively to determine the vacuum in condensers, so that it is important to know the weight of mercury in pounds per square inch, and to know how the weight or density varies with change of temperature. The errors introduced by neglecting to correct for changes of density are much smaller for steam pressures above atmospheric, than when measuring high vacuum.

For accurate pressure measurement mercury should be pure, as impurities alter its density. Impure mercury shaken with air yields a black powder, caused by the oxidization of the impurities. Another indication is the "tail" which is seen when a drop of the metal is run over an inclined surface. Pure mercury has a silver-white color and globules retain a perfectly spherical shape. The surest way to free it from impurities is to distill it. Its boiling point is about 675 deg. F. It may be more easily purified by treating it with a 5 or 10 per cent. solution of nitric acid. The mercury is run through a very small tube into the vessel containing the nitric acid, and the treatment may have to be repeated several times.

The weights of mercury in the accompanying table, which have been transposed from metric to English units, were determined by Thiesen and Sched.

WEIGHTS OF MERCURY AT DIFFERENT TEMPERATURES

Temperature Deg.	Weights of Mercury lb. per cu.in.	Correction Factor to Reduce to 32 deg.F.
32.0	0.49116	1.0000
39.2	0.49080	1.0008
50.0	0.49028	1.0018
59.0	0.48980	1.0028
62.6	0.48964	1.0032
68.0	0.48932	1.0037
77.0	0.48895	1.0045
86.0	0.48850	1.0053
95.0	0.48803	1.0064

As the third column shows, the density of mercury decreases about 0.64 per cent. when the temperature is increased from 32 to 95 deg. F. From the second column the following values have been computed. At 62.6 deg. F., a column of mercury 1 in. high produces a pressure of 0.48964 lb. per sq.in., 7.834 oz. per sq.in., 13.570 in. of water at 62.6 deg. F., 70.502 lb. per sq.ft., 1.1308 ft. of water at 62.6 deg. F., 927.95 ft. of air. The last value is for dry air at a pressure of 29.921 in. of mercury and 62.6 deg. F.

Dundee Meeting of British Association

By John B. C. Kershaw

A review of the papers bearing upon power generation and relating to heat flow from gases into cylinder walls of gas engines, coal mine explosions, new bomb calorimeter, gas turbines and heat loss from bare pipes.

On Sept. 4 to 11, the eighty-second annual meeting of the British Association for the Advancement of Science was held at Dundee, Scotland. Five papers bearing upon power generation were contributed to the engineering section of the association, and these are abstracted in the following:

Efficiency vs. Nuisance

The presidential address to the members of the engineering section, by Prof. A. Barr, centered on the duty of the engineer to pay more attention to artistic considerations in the pursuit of his calling, and to practice the profession with more regard for the material interests and aesthetic susceptibilities of the general public. According to Professor Barr, when one examines into the immediate causes of the injuries and inconveniences that result from the activities of engineers, one finds that they are due in almost all cases to failures rather than to successes. The more completely the engineer achieves the primary end of his work the less is the damage or injury that can be laid to his charge. Following up this idea, Professor Barr discussed the subject of power generation from solid fuel in some detail, and gave it as his opinion that the present wasteful methods would soon give place to others that would be both more scientific and less harmful to the community. Following are some extracts from his address:

Progress towards economy is leading to a reduction, and possibly to the entire elimination of all the nuisances associated with the older methods of power and heat production. The great improvements that have recently been made in producer plants and gas engines have rendered out of date, as regards economy, at least the smaller sizes of steam plants which are so fruitful a source of injury and inconvenience to the community; and we now have engines of the Diesel, and the so-called semi-Diesel types, that can utilise natural oils, and oils obtained in the distillation or partial distillation of coal, not only with an efficiency hitherto unattained in heat engines, but "without injury or damage to any one single person"—except possibly the maker of inferior plants.

Present indications point to the coming of a time in the near future, when the power and heat required for industrial and domestic purposes will be distributed electrically in a perfectly inoffensive manner from large central stations; and even at these stations there will be no pollution of the atmosphere that could give the most sensitive of critics any just grounds of complaint against the intrusion of science into our lives.

It is just a hundred years since passengers were first carried on the Clyde in a mechanically propelled ship, and today —when they are not too completely obscured by smoke—we can see the successors of the 'Comet,' plying on that river with power plants of greatly superior overall efficiency but showing little advance in regard to the combustion of the fuel. Had the emission of smoke from river craft been prohibited years ago, there is little doubt that engineers would have let few days pass without arriving at some solution of the problem of inoffensive power production on steamboats, and the demand for economy would have looked after itself. How much better it would be were engineers to take the wider view of their duties and responsibilities and realise that they were acting contrary to the true spirit of their profession, when they produce appliances that pollute the atmosphere for miles around, to the hurt and inconvenience of those whose use they are intended to serve. But this year a ship has left the Clyde that we hope may be the forerunner of a new race, which will attain a higher efficiency than any of the direct descendants of the 'Comet.'

Gaseous Explosions

The fifth report of the committee appointed by the British association to investigate "Gaseous Explosions," with special reference to the temperatures attained in internal-combustion engines, was presented by Dugald Clerk. In the present report, the committee has dealt chiefly with the question of the rate of heat flow from the gases, or working fluid, into the cylinder walls. The report states that it is now fully recognized that a great part of the difficulties experienced in the construction and working of internal-combustion engines is due to heat flow, and that the subject has been brought into greater prominence in recent years by the construction of larger engines and cylinders in which these difficulties have only partially been overcome.

The important practical question is the mean rate at which heat goes into each part of the surface, and the resulting mean distribution of temperature. The chief problem in designing large gas engines is to control the mean temperature distribution by water jacketing or otherwise in such a way, that the metal does not get overstrained by unequal expansion, nor reach a temperature sufficient to ignite the gas. The temperature gradient necessary to sustain the flow of heat from the inside of a combustion chamber to the external water is not likely to exceed 120 deg.F. per inch. At places where the metal is not thick, and effective external circulation of water is possible, cooling does not present great difficulty; but at places which are not near to the cooling water, so that the heat has to travel a long way, the temperature must be high to give the necessary gradient. Thus the central portion of the head of an ordinary flat-faced piston, if not water cooled gets very hot, reaching a temperature of perhaps 1100 deg.F. in a four-cycle engine of 24 in. bore. The piston expands considerably in consequence, the expansion being greater at the center than at the edge, which is accordingly put into tension. In larger cylinders the stresses in the piston set up by unequal heating, and the danger of pre-ignition arising from the hot metal, necessitate the cooling of this part by the circulation of oil or water. Even then the greater thickness of metal in certain parts of the combustion chamber, and the difficulty of keeping the water flowing properly in every corner, may cause high local temperatures.

The heat carried away by the cooling water and by radiation, is the total given to every part of the walls, and its measurement gives no information on the important question of the manner in which the flow is distributed over the walls. It is certain, however, that the greater part of the heat flow in a cycle occurs in a comparatively short time just after the moment of ignition, and passes therefore into the surface of the combustion chamber and valves and into the face of the piston. But little goes into the barrel of the cylinder which is not uncovered, until the density and temperature of the gases have fallen. That this must be so is obvious, but the magnitude of the effect is perhaps not generally recognised. Dugald Clerk found in his experiments on the compression and expansion of flame that the average heat flow per square foot per second in the first three-tenths of the stroke, is three times that of the average over the whole stroke for equal temperature differences, and he calculates that the actual rate of heat flow in the first three-tenths, is six times that of the whole stroke in ordinary gas engines, working at full load. This estimate, however, does not include loss due to radiation before maximum temperature. In the actual firing and expansion stroke of a gas engine the difference must be even more when radiation and other losses incurred before maximum temperature are included, and it is probable that in discussing the problem of cooling the metal, it is a sufficiently good approximation to neglect the heat flow into the outer half of the barrel altogether. Professor Hopkinson informs the committee that he has worked a gas-engine cylinder of over 30 in. diameter, in which there was no water circulation round the barrel at all. The whole of the heat passing into the barrel was in this case removed either by radiation or by conduction into the piston, nor was the cooling which was applied to the piston much more than that found necessary on other parts of the walls of the combustion chamber. In small engines with uncooled pistons, the water jacket around the barrel is necessary to keep the piston cool.

In the scientific analysis of gas-engine phenomena, the facts stated in the last paragraph are important, because they show that the heat flow is not much different from that which would occur in a closed vessel of invariable volume having the form and size of the combustion chamber, the mixture fired having of course the same composition, density, movements, etc., as in the engine. Some

allowance must be made for the fall of temperature and density which occurs in the initial stages of the expansion in the engine, but this will be of the nature of a correction, and will not affect the value of the general conclusions as to the effect of the various factors in heat flow which may be drawn from closed vessel experiments.

The report then discusses in detail the various factors that affect heat flow, these being enumerated as follows: The state of cleanliness and brightness of the internal walls, radiation from the gas, effect of cylinder dimensions, effect of density and effect of turbulence. The last-named effect is one discovered by Dugald Clerk, and is due to the eddying or turbulent state of the gases within the cylinder, following upon the suction stroke of the engine. In consequence of this turbulent state of the gases, the explosion occurs more rapidly than in a quiescent gas mixture, and the effective heat conductivity of the gas is increased. Dugald Clerk has worked out a practical method of measuring and recording this increased conductivity, and the report contains two diagrams illustrating his method and its results.

Prevention of Coal Mine Explosions

Professor Dixon, of Manchester, followed Dugald Clerk with an account of the work carried out at the government's experimental station, for investigating the causes and prevention of coal-mine explosions, located at Eskmeal, near Scawfell in Cumberland. This plant includes an explosion gallery 900 ft. long and 7.6 in. in diameter, with similar galleries of smaller dimensions, and apparatus for charging these galleries with any desired gas mixture and for observing the force of the explosions produced by firing the mixture. So far as the experiments have gone, they prove that the best way to prevent dust explosions in mines, is to use some incombustible dust, with which the combustible coal dust must be covered in every gallery and other place where it collects.

Incidentally the experiments have proved the correctness of Dugald Clerk's observations on the effects of turbulence upon the speed and force of the explosive wave, a small fan being introduced into the gallery to create the desired turbulent effect. The most destructive explosions were obtained when this fan was run at a high speed to churn up the dust and gas mixture.

New Bomb Calorimeter

A paper, of which the following is an abstract, was read by Robert S. Whipple upon a new form of bomb calorimeter for testing the calorific value of coal and other forms of fuel:

The bomb form of calorimeter is now frequently employed to determine the calorific value of coal, because the combustion of the coal is more complete than in those calorimeters in which the oxygen is admitted at atmospheric pressure or slightly above.

The instrument designed by Prof. C. Fery, of the Ecole de Physique et de Chimie, Paris, is of the bomb type, being in general design somewhat similar to the Mahler instrument, but greatly simplified in its details. The bomb consists of a light iron vessel weighing about 2.2 lb., and having a capacity of about 15 cu.in. supported in the center of a brass vessel by two disks of constantan (an alloy of copper and nickel). As the mass of metal in the bomb is small and no water jacket is used, the rise in temperature of the bomb due to the combustion of the coal is large, averaging about 36 deg.F. when 0.5 gm. of coal is burnt. It is evident that with such a large temperature rise the measurements need not be made with the same degree of accuracy as in the case of the usual type of bomb calorimeter, in which the temperature rise amounts to only 4 or 5 deg. F. for the same quantity of coal.

The temperature rise is measured by the thermo-electric force generated by the constantan disks and the iron bomb, the surrounding envelop acting as the cold junction of the thermocouples. These couples give an electromotive force of 22.2 micro-volts per degree F., giving on a pointer galvanometer a deflection of 2.4 in. for a rise of 36 deg.F. It is thus possible to estimate the temperature rise of the bomb to $\frac{1}{360}$ of the total rise in temperature.

The loss of heat from the bomb is due to three causes: (1) conduction along the constantan disks; (2) convection currents in the air space surrounding the bomb; and (3) radiation from the walls of the bomb.

The effect of the loss by conduction is only to lower the maximum temperature obtainable as compared with what it would have been if the supporting disks were nonconductors. The losses due to convection and radiation are negligible as compared to those due to conduction.

The coal under examination is placed on a small tray in the bomb, the latter being filled with oxygen at about 200 lb. pressure. The coal is fired electrically, the maximum temperature rise on the galvanometer taking place about $1\frac{1}{2}$ min. after ignition.

Experiments show that the gas pressure can be varied from 150 to 250 lb., and the weight of coal burnt from 0.2 to 0.7 gm., without any difference in the results obtained. The instrument is standardized by burning either sugar, carbon, or standardized coal briquets, the calorific values of which are known.

Gas Turbines

On Sept. 9, an interesting discussion upon "Gas Turbines" was opened by a paper from Dugald Clerk. The author stated that engineers engaged in the development of the internal-combustion engine, have long recognized that great advantages would come from substituting rotary for reciprocating movement, and accordingly many attempts have been made to produce a commercial gas turbine. So far, no attempt has succeeded; the practical difficulties have proved too serious. Much useful knowledge, however, has been obtained by able experimenters, and we are now in a position to consider the difficulties afresh, with some experimental data at our disposal.

Details of the turbines and work of Armengaud, of Karovodine and of Holzwarth were given.

The Armengaud turbine was of the constant-pressure type, and the largest experimented with gave 300 b.hp. with a fuel consumption of 3.9 lb. of petrol per b.hp.-hr. This was nearly eight times the petrol consumption of the ordinary internal-combustion type of engine, and was much too high for this turbine to be considered as a commercial engine.

The Karovodine turbine was of the explosion type without compression, and of exceedingly ingenious construction. It contained four explosion chambers, having four jets actuating a single turbine wheel of the De Laval type, about 6 in. in diameter, and running at 10,000 r.p.m. This turbine gave 1.6 b.hp. with a fuel consumption of 6.5 lb. petrol per b.hp.-hr., and thus consumed more than double the petrol used by the larger constant-pressure gas turbine of Armengaud.

The Holzwarth gas turbine was also operated on the explosion principle, but the construction and action were very different from that of Karovodine. The turbine in general arrangement outwardly resembled the Curtis steam turbine, in that the turbine wheel rotated in a horizontal plane, the spindle or shaft was vertical, and a dynamo was mounted on this spindle above the turbine. In the Holzwarth turbine ten combustion chambers were provided, each of a pear or bag shape. They were arranged in a circle around the wheel, and were cast so as to form the base of the machine. The wheel was of the Curtis type, with two rows of moving and one of stationary blades.

The inventor has built a gas turbine of a rated capacity of 1000 hp. of this type, the wheel of which rotates at 3000 r.p.m. Many practical difficulties have been met with in the selection of a gas suitable for the operation of this turbine, and ultimately the turbine was worked with a producer gas made from coke. The maximum power obtained from the turbine in the early experiments was 160 b.hp., with all ten explosion chambers in operation, and an explosion pressure in the chambers of 45 lb. above atmospheric pressure was obtained. It was found, however, that higher explosion pressures and a greater horsepower per chamber could be obtained with fewer chambers in operation. With five chambers, a total of 145 hp. was obtained, at 29 hp. per chamber; and with four chambers, the total was 121.6 b.hp. at 30.4 hp. per chamber. With 10 chambers in operation the power fell to about 15 hp. per chamber. From these experiments, Mr. Holzwarth concluded that the successive explosions interfered with each other and prevented proper charging and action.

The latest results with this turbine (communicated by Mr. Holzwarth to Dugald Clerk by telegraph, specially for

the meeting) have shown a total power production of 450 hp. with this turbine, and an efficiency of 23 per cent., but it is not clear upon what basis this efficiency is calculated.

Mr. Clerk in closing his paper, summarized the present position with regard to the gas turbine, as follows:

> The existing internal-combustion engines are quite satisfactory for small and moderate power units; but the weight increases so rapidly with increase of cylinder diameter that large units such as 20,000 hp. per shaft, easily attained by the steam turbine, have proved quite impossible for the reciprocating gas engine. To apply internal combustion for the purpose of such large units, it appears necessary to dispense with the cylinder piston and crank. I fear that this cannot be done on the lines of either constant-pressure or explosion turbines here shortly discussed. The results obtained only appear to show that progress can hardly be expected on the lines of flame impinging on turbine blades, either in impulse or reaction turbines. Several engineers have suggested the use of explosion to give water velocity, which velocity actuates a water turbine of some form. Mr. Humphrey, in his paper to the Institution of Mechanical Engineers, figures such a combined gas and water turbine; and although in the form suggested by him the conditions required too cumbrous a machine, yet it does seem that the more hopeful line is to use explosion and expansion to give water velocity, and so avoid all heat difficulties in the turbine part of the apparatus. True, such an arrangement still necessitates a reciprocating mass of water, but it will probably be found that great gain in weight can be obtained by the suppression of the piston, connecting rod, engine frame and crank. This line seems to me to be a much more hopeful one than any scheme involving the direct contact of flame with turbine blades.

Sir William White, who opened the discussion, emphasized the advantages of the rotary principle for engines designed for ship propulsion, and gave as examples the comparatively small space (relative to the ship's tonnage) occupied by the enormously powerful engines of torpedo destroyers, and of the "Mauretania" and "Lusitania," and also the freedom from vibration of these ships, as compared with ships of similar size and power, driven by reciprocating engines. Sir Charles Parsons was experimenting on the subject of gas turbines, but the difficulties enumerated by the author of the paper had so far blocked the way to all further progress toward the design of a successful and economical gas turbine.

Prof. Sylvanus Thompson referred to the upper and lower temperature limits (1800 deg. C. and 400 deg. C.) given for gas-turbine work in the course of the paper, and thought that these could be improved upon. The use of carborundum for lining the explosion chambers of gas turbines did not seem to him a practicable means of protecting the metal from the hot gases, for the carborundum would be certain to crack and become disintegrated, with disastrous results to the turbine blades. Mr. Ferranti was stated to be carrying on experiments with a new type of gas turbine, and Professor Thompson would have liked some information as to the progress he had made in solving the difficult problem.

Sir Alfred Ewing thought that the difficulty as regards temperature would be overcome by the discovery of a metal (or material) which would prove durable at temperatures of 1800 deg. C. or over, and could be used for cylinders and explosion chambers of gas engines and gas turbines.

Dr. Robert Mond gave details of the experiments made by Humphreys some years ago, with the combined gas and water turbine, and stated that these experiments failed since this turbine required another engine to operate it.

F. Samuelson defended the steam engine and steam turbine as prime movers against the attack of gas-engine and gas-turbine makers, and was convinced that much higher efficiencies than had yet been obtained could be reached with steam turbines, if steam pressures up to 400 lb., and steam temperatures of 750 deg. F. were used for their operation. This would increase the practical efficiency of the turbine by 20 per cent.

Dugald Clerk in his reply to the discussion explained that such high temperatures as the last speaker advocated for steam turbines were impracticable, since there is some alteration in the physical structure of the iron which caused its swelling or "growth" when exposed repeatedly to temperatures of 700 deg. F. Mr. Parsons had convinced himself and the scientific public that the only practicable method for using higher temperatures than are at present customary in steam-turbine practice, was to utilize the first pressure and temperature drop, in the impulse type of turbine.

As regarded the position of the gas engine and gas turbine in face of the growing competition of the Diesel type of motor, he pointed out that the check to any great expansion of the use of oil engines, was the comparatively limited supply of oil fuel.

The total world's output of coal was now in the neighborhood of 1100 million tons per annum; while the total oil production was only about 45 million tons. From this latter total must be deducted the oil required for illuminating and lubricating purposes, also that for motor vehicles and aëroplanes. The balance represented the oil for power and heating purposes, and as compared with the 1100 million tons of coal, it represented a very small proportion of the world's fuel supply. The makers of steam and gas engines need not be apprehensive therefore as to the growing competition of the oil-driven type of engine, and so long as cheap coal supplies are available the future of their industry was safe.

Heat Loss from Bare Pipes

An interesting paper upon the "Transmission of Heat from Steam-heated Pipes through Conducting and Nonconducting Materials" was communicated by C. H. Lander and J. E. Petavel. The results of this investigation were important in their bearing upon the choice and application of heat-insulating coverings for steam pipes and boilers.

The rate of flow of heat from a hot metal cylinder depends on the temperature of the cylinder, its diameter, and the pressure of the air surrounding it. A number of measurements have been made, using cylinders ranging in diameter from 0.02 in. to 12 in., the smallest ones being heated electrically up to a temperature of 2000 deg. F. while for the larger ones steam heating was adopted.

Many of the radiating cylinders were surrounded by an air-tight inclosure, and the pressure of the air within this was varied from atmospheric pressure up to 3000 lb. per sq.in. The part of the work bearing on the loss from steam pipes of ordinary sizes may be summed up as follows:

Measuring the heat loss per hour in British thermal units per square foot per degree Fahrenheit of temperature difference, with saturated steam at atmospheric pressure and 100 lb. gage, the external temperature being 60 deg. F., the results are shown in the accompanying table.

Under these conditions the heat loss is mainly due to convection. Taking as an example a pipe of 1-in. external diameter with emissivity of 3 B.t.u. for steam at atmospheric pressure, the loss by radiation will be 0.44, or say, 15 per cent.

The loss by conduction in an inclosure 8 in. greater in diameter than the pipe, figures out at 0.13, or about 4 per cent. and would be rather less in an inclosure of larger diameter. Thus convection accounts for 81 per cent. of the loss from a 1-in. pipe containing steam at atmospheric pressure, and a similar reasoning would lead to the conclusion that it is about 80 per cent. for steam at 100 lb. pressure. For very small diameters below 1 in. conduction becomes relatively more important, and may even amount to one-half of the total.

The figures refer to an oxidized surface, such as that of the ordinary unpainted steam piping. Careful polishing would diminish the amount lost by radiation to one-quarter, while machining the pipe to an ordinary finish, would halve the radiation.

In the first case, the loss from the pipe containing steam at atmospheric pressure, would decrease to 2.67 and the convection would be 91 per cent. of the total, in the second the total loss would decrease to 2.78 and the convection would be 87 per cent. of the total. As an illustration the 2-in. radiator used in the above work, lost 2.55 B.t.u. per deg.

when coated with lampblack varnish, this being reduced to 2.32 B.t.u. after cleaning.

The object attained by surrounding a pipe with insulating material is to check convection currents and thus eliminate or diminish the loss due to this cause, the insulating material also reduces radiation.

HEAT LOSS FROM METAL SURFACES SUCH AS BARE STEAM PIPES OF VARIOUS DIAMETERS

External dia. of radiator in.	Heat loss per hr. in B.t.u. per sq.ft. per deg. F.		Conduction only, taking K = 144*	
	0 lb. gage	100 lb. gage		
0.024	9.20	10.5	3.4	Measured in an enclosure
0.043	6.21	6.90	2.2	0.81 in. dia.
0.20	3.94	6.0	0.39	Assuming layer of air to be 4 in. thick round radiator
1.00	3.00	4.0	0.13	
2.00	2.55	2.8	0.089	
3.00	2.30	2.4	0.074	
5.00	2.05	2.1	0.060	
12.00	1.97		0.047	
Flat layer 4 in. thick.			0.036	

*The constant K is given in B.t.u. per hr. per sq. ft. of surface, per deg. F. Temperature difference, for a layer 1 in. thick.

All solid materials used as insulators are better conductors than air, and using an insulating material, therefore, really increases the conduction loss. For instance, with a wire of small diameter, in air at low pressure, conduction is more important than convection, and an insulating covering will actually increase the thermal loss.

Even with a 1-in. pipe, a material such as slag-wool or asbestos if applied in a solid compressed block, would increase the total loss. The insulating effect obtained depends on so arranging the substance that the least quantity of material will subdivide the space around the pipe into separate air cells as perfectly as possible. Thus the efficiency of slag-wool is at a maximum when the solid material occupies about one-fourteenth of the space. Under these conditions the loss by conductivity is increased from 0.13 to 0.40, but assuming radiation to be eliminated, the loss by convection is reduced from 3.2 to 0.14 and the total loss is only about one-seventh that of the bare pipe.

When the material is either more closely or more loosely packed, the total loss increases; in the former case because of the increased conductivity of the solid material, in the latter case because of the greater loss by convection.

In certain substances, such as cork or wood, the natural arrangement of the material is cellular, and as such, satisfactory from the point of view of heat insulation.

Finally, with regard to the effect of diameter, the rate of loss falls at first rapidly as the diameter increases, but the variation becomes small for cylinders above 4 in. in diameter.

Combined Hoist and Air Compressor

By B. J. Lowe

At a certain mine compressed air is used to operate the brake on a single-drum, direct-motion, Corliss winding engine handling but one cage for elevating men from the mine 1000 ft. below the surface. The engine had originally been steam-driven, having two 18x48-in. high-pressure cylinders. An 8-ft. drum was mounted on the shaft. The engine has been remodeled and is now driven by a 720-hp. geared motor. The gear teeth are of the double-herringbone type and have a ratio of 4.5 to 1.

The pinion is made of nickel steel and the spur wheel of cast steel, the shaft of which couples direct to the drum shaft by a "ham" coupling. One cylinder of the engine is used, the crosshead and connecting-rod being coupled to the crank. On the coupling side of the drum the crank has been removed and a crank disk substituted which contains the male side of the coupling. The motor is operated by three-phase alternating current.

In lowering the men, the post brake is released and the cage descends of its own weight to the bottom of the shaft. While the rope and drum are paying out the steam cylinder is turned into an air compressor, as the valve-gear is still on the cylinder side and the reversing link is set against the engine. The exhaust valves therefore are the admission valves and the former admission valves are the exhaust valves.

The air is compressed against a special relief valve under the operators' platform, which is operated by a lever. The operator controls the speed of the drum by opening and closing this relief valve, reducing or increasing the air pressure against the engine piston. The new arrangement is clearly illustrated in Fig. 1.

The brake path of the drum wore out of true, and a special stand was rigged up and fastened to the brake post. Upon this stand a rock-drill cradle and cylinder were placed to act as a slide rest, and upon the cylinder an electric emery grinder was fastened. To operate, the motor driving the drum is started and then the grinder is set to work truing up the brake path. The arrangement is shown in Fig. 2. This made a very good job and gave a better gripping effect for the brake blocks, which is very essential on a winding engine raising or lowering men.

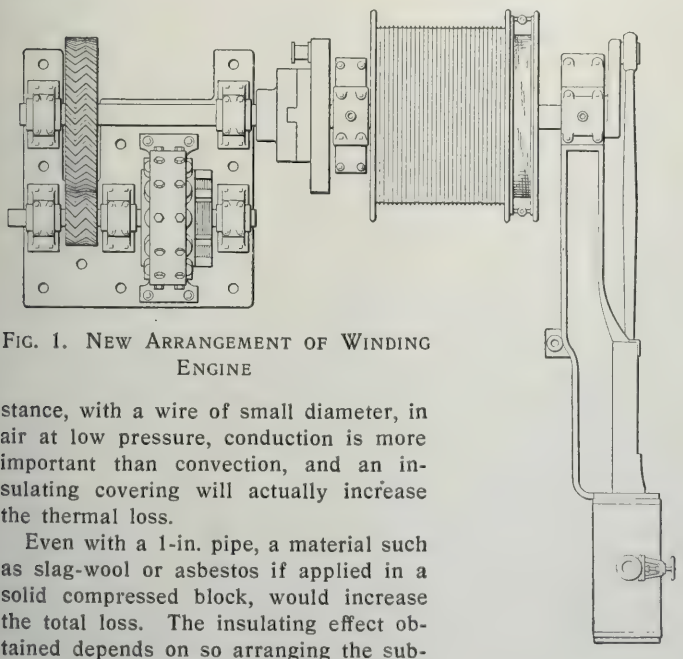


Fig. 1. New Arrangement of Winding Engine

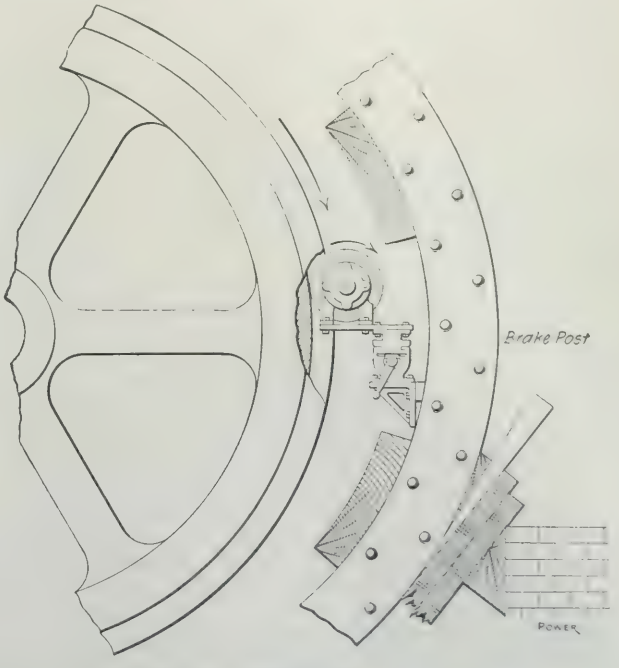


Fig. 2. Turning the Brake Rim

The Meklong Railway Co., Ltd., of Siam, is about to experiment with an engine burning both firewood and oil.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Use of Back-Pressure Valve

Can exhaust steam be used for supplying heating coils of a drying room without having an exhaust back-pressure relief valve?

B. L.

A back-pressure relief valve should be employed to regulate delivery of steam of sufficient pressure to the heating coils and to relieve the exhaust of unnecessarily high back pressure.

Hard Drawn vs. Annealed Wire

What is the relative resistivity of hard-drawn copper wire and of annealed copper?

J. D.

The resistivity of hard-drawn copper wire is about 2.7 per cent. greater than that of annealed copper.

Evaporation from and at 212 Degrees

What is meant by equivalent evaporation from and at 212 deg.?

G. S.

The relative quantity of water that would be evaporated if the same amount of heat had been absorbed only as latent heat of vaporization at 212 deg. F., and under atmospheric pressure.

Calculating Length of Belt

Can you give a practical formula for calculating the length of an open belt, knowing the distance between centers and diameters of the pulleys?

R. A.

Where

D = Distance between centers;
R = Radius of the larger pulley;
r = Radius of the smaller pulley;
x = Required length of belt;

all dimensions being taken either in feet or in inches, a close approximation to the true length is given by the formula

$$x = 2D + \pi (R + r) + \frac{(R - r)^2}{D}$$

Bright Work on Engines

What are the practical advantages of keeping engine parts bright?

J. C.

Heat radiates slower from highly polished surfaces than from dull or rough surfaces; therefore heads and other uncovered parts of engine cylinders should be kept highly polished to hinder cylinder condensation and obtain higher economy of steam. All working parts of an engine should be kept bright and clean by frequent grooming, not only for good appearances, but for early discovery of any defects which may have developed in the material or adjustment of working parts.

Horsepower Constant

What is the horsepower constant of an engine having piston 20 in. diameter, with stroke of 40 in. and running at 70 r.p.m.?

D. P.

The area of the piston would be $20 \times 20 \times 0.7854 = 314.16$ sq.in. and with the stroke of 40 in. and 70 r.p.m., the piston speed would be

$$\frac{40}{12} \times 2 \times 70 = 466.66 \textit{ ft. per min.}$$

As the horsepower constant is the number of horsepower that would be developed per pound of mean effective pressure, it is the same as the number of horsepower developed by 1 lb. mean effective pressure. The horsepower constant of the given engine, therefore, is

$$\frac{1 \times 314.16 \times 466.66}{33{,}000} = 4.443$$

Stopping a Boiler Leak

What is the best way to stop a leak which appears to extend for 2 to 3 in. along one of the girth seams on the bottom of a 66-in. return-tubular boiler which is in first-class condition in other respects, if calking the leak has proved to be only a temporary remedy?

J. P.

Failure to hold after careful calking may indicate that the leak is due to greasy feed water. Feed about a half pound of soda ash per hour during several days running, then wash the boiler out with clean water, scrape away the former calking and recalk lightly. When the boiler is refilled add 5 to 10 lb. of cornstarch. Further light calking may be necessary, but if a small leak cannot be taken up in the manner described, it may be due to folds of the shell plate which can only be made tight by removing a number of rivets on each side of the leak, drawing the plates close together by bolts and pounding, and replacing the bolts by a good job of snap riveting and calking.

Power to Drive an Ammonia Compressor

What would be the horsepower required for a double-acting ammonia compressor, 15x30 in., 65 r.p.m., with a suction pressure of 18 lb. and a condenser pressure of 200 lb.?

H. R. J.

The refrigerating capacity of the compressor may be expressed by the equation

$$T = \frac{S \times D \times E}{F}$$

in which

T = Refrigerating capacity of compressor per 24 hr.;
S = Piston speed in feet per minute = 325 ft.;
D = Apparent displacement in cubic feet for 1 ft. of piston travel = 1.227;
E = Displacement efficiency of the compressor = 80 per cent.;
F = Cubic feet of gas per minute per ton per 24 hr. = 3.82.

Substituting these values in the formula gives

$$\frac{325 \times 1.227 \times 0.80}{3.82} = 83.5 \textit{ tons}$$

and taking the horsepower per ton as 1.65 would make

$$83.5 \times 1.65 = 137.77 \textit{ hp.}$$

or approximately 140 hp.

Accumulation of Back-Pressure

Where exhaust steam from a pair of engines is used for heating factory buildings, what may be the cause and remedy for occasional failure of the balanced back-pressure valve to relieve excessive back pressure?

J. N.

The trouble is likely to occur where the back-pressure valve is at the foot of an exhaust pipe which has its surface exposed to the weather. Steam leaking through the back-pressure valve after condensing in the exhaust pipe above the valve, falls to the valve and helps to weigh it down. If the exhaust is then rejected by the heating system, back pressure increases, finally causing the engine to slow down unless sufficient back pressure to raise the valve has sooner accumulated. A remedy is the simple one of relieving the condensation formed above the back-pressure valve by a small bleeder, which should be protected from frost.

Study Questions

This Week's Questions
Last Week's Answers

(111) If the resistance of 1000 ft. of copper wire at 90 deg. F. is 100 ohms, what will be the resistance per foot at 75 deg. F.?

(112) If an engineer is now one and a third times as old as he was when he took charge of his present plant, and an oiler, who was born five years before the engineer took charge, is now two years older than half as old as the fireman was when the engineer took charge; how old is each now if the sums of their ages, when the older is as old as the engineer was when he took charge, will be 150 years?

(113) Steam enters a jet condenser at 16 lb. abs. pressure. The temperature of the injection water is 48 deg. F. If the temperature of the discharge water is 115 deg. F., how much injection water (pounds) is necessary per pound of steam?

(114) A hollow shaft has an outside diameter of 3 in., and an inside diameter of 2 in. When twisted by a force of 3000 lb. at a distance of 1 ft. from the center or axis of the shaft, what will be the maximum unit stress developed?

(115) A volume of air under 60 lb. pressure is at a temperature of 350 deg. F. If the temperature is reduced to 32 deg. F., what will be the pressure?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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(106) The field current will be

$$\frac{E}{r} = \frac{110}{55} = 2\ amp.$$

Therefore the total armature current is 152 amp., and the armature loss is

$$I^2R = (152)^2 \times 0.03 = 693.12\ watts$$

(107) Four volumes may be considered—that of the water in the vessel (let it equal V_1), that of the ball (V_2), that of the frustum of a cone circumscribing the ball (V_3) and that of the frustum of a cone left beneath the ball (V_4).

By the conditions of the problem

$$V_1 + V_2 = V_3 + V_4$$

The formula for the volume of the frustum of a cone of revolution is

$$V = \tfrac{1}{3}\pi H\ (R^2 + r^2 + Rr)$$

where

$\pi = 3.1416$;
H = Height of the frustum;
R = Radius of one base;
r = Radius of the other base.

From this the volume of the vessel is

$$V = \tfrac{1}{3} \times 3.1416 \times 10\left[\left(\frac{9}{2}\right)^2 + \left(\frac{9}{4}\right)^2 + \left(\frac{9}{2} \times \frac{9}{4}\right)\right] = 371.1015\ cu.in.$$

A quarter of its contents (the volume of the water) will be

$$V_1 = \frac{V}{4} = 92.7754\ cu.in.$$

The formula for the volume of a sphere is

$$V = \frac{4}{3}\pi r^3$$

where

$\pi = 3.1416$;
r = Radius of the sphere.

The ball's volume then will be

$$V_2 = \frac{4}{3} \times 3.1416 \times r^3 = 4.1888\, r^3$$

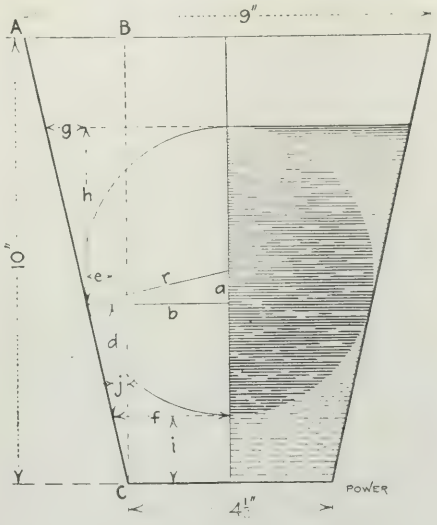
and r is the unknown radius to be determined.

The volume of the circumscribing frustum about this sphere will be

$$V_3 = \tfrac{1}{3} \times 3.1416 \times 2r\ [(b + g)^2 + f^2 + (b + g)\ f]$$

where (referring to the diagram)

$b + g$ = Radius of the top base;
f = Radius of the bottom base.



These three quantities can be expressed in terms of r by the relations of similar triangles as follows:

In the triangle $ABCA$ the side AB will be the difference between the radii at the top and bottom of the vessel, or 2¼ in.; the side BC is the height, 10 in., and the side AC is

$$\sqrt{AB^2 + BC^2} = \sqrt{\left(\frac{9}{4}\right)^2 + 10^2} = \frac{41}{4} = 10\tfrac{1}{4}\ in.$$

Then the radius of the sphere r, as drawn perpendicular to the side AC where the sphere is tangent to the vessel, and the line b perpendicular to BC (half a chord btween opposite points of tangency) and a, a perpendicular to this chord passing through the center of the sphere, form a triangle similar to $ABCA$. Hence

$$r : AC :: b : BC$$

or

$$r : \tfrac{41}{4} :: b : 10$$

$$b = \frac{10r}{\frac{41}{4}} = \frac{40r}{41}$$

Also by similar triangles

$$g : a :: h : b$$

or

$$g = \frac{ha}{b}$$

but, by construction

$$h = r + a$$

Again by similar triangles

$$r : AC :: a : AB$$

$$r : \tfrac{41}{4} :: a : \tfrac{9}{4}$$

$$a = \frac{\frac{9}{4}r}{\frac{41}{4}} = \frac{9r}{41}$$

Then

$$h = r + \frac{9r}{41}$$

and

$$g = \frac{\left(r + \frac{9r}{41}\right)\left(\frac{9r}{41}\right)}{\frac{40r}{41}} = \frac{45r}{164}$$

Therefore

$$b + g = \frac{40r}{41} + \frac{45r}{164} = \frac{205r}{164}$$

By construction

$$f = b - e$$

and by similar triangles

$$e : a :: d : b$$

or

$$e = \frac{ad}{b}$$

a and b have already been determined in terms of r and by construction

$$d = r - a = r - \frac{9r}{41}$$

hence

$$e = \frac{\left(\frac{9r}{41}\right)\left(r - \frac{9r}{41}\right)}{\frac{40r}{41}} = \frac{36r}{205}$$

and

$$f = \frac{40r}{41} - \frac{36r}{205} = \frac{164r}{205}$$

Substituting now in the equation for V_3

$$V_3 = \tfrac{1}{3} \times 3.1416 \times 2r\left[\left(\frac{205r}{164}\right)^2 - \left(\frac{164r}{205}\right)^2 + \left(\frac{205r}{164} \times \frac{164r}{205}\right)\right]$$

which reduces to

$$V_3 = 6.7073r^3$$

The volume of the frustum below the sphere is

$$V_4 = \tfrac{1}{3} \times 3.1416 \times i\left[f^2 + \left(\tfrac{9}{4}\right)^2 + \left(f \times \tfrac{9}{4}\right)\right]$$

f is already known in terms of r. By similar triangles

$$i : b :: j : a$$

$$i = \frac{bj}{a}$$

b and a are known in terms of r and, by construction

$$j = f - 2\tfrac{1}{4}$$

hence

$$i = \frac{\left(\frac{40r}{41}\right)\left(\frac{164r}{205} - \frac{9}{4}\right)}{\frac{9r}{41}} = \frac{1312r}{369} - 10$$

Therefore

$$V_4 = \tfrac{1}{3} \times 3.1416\left(\frac{1312\,r}{369} - 10\right)\left[\left(\frac{164\,r}{205}\right)^2 + \left(\frac{9}{4}\right)^2 + \left(\frac{164\,r}{205} \times \frac{9}{4}\right)\right]$$

which reduces to

$$V_4 = 2.383r^3 - 53.0145$$

Now substituting in the original equation

$$V_1 + V_2 = V_3 + V_4$$

$$92.7754 + 4.1888r^3 = 6.7073r^3 + 2.383r^3 - 53.0145$$

This reduces to

$$r^3 = 29.7439$$

and extracting the cube root

$$r = 3.0984$$

or approximately 3.1 in. The diameter of the ball is therefore 6.2 in.

108) The gage pointer indicating 9 in. of vacuum shows that the pressure within the condenser is that amount less than atmospheric pressure. As 1 in. of mercury corresponds to 0.49 lb. per sq.in. and atmospheric pressure is approximately 14.7 lb., the condenser pressure is

$$14.7 - (0.49 \times 9) = 10.29$$

or approximately 10¼ lb.

(109)

$$I.hp. \times Efficiency = B.hp.$$

The total steam consumption = 28.6 × i.hp., or

$$28.6 \times \frac{B.hp.}{E} = \frac{28.6 \times B.hp.}{0.87}$$

Then the steam consumption per brake horsepower would be

$$\frac{28.60}{0.87} = 32.87\ lb.$$

(110) The thermal efficiency is the ratio of the heat utilized to the heat supplied. If all the heat taken from the steam were converted into work the efficiency would be

$$\frac{T_1 - T_2}{T_1}$$

where

T_1 = Initial absolute temperature;

T_2 = Final absolute etemperature.

From steam tables the temperature of steam at 85 lb. abs. is 316.3 deg. F. and at 2 lb. abs. 126.15 deg. F. To convert these to absolute temperatures 461 deg. must be added to each. Then the efficiency will be

$$\frac{(316.3 + 461) - (126.15 + 461)}{(316.3 + 461)} = \frac{190.15}{777.3} = 0.2446$$

or nearly 24½ per cent.

The energy from Niagara Falls, including operation on both sides of the falls, is used at the rate of 126,000 hp. for electro-chemical processes 56,200 hp. for railway service, 36,400 hp. for lighting and 54,500 hp. for various industrial services, the total being 273,140 hp. Since the water of Niagara Falls represents probably 5,000,000 hp., it would seem that only about 5.5 per cent. of the available power is being utilized at present.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

A railroad company saves $81 a year by reducing the length of its pins one-sixteenth inch. When catching the point of this small economy we remember that the last pin which entered our system was about that much too long.

HORAN—Th' paapers sez we've no official standard fer drinkin' wather in these U. S.

DORAN—They're roight, Terence. Ivry toime Oi gets a sthick in moine, Oi puts me foot in ut.

Power-plant proverb: Into the exterior darkness cast thy whinings and grumblings. Then thy days will be long in comfort of toil, thy fellows at peace with thee, and thy boss more satisfied.

Back to the soil! is all very well for the man who wants to do farming, but there's 57 varieties of "soil" in a power plant a clean man will back away back from.

Let's have a "hall of fame" for those men in our field whose names hit off their vocation, or are at variance with it—like W. G. Snow, an authority on heating and ventilation. Mr. Snow caps the peak of this lofty mountain of unknown talent. Enter your neighbor if he is eligible, and POWER readers will soon have a hall of fame that will make the Singer Building look like a bungalow.

"Prosperity is the lot of the coal trade," says a coal paper. The reader sending the most appropriate comment on this item, omitting profanity where possible, can have it published gratis.

A twice-told tale worth repeating: An owner in checking up his bill for the equipment just installed in his newly acquired power plant, complained that 120 pieces were still missing. "I've got all but the 120 volts you said would go with the generator," he wrote the manufacturers.

Hex Nutt says that there's lots in a name when it has a nutty flavor. Now he's afraid that owing to reports of an early winter the squirrels will get him.

At last Europe is afforded a chance to reduce friction and pour oil on the troubled waters. Nearly 142,000,000 gal. of lubricating and paraffin oil have been imported from the United States so far this year.

Smoke Abatement Meeting

That engineers are still interested in smoke abatement, popular opinion to the contrary notwithstanding, is shown by the number of meetings held by engineering bodies lately. Within a month there were three—one the meeting of the International Association for the Prevention of Smoke at Indianapolis. The other two were more local in character. In Philadelphia, Oct. 5, was a joint meeting of the members of the American Society of Mechanical Engineers in that city with the Engineers Club of Philadelphia.

Oct. 8, the New York membership of the American Society of Mechanical Engineers held a very enthusiastic public meeting, attended by some 300, in which various municipal and technical associations and clubs were invited to participate. The topic was introduced by George H. Perkins, head of the engineering department of the Lowell Textile School, who gave a report of the International Smoke Abatement Exhibition and Conference held in London last March.

From this outline of the work that is being done abroad, the attention of the meeting was turned to this country. Several who have been "on the firing line" in every sense of the word, in the battle to reduce the smoke nuisance in some of our Western cities gave interesting accounts of this work and what was accomplished. Among these were P. P. Bird, formerly smoke inspector of Chicago, and a written discussion from O. Monnet, now holding that position, was also read. The problem from the municipal standpoint was further discussed by O. P. Hood, chief mechanical engineer of the Bureau of Mines, Washington, D. C.; Lieut. James Reed, Jr., U. S. N., assistant director of Public Works, Philadelphia, and John M. Lukens, chief of the Bureau of Boiler Inspection, Philadelphia.

Prof. L. P. Breckenridge, Yale University, who, as head of the Engineering Experiment Station at the University of Illinois, did some signal work in the study of the smokeless combustion of soft coal, contributed a very pertinent discussion dealing not only with the problem as related to the city, but from its scientific side as well.

Others that spoke on the engineering phase, pointing out the causes of smoke production and means for preventing it were Albert A. Cary, George L. Prentiss, William Kent, Prof. D. S. Jacobus and A. A. Adler.

Written discussions had been offered by several who could not be present, but the lateness of the hour prevented their being read. These contributions will be digested in the Society's *Journal* later. Among their authors were Charles T. Main, Boston, Mass.; Horace C. Porter, Experiment Station, Bureau of Mines, Pittsburgh; Prof. J. M. Moyer, Pennsylvania State College; Perry Barker, Arthur

D. Little, Inc., Boston; H. V. Coes, Lockwood, Greene & Co., Chicago; Prof. C. H. Benjamin, Purdue University; E. C. Fisher, Wickes Boiler Co., Saginaw, Mich.; C. S. Davis, William B. Pierce Co., Buffalo, N. Y.; George Henderson, Baldwin Locomotive Works, Philadelphia; J. S. Shoemaker, Cumberland Mills, Maine; E. A. Uehling, Uehling Instrument Co., Passaic, N. J., and M. C. Huyette, Murphy Iron Works, Buffalo, N. Y.

Fatality at Worcester, Mass., Power Plant

On Oct. 9, a boiler tube in the power plant of the Worcester Consolidated Street Ry. blew out, killing Arnold S. Allen, the chief engineer, and injuring two firemen.

Nothing more is known of the accident, as the dispatch was received when the forms for this issue were being closed up.

Explosion on Destroyer "Walke"

An explosion on the destroyer "Walke," on Oct. 1, killed three men and injured six. The boat was making speed trials in a heavy sea off Brenton's Reef Lightship, near Newport, R. I., when, it is reported, her port turbine blew off its head casing and steam chest.

Perfect discipline was maintained, relief measures were instituted promptly, and the destroyer was able to reach port unassisted, running at half speed, driven by her starboard turbine, which was uninjured.

OBITUARY

William H. Corbin

On Sept. 25, William Horace Corbin, vice-president and director of the Joseph Dixon Crucible Co., Jersey City, N. J., died at his country home at the Hartwood Club, Sullivan County, N. Y., after an illness of several weeks.

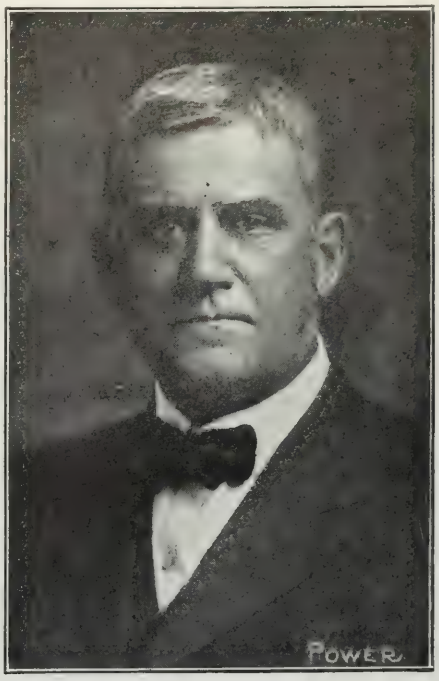
Mr. Corbin was born in McDonough, Chenango County, N. Y., July 12, 1851, the son of Eli L. Corbin and Abigail (Taintor) Corbin. He was educated at Cornell University and Columbia College Law School, from which he was graduated in 1872 and admitted to the New York bar. He came to Jersey City, and was admitted to the New Jersey bar in 1874, making his home in Elizabeth.

For several years, Mr. Corbin was the managing clerk of Senator Brinkerhoff, but in 1881 became a member of Collins & Corbin, which had been formed in 1875 by his brother Charles and Gilbert Collins. That partnership continued until the death of Charles L. Corbin in August, 1911, the firm continuing until W. H. Corbin's death.

Mr. Corbin occupied many positions of honor and responsibility in city and state affairs, helped revise the general corporation act of 1896, and was the author of an annotated edition of that law.

He was active in organizing several large corporations and was a member of the Union League Club of New York, the Union League Club of Hudson County, the Lawyers' Club and the Railroad Club of New York, the Town and Country Club of Elizabeth and the Carteret Club of Jersey City.

Socially, Mr. Corbin was a genial and pleasant companion and deeply religious. Mr. Corbin married Miss Clementine Kellogg, daughter of the late Elijah Kellogg, of Elizabeth. He is survived by his



William H. Corbin

widow and two sons, Clement K. Corbin, of the firm of Collins & Corbin, and Horace K. Corbin, an engineering contractor.

The Jersey City *Journal* said of Mr. Corbin: "In spite of the demands on his time made by good works, he was diligent in business and leaves a competence for his family, besides the heritage of a good name and the sympathy of the entire community."

SOCIETY NOTES

At a meeting of the board of governors of the American Society of Heating and Ventilating Engineers, Sept. 28, the following were appointed to coöperate with a committee of the National District Heating Association on educational matters in heating and ventilation: E. F. Capron, Chicago, Ill.; D. M. Quay, Cleveland, Ohio; W. F. Davendorf, Rochester, N. Y.; M. L. Foote, Columbus, Ohio, and W. C. Green, Cincinnati, Ohio.

The annual meeting of the American Society of Engineer Draftsmen, held in Teachers College, Columbia University, on Tuesday, Oct. 1, was the most important convention yet held by the organization and much was accomplished.

In his opening remarks the president displayed a unique chart showing that the society had quadrupled in the past year. Among other evidences of progress was cited the fact that members had secured, by voluntary subscription, the latest pattern stereopticon-projectograph for its monthly meetings and for preparing illustrated lectures to circulate among its branches. A construction program was adopted for the coming year, which included the establishment of a mutual benefit section, involving a form of insurance.

The following officers were elected for 1913: Prof. Charles William Weick, of Columbia University, president; William B. Harsel, first vice-president; Charles A. Clark, second vice-president; C. B. J. McManus, third vice-president; L. T. Maenner, fourth vice-president; E. F. Chandler, Henry L. Sloan and C. W. Fleming, board of governors. The enthusiasm shown by members the past year forms the very best augury for a still more rapid expansion in the year to come.

PERSONAL

J. L. Hopkins will manage the Pittsburgh branch recently opened in the Westinghouse Bldg. by the Durabla Manufacturing Co., New York City.

Louis M. Pawlett has resigned as executive engineer for the Locomobile Co. of America, to devote himself to practice as a specialist in consulting and development work in all departments of the field of motor-transportation engineering. His offices are at 1786 Broadway, New York City.

J. McA. Duncan has been appointed district manager of the Westinghouse Electric & Manufacturing Co. for the Pittsburgh district in place of W. F. Fowler, who has resigned to accept a position with the W. S. Kuhn Corporation. Mr. Duncan has been in the employ of the Westinghouse Electric & Manufacturing Co. for about 25 years, and is one of the original group of eight men taken from the Union Switch & Signal Co. to form the electric company.

Soft metals, such as britannia-metal or antimonial-lead, should not be cleaned in the same kettle as that used for removing the grease from brass goods. Tin and lead, contained in the soft metals, are dissolved by the potash or soda in the alkali, and the brass will then become coated. Separate kettles should be used.—*Brass World.*

Moments with the Ad. Editor

Lamoni, Iowa, and Utica, N. Y., are two widely separated places. If we were back in the good old stage coach days, a journey from one of them to the other would be an undertaking too formidable for any ordinary man.

One community might be wiped off the map by an earthquake and it would be months before the other would know about it.

Nowadays, thanks to the railroad, the telegraph and the newspaper, these places are, figuratively speaking, next door neighbors.

Each gets in touch with the other and gets the other's viewpoint.

Two engineers, for instance, through Power, are getting the self same advanced information on power plant practice.

They get the same view of advertising, too; if you doubt that, read their letters—the first is from Roscoe Willey of Lamoni:

Your request of a few weeks ago for letters telling what we bought through the selling section of Power has interested me in that it gives me satisfaction to tell the public what I've found to be good goods.

I take Power as much for the ads as for the help it gives on plant operation. When it arrives I start at the front cover and go straight through before stopping, as I am anxious to know what has happened in the other fellows' experiences and what new device has come out.

When the supply salesman comes around I always know what I want and I tell him if his house does not carry it, I prefer to order direct from the maker rather than to try "something just as good."

Give me a manufacturer that's not afraid to make a noise about his product and offers you a free working sample or to install apparatus on trial for approval and he's the man for me.

I have, through samples, obtained a certainty that could be obtained no other way.

I want to head my list with the "Philadelphia Compressed Air Grease Cup." That's the best cup made.

McClave Shaking Grate, Albany Grease, Palmetto Packing, Jenkins Bros. Globe Valves, American Gauges, Moncreiff Gauge Glasses, Dean Boiler Tube Cleaner, two Murray Corliss Engines, Dixon's Flake Graphite, Kewanee Unions, I. C. S. Reference Books, McGraw-Hill books and Smooth-on Cement were called to our attention through Power.

We are to be in the market for mechanical stokers and a new boiler before long, and you bet we are awake studying that class of goods.

And Wallace Johnson of Utica differs from his Iowa brother only in the details of what his concern has bought through Power:

From my point of view the advertising pages in Power are of as much value to the engineer as any part for it is there we find what new and up to date things are to be had.

I think most all of us can remember when we have gone to a lot of trouble and expense to make something to get along with and then found out that the same thing much better could be bought had we known about it.

I am now using a Bury Air Compressor that I saw described in Power and it was just the very thing needed to fill the place, and I looked back a long way to find that particular issue of the paper.

I have also an American-Thompson Indicator from the American Steam Gauge & Valve Mfg. Co., and two reducing wheels; two Manzell Cylinder pumps; one Lagonda Tube Cleaner; Garlock packing everywhere. And last, but not the least by any means, a Westinghouse Leblanc Condenser, now taking care of about 1600 to 1800 h.p. I first saw this condenser and air pump advertised in Power and it looked good to me. All these I read about and now have and every one great and small is giving splendid satisfaction.

* * *

Yes, it pays to read Power advertising and buy goods advertised in Power.

POWER

Vol. 36 NEW YORK, OCTOBER 22, 1912 No. 17

Who Is Responsible?

REPORTS are received every few days of boiler tubes bursting, blowing open the furnace doors and severely if not fatally injuring one or more firemen. Why endanger life from this cause when a simple safety appliance could be so easily attached to the furnace front?

Germany has enforced a law prohibiting placing boilers fitted with steam drains beneath rooms occupied by workmen. There is also a law demanding that all boiler furnaces must be equipped with some type of door that will resist the force exerted against them when a tube bursts.

Good for Germany! But what is America going to do about this avoidable danger?

Operation of Wisconsin's Capitol Plant

In these days of active competition between the central station and the isolated plant, it is essential to keep an accurate system of records so as to know exactly the cost of power, and to compare present with past performance. An annual report insures that this comparison will be made and is as instructive to the engineer as it is valuable to the owner. Perhaps at no other plant in the country are the records more complete in every detail than at the plant furnishing heat, light, power and water for Wisconsin's new capitol building and grounds. Every year a complete report is made by John C. White, chief operating engineer, to the superintendent of public property, and, for 1911, that part referring to operation and apportionment of operating and fixed costs may be of general interest.

Equipment of plant, operating records and costs, and apportionment of initial investment and operating and fixed costs between the various services of the plant.

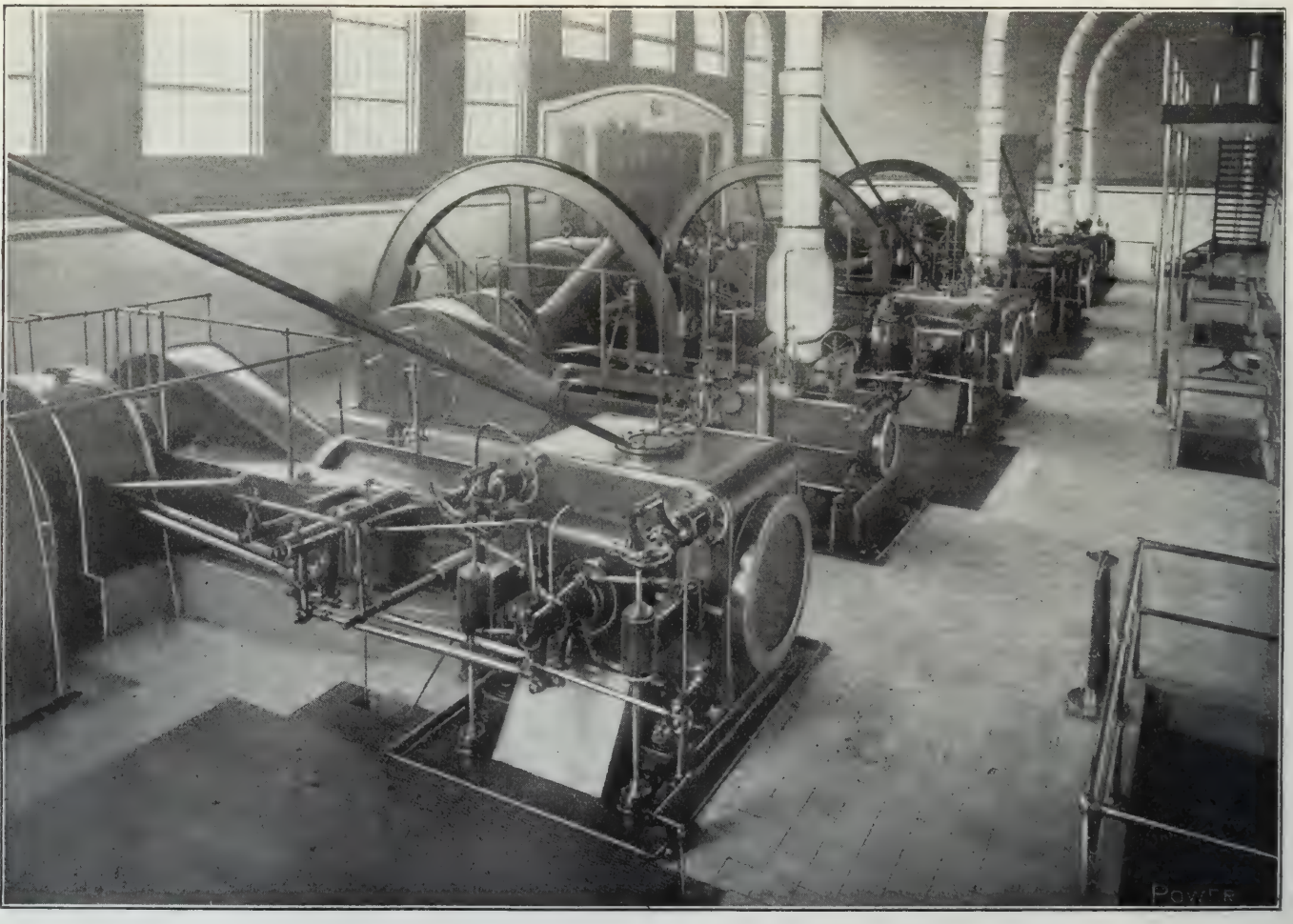


FIG. 1. ENGINE ROOM OF WISCONSIN'S CAPITOL PLANT

PLANT EQUIPMENT

The building is a one-story structure with basement in which the condensers and other steam auxiliaries are located. The engine room is approximately 40x98 ft. and the boiler room, 82x95 ft. In the latter are four Stirling water-tube boilers rated at 400 hp., and space has been reserved for four more similar units when required. Murphy smokeless furnaces, having a grate area of 80 sq.ft. each, serve the boilers. The chimney is 10 ft. in internal diameter, 250 ft. high and of radial brick. The draft is controlled by a Spencer damper regulator operating a damper in the main breeching according to the steam pressure.

Coal is first weighed on a Howe railroad-track scales and, after passing through a crusher, is delivered by a conveying system to any one of eight bunkers, each having a capacity of 200 tons. From the bunkers it is fed by gravity through Avery automatic scales and protected chutes to the magazines at the furnaces. Ashes are handled by the same conveyors used for coal, but in the reverse direction.

Feed water from the general-service mains or from the hotwell is supplied through a feed-water heater to the boilers. The feed is controlled by Liberty regulators and overpressure on the line is prevented by governors on the pumps, of which there are two. For general service there are two duplex pumps, each having a capacity of 500,000 gal. per day of 24 hr., and controlled by Fisher governors. For fire protection there are two motor-driven three-stage centrifugal pumps each of 2,500,000 gal. a day capacity against a total head of 357 ft. when running at 1200 r.p.m. A steam-driven two-stage air compressor completes the pump-room equipment.

The engine-room equipment consists of three Nordberg cross-compound condensing Corliss engines, with cylinders 14 and 28 by 36 in., operating at 120 r.p.m., each served by a Nordberg crank

and flywheel jet condenser, and one twin noncondensing engine with 14x36-in. cylinders operating at 120 r.p.m. To each of the four engines is directly connected a 250-kw. generator, making the total normal output 1000 kw. Also in the engine room are two balancer sets, each having a capacity of 12½ kw., and the main switchboard with the usual equipment of instruments and switches.

The keeping of accurate station records has been simplified, as far as possible, by installing the following curve drawing and recording devices: Four venturi meters with indicating dial and chart recorders, one on each discharge pipe from the boiler-feed pumps, one on the 3-in. high-pressure steam line in the tunnel and one on the 12-in. low-pressure steam line in the tunnel; one curve-drawing ammeter on the main busbars, recording pressure gages for steam and water, Avery automatic scales for coal, indicating thermometers for all important temperature records except flue gases, indicating draft gages, integrating wattmeters on generator leads and indicating volt and ammeters. The station is also provided with a mercury column for checking up vacuum gages, a dead-weight gage tester for pressure gages and a pair of indicators for engine and pump use.

FIG. 2. FRONT EXTERIOR OF PLANT

The first boiler was fired on Mar. 16, 1910, and by the first of 1911, the plant was in regular commission and operated under the following conditions: Steam pressure at boilers, 140 lb.; steam pressure on heating main, 7 to 8 lb.; water-pressure domestic service, 90 to 100 lb.; water-pressure fire service, 150 lb.; voltage, power circuits, 230; voltage, lighting circuits, 115.

During the heating season, the engines are run simple, noncondensing, and the exhaust steam is used for heating as far as it will go. There is use for practically all of it from Oct. 1 to June 1. The accompanying tables and curves show in detail the distribution of power, heat and the relations of the various services to each other and to the weather conditions.

Fig. 5 is a typical load curve showing the variations for the 24 hr. ending at midnight, Dec. 22, 1911. The peak in the morning occurs between 10 and 11, instead of between 8 and 10 o'clock. Fig. 6 shows the monthly output in kilowatt-hours for the year 1911. The large outputs for March, April and May were due to the night sessions of the legislature. Fig. 7 is a temperature-coal curve for 1911. The fan system was not in service in the east wing during the first two or three months of the year. Fig. 8 is a temperature-coal curve for January, 1912. It shows with fair accuracy the relation of coal consumption to temperature for a building served from an independent plant. The service requires furnishing steam to the capitol for various uses throughout the entire year, and this with that required for operating the generating units necessitates operating the plant continuously whether any heating is done or not. The minimum load in the summer is not over 200 boiler horsepower, about half rating for one boiler, and it has not exceeded 300 boiler horsepower during the four warm months, June to September, inclusive. This establishes a minimum consumption of about 10,000 lb. of feed water for the plant during the nonheating period and running condensing. When heating begins and the exhaust from the engines is used for that purpose, the quantity is probably not increased and may be reduced somewhat, so that out of a total evaporation of 20,000 lb. per hr., as accounted for by the meters, from 10,000 to 12,000 lb. is probably delivered to the capitol.

Fig. 9 is a series of curves showing ash content, the heat values of the fuel and the rate of evaporation per pound of coal as fired and per pound of combustible. The feed water as recorded by the meters was discounted 10 per cent., and the coal used as recorded by the scales was increased 5 per cent. to cover any errors that might creep into the records. It may be noticed that the evaporation is higher in the warm months than in the winter. The average for the year is about 10 lb. per pound of combustible corresponding to an efficiency of boiler and grate of about 73 to 76 per cent.

Fig. 10 is a feed-water chart for the period from 6:30 a.m., Dec. 22, to 6:30

FIG. 3. EAST HALF OF BOILER ROOM

a.m., Dec. 23. The average temperature for the day as given by the weather report was 31 deg. This chart was made to show the effect on the fuel consumption of running simple and exhausting into the heating system as against running compound condensing and furnishing live steam for heating. The former method was in use when the record was started at 6:30 a.m., Dec. 22. At 9:30 a.m., a compound-condensing engine was started and it carried the load from that time until 6:30 p.m. It may be noticed that the feed-water rate increased at once from about 14,000 lb. per hr. to about 18,000 lb., and that it remained at that point until after the load was changed back to the other engine. About 500 lb. more coal per hour was required, or from 5 to 6 tons per day, although the same work was being done. The above data is a convincing argument in favor of the method under which the plant is being operated. Stated in terms of efficiency, about 12 per cent. of the fuel value is being used in the compound-condensing engine, mechanical energy alone being produced, while in the simple engine from 6 to 8 per cent. of the steam passed is converted into mechanical energy, leaving the remainder, 92 to 94 per cent., available for use in the heating system and thus securing an overall efficiency of practically 100 per cent. for this method.

FIG. 4. AVERY AUTOMATIC SCALES ABOVE BOILERS

TABLE 1. RECORDS FOR THE YEAR 1911

Month	Coal		Feed Water			Output		Pumpage Record	
	Monthly total, tons	Daily average tons	Monthly total lb.	Daily average	Per hour	Total kw.-hr.	Daily average	Total gallons	Daily average
January	491.05	15.85	9,880,000	318,700	13,250	74,070	2,390	1,059,000	34,200
February	459.80	16.40	8,793,000	314,000	13,900	71,280	2,545	681,000	24,300
March	512.95	16.50	9,529,000	307,400	12,800	84,720	2,730	1,176,000	38,000
April	461.20	15.35	9,187,000	306,233	12,750	73,840	2,460	891,000	29,800
May	417.45	13.45	8,809,000	284,000	11,840	71,640	2,310	1,572,000	50,700
June	398.40	13.25	8,663,000	289,000	12,040	71,980	2,400	1,487,000	49,500
July	330.90	10.94	7,162,000	231,000	9,620	60,000	1,940	617,000	20,000
August	310.40	10.00	6,698,000	216,000	9,000	62,600	2,020	835,000	27,000
September	344.15	11.45	7,891 000	263,000	10,950	58,580	1,955	1,499,000	50,000
October	442.55	14.25	9,326,000	300,900	12,500	70,100	2,260	1,195,000	36,000
November	545.75	18.20	9,929,000	368,000	15,320	61,050	2,040	2,348,000	77,200
December	562.30	18.14	10,864,000	374,000	15,590	64,780	2,090	1,914,000	61,700
Total	5276.90		106,721,000			824,640	2,262	15,274,000	41,533

TABLE 3.—APPORTIONMENT OF COSTS TO VARIOUS SERVICES

Item	Total cost	Steam	Electric	Other uses
Land	$30,100.00	$13,600.00	$4,400.00	$12,100.00
Tunnel	49,214.00	17,963.00	2,559.00	28,692.00
Tun. wiring	1,085.00	181.00	0.00	904.00
Chimney	16,675.00	16,675.00	0.00	0.00
Main bldgs.	123,503.00	75,003.00	38,500.00	0.00
Conv. bldg.	20,000.00	20,000.00	0.00	0.00
Boiler equipment	87,115.00	87,115.00	0.00	0.00
All piping	69,093.00	56,800,00	3,093.00	10,000.00
Gen. and trans.	76,922.00	0.00	76,922.00	0.00
Water works	28,160.00	0.00	0.00	28,160.00
Totals	$501,867.00	$286,537.00	$125,474.00	$89,856.00
Per cent	100.00	57.20	25,00	17.80
Interest and depreciation at 8%	$40,149.36	$22,922.96	$10,037.92	$7,188.48

In Table 1 are given the more important items of the monthly records for 1911 just as they came into the office. Coal weights are from the readings of the Avery scales, kilowatt-hours from the Thompson recording wattmeters and pumpage from the displacement of pumps with a 3 per cent. allowance for slip and leakage.

The coal received at the plant from April, when the new coal contract was put into force, to the end of the year amounted to 5212.95 tons. Corrected for moisture, when it proved to be above or below the specified amount, 7.95 per cent., the tonnage was 5190.69 and cost $13,307.79. On Jan. 1 there was enough coal on hand to run until Apr. 1, so that the quantity given may be taken as a year's supply. Deductions made for excessive moisture and other penalties inflicted amounted to $245.08, which was more than enough to cover all costs of analyses and tests.

TABLE 2.—OPERATING EXPENSES OF PLANT

Lubricants	
Internal cylinder oil, 49 gal. @ 75c.	$36.75
Internal cylinder oil, 496 gal. @ 43.4c.	215.27
External oil, 50 gal. @ 9c.	4.50
External oil, 202.1 gal. @ 20.4c.	41.23
Grease, 75 lb. @ 15c.	11.25
Total.	$309.00
Boiler Compounds	
Six barrels, 3180 lb. @ 8c.	$254.40
Waste and Rags	
No. 2 white waste, 4 bales, 459 lb. @ 10.5c.	$48.20
No. 1 white waste, 2 bales, 205 lb. @ 12.5c.	25.63
Washed rags, 100 lb. @ 4c.	4.00
Total.	$77.83
Unclassified	
Packings, gage glasses, paints, coke, graphite, soap, washing powder, brooms, mops, etc., estimated.	$100.00
Repairs	
Estimated 2 per cent. on $75,000 of operating equipment.	$1500.00
Labor	
Chief operating engineer.	$2500.00
Three assistant engineers @ $1200.	3600.00
Three firemen @ $900.	2700.00
One helper and steam fitter @ $900.	900.00
One janitor @ $840.	840.00
Six helpers @ $720.	4320.00
Total.	$14,860.00
Total operating expenses including coal.	$30,409.02

Operating expenses are given in Table 2.

APPORTIONMENT OF FIXED CHARGES

In the following analysis the various services are classified with reference, first, to their relative importance, and, second, to their possible competitive or noncompetitive obtainment. It is worthy of note that the furnishing of the noncompetitive services would have required

practically all of the investment, except that directly chargeable to electric service and spent for such equipment, had the latter been omitted. The services are classified as

Heating .....noncompetitive.
Tunnels for traffic as warehouse connection .....noncompetitive.
Water for fire protection..noncompetitive.
Water for general service...competitive.
Current for light and power..competitive.

In Table 3 the division was made under the various headings as set forth below:

Land—proportional to areas occupied by main buildings and accessories.

Tunnel—clear area charged to traffic as warehouse connection; remainder proportional to area occupied. There is no occasion for tunnels for either cables or pipes.

Chimney—all to steam; whether or not there were generators, a chimney is necessary.

Main buildings—proportional to cubic contents. Bunkers—considered double construction for that space.

Conveyor building—all to steam. Its only use is to handle coal.

Boiler equipment—all to steam.

Piping—proportional to estimated cost for each service.

Generators and transmission—all electric.

Waterworks—all charged to other uses.

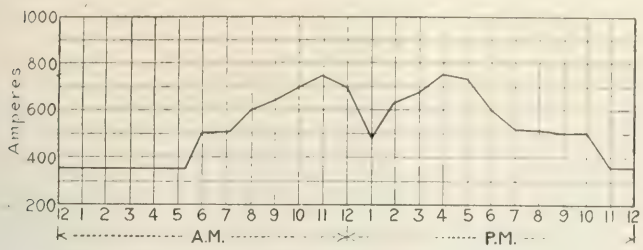


FIG. 5. TYPICAL DAILY LOAD CURVE, DECEMBER, 1911

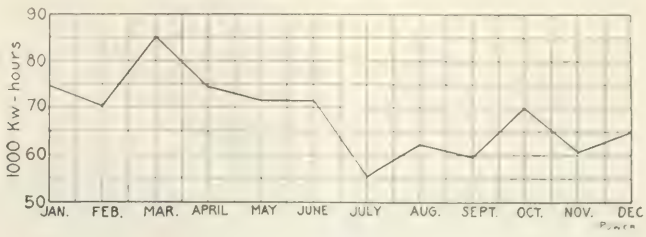


FIG. 6. MONTHLY OUTPUT IN KILOWATT-HOURS

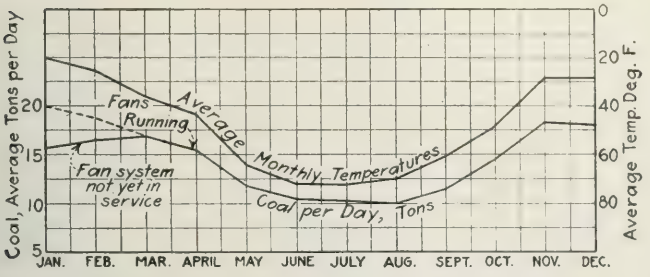

Coal, Average Tons per Day
Average Temp. Deg. F.
Fans Running
Average Monthly Temperatures
Fan system not yet in service
Coal per Day, Tons
JAN. FEB. MAR. APRIL MAY JUNE JULY AUG. SEPT. OCT. NOV. DEC.

FIG. 7. TEMPERATURE AND COAL CURVES FOR 1911

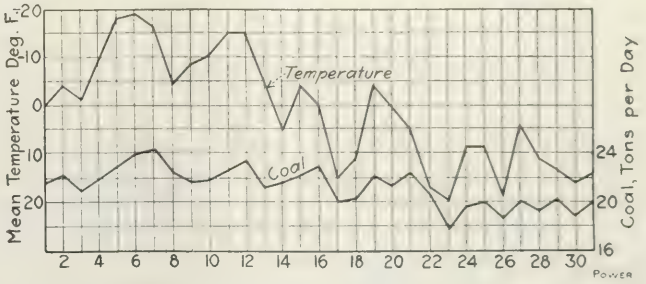


FIG. 8. TEMPERATURE AND COAL CURVES FOR JANUARY, 1912

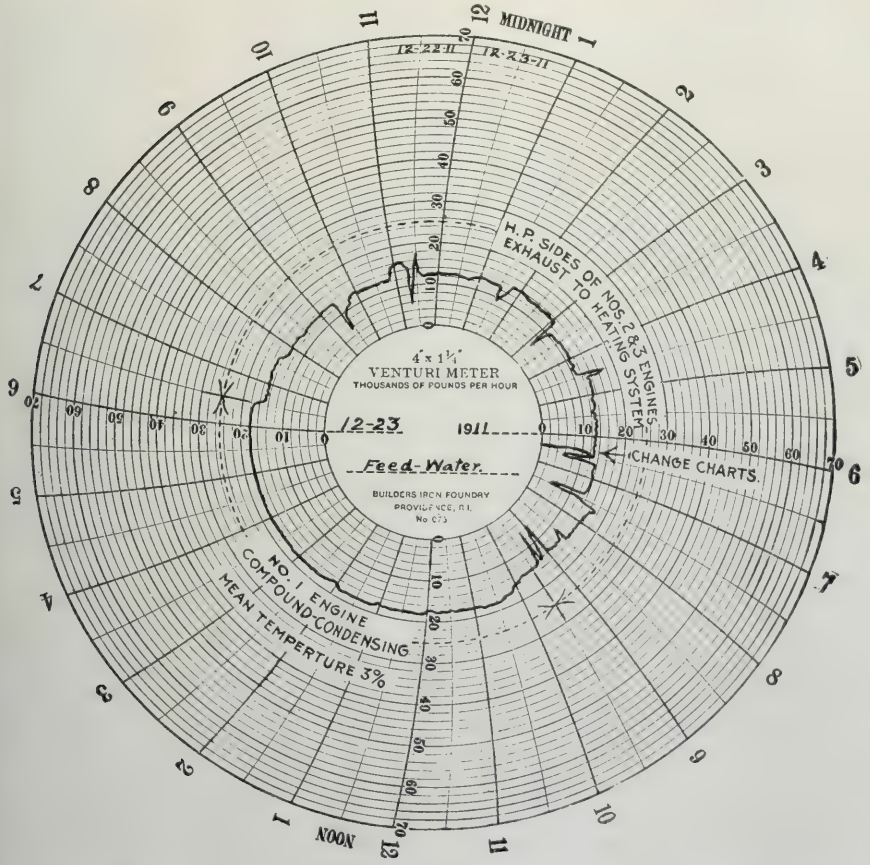


FIG. 10. FEED WATER CHART SHOWING ECONOMY OF USING EXHAUST FOR HEATING

APPORTIONMENT OF OPERATING EXPENSES

The total operating cost shown elsewhere to amount to $30,409.02 for the

TABLE 4. APPORTIONMENT OF OPERATING EXPENSES

Item	Total	Steam	Electric
Fuel	$13,307.79	$9,634.84	$3,672.95
Lubricants	309.00	109.00	200.00
Boiler compounds	254.00	200.00	54.00
Waste, rags	77.83	27.83	50.00
Unclassified	100.00	50.00	50.00
Repairs, estimated	1,500.00	1,000.00	500.00
Labor	14,860.00	12,000.00	2,860.00
Totals	$30,408.62	$23,021.67	$7,386.95
Per cent	100.00	75.70	24.30

TABLE 5.—GENERAL SUMMARY

Item	Steam	Electric	Other uses
Operating	$23,021.67	$7,386.95	0.00
Int. and dep.	22,922.96	10,037.92	$7,188.48
Totals	$45,944.63	$17,424.87	$7,188.48

TABLE 6.—UNIT COSTS

Item	Total cost	Total output	Unit cost
Electricity	$17,424.87	824,640 kw.-hr.	2.11c per kw.-hr.
Steam	$45,944.63	2,784,000 boiler hp.	1.65c per boiler hp. hr.
Cost per kw.-hr. exclusive of interest and depreciation			0.899c
Cost per boiler hp.-hr. exclusive of interest and depreciation			0.827c
Cost to evaporate 1000 lb. water exclusive of interest and depreciation			23.97c

year ending Dec. 31, 1911, is apportioned in Table 4 between steam and electric. No separation has been attempted as between steam and pumping operations and other uses. A rational division would be almost impossible to obtain and of no value. All of the pumps are steam-driven except those for tunnel drainage and fire pumps.

Table 5 gives a general summary of the apportionment and Table 6 unit costs.

As the plant is being operated at considerably less than half capacity for steam and at only about one-fourth capacity for current the overhead charges will be reduced greatly per unit of output as the plant takes on the load for which it was designed. With the completion of the capitol and the construction of the other buildings contemplated, a much better showing can be made.

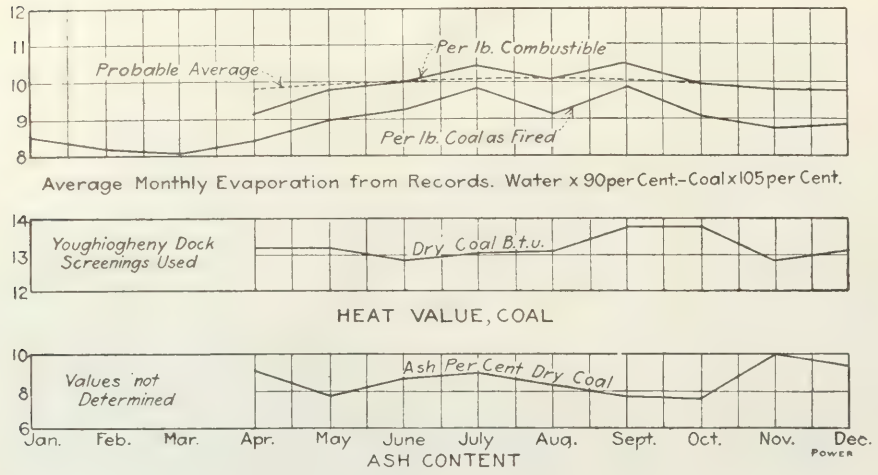

Probable Average
Per lb. Combustible
Per lb. Coal as Fired
Average Monthly Evaporation from Records. Water x 90 per Cent.-Coal x 105 per Cent.
Youghiogheny Dock Screenings Used
Dry Coal B.t.u.
HEAT VALUE, COAL
Values not Determined
Ash Per Cent Dry Coal
Jan. Feb. Mar. Apr. May June July Aug. Sept. Oct. Nov. Dec.
ASH CONTENT

FIG. 9. COAL VALUES AND EVAPORATION

Replaced Stokers with Shaking Grates

BY V. L. BOWLES

Although contrary to present-day practice and the cause of much criticism, the change from stoker equipment to hand firing has borne out the judgment of the writer to such extent in the plant he operates that the balance of the stoker equipment will be replaced by hand-fired furnaces.

The reasons for the change were high cost of upkeep, large amount of combustible in the ash, unreliability during long and hard periods of operation and the fact that no extra labor would be required.

The stoker referred to was one of the most modern of the toboggan-grate type. Due to the layout of the plant, proper coal-handling apparatus could not be installed with any degree of success; therefore all coal was weighed and handled by hand.

There are three boilers of the vertical water-tube type of 100, 250 and 350 hp. respectively, furnishing steam for a power plant and about 60,000 sq.ft. of direct radiation, all condensation being returned to the boilers during the winter with sufficient raw water to prevent pitting.

During severe winter weather the two larger boilers will handle the work without excessive overload, but there is no reserve in case of breakage to either stoker. This feature would, of course, be eliminated by a hand-fired installation; the quality of the feed water is such that the boilers could be operated for a long period, if necessary, without cleaning.

All three boilers are equipped with dutch-oven furnaces, the smaller one being hand-fired.

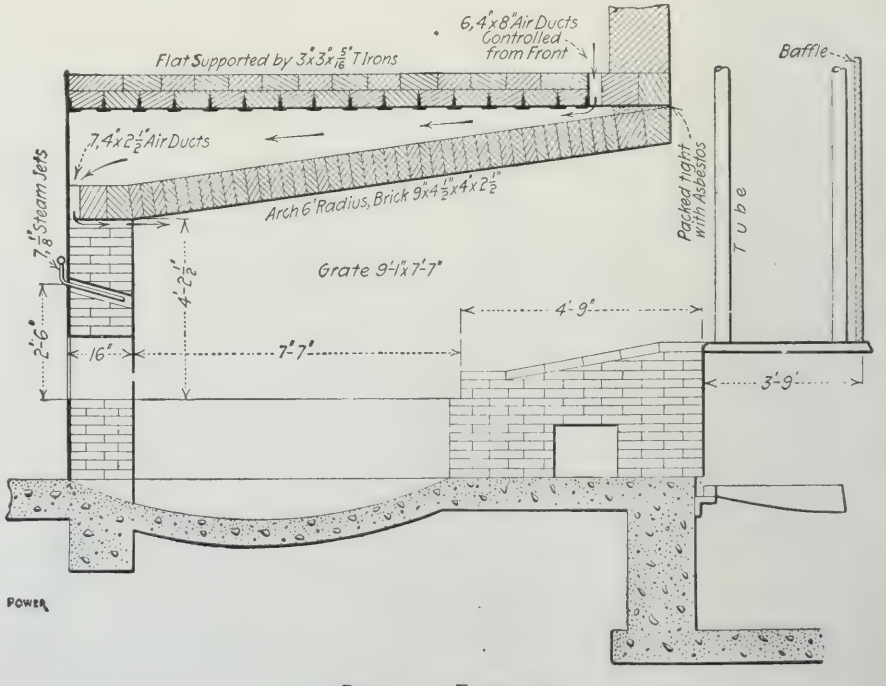
After long and close observation of the different methods of firing, and comparison of results relative to economy, reliability, upkeep and labor, the stoker on the largest boiler was discarded and shaking grates installed.

The accompanying longitudinal section of the new furnace, with principal dimensions, will show the general construction.

Steam jets were an advantage in maintaining a higher and more uniform temperature and eliminating smoke, which has been cut down 75 per cent. Although the new furnace has been in use but a comparatively short time, a saving of approximately 4 per cent. in coal of the same grade as formerly used with the stoker, with about the same load, a much cleaner ash, more uniform steam pressure and less labor for the fireman resulted. The last two statements may appear impossible, nevertheless they are true, due partly to local conditions.

There is no question but that the upkeep will be reduced at least 80 per cent. The accessibility is also much greater. Sudden overloads are handled with just as much ease and with less variation of steam pressure than formerly.

Three electric pyrometer couples are used for obtaining furnace temperatures, one in the first pass, 9 ft. above the drum, one in the second or downward pass, 9 ft. from the lower drum and one at the base of the stack.



DESIGN OF FURNACE

The temperature at the first couple will vary from 1400 to 1900 deg.; at the second, from 675 to 877 deg., and the stack temperature will vary from 550 to 675 deg. with a draft over the fire of from 0.2 to 0.35 in., depending on the load.

The CO_2 runs almost exactly the same as with the former stoker equipment.

It has been found that close attention to the temperatures at the three different points will net better resuls than equal attention to the stack gases, although nearly the same methods are employed in arriving at the best values. Different grades of coal will be tried out in the near future with no doubt still better results.

Burning Natural Gas under Boilers

By Leon B. Lent

Notes on the use of natural gas as fuel, together with data compiled from different sources, showing the results of actual tests.

In the matter of burning natural gas, the experience of several authorities shows agreement in several points, but there is a radical difference of opinion on others. Accurate information is somewhat meager and in most cases it is difficult to reduce results to a comparative basis.

Burners

All types of burners seem to show about the same efficiency. The majority are of the inside-mixing type; that is, the gas is delivered under a pressure of from 4 to 8 oz. of water (sometimes as high as 30) and in a direction somewhat tangential to the axis of the burner. This produces a helical swirl of the gas and the induced air and is conducive to a more thorough mixture. The proportions of gas and air should be under accurate control. Better results are obtained (as with oil fuel) by using several burners of smaller size than one or more burners of larger size.

Furnaces

According to J. Rowland Brown (*Transactions*, A. S. M. E., Vol. 27, page 188), the greatest difference in efficiency between two different boilers lies in the location of the brick checkerwork in the furnace, the best results being secured with this about 3 ft. beyond the burner. This checkerwork acts as an igniter for the gases with a second checkerwork 3 ft. further on to break up the flame passing through the first. Mr. Brown also contends that the overhead arch does not make as much difference in a gas furnace as in a coal furnace, the checkerwork being the important factor.

The same conditions are necessary for the efficient combination of gas as for any other fuel; namely, proper proportions of gas and air well mixed and burned under the highest possible furnace temperature. The latter means that the combustion chamber should be surrounded with refractory material capable of assuming and maintaining a sufficiently high temperature (in good practice a high state of incandescence).

The furnace should be of sufficient volume if possible to allow complete combustion before it is quenched by contact with the cooler heat-absorbing surface of the boiler proper.

While some tests show practically equivalent performances when gas is burned with a blue flame and with a white or straw-colored flame, it is generally conceded that the blue flame indicates more efficient combustion, a higher temperature and a superior performance, other things being equal.

Boiler Performance

Assuming natural gas to have a heat value of 1000 B.t.u. per cubic foot and a boiler horsepower to be equivalent to $34.5 \times 970.4 = 33478.8$ *B.t.u.* absorbed per hour, the gas consumption for various combined boiler and furnace

TABLE 1. EVAPORATION AND GAS CONSUMPTION AT DIFFERENT BOILER EFFICIENCIES

Combined boiler and furnace efficiency, per cent.	Cu.ft. of gas per b.hp.; heat value = 1000 B.t.u. per cu.ft. of gas $\left(\frac{34.5\times970.4\times100}{1000\times\text{Efficiency}}\right)$	Evaporation from and at 212 deg. per cu.ft. of gas $\left(\frac{1000\times\text{Efficiency}}{970.4\times100}\right)$	Evaporation from and at 212 deg. per lb. of gas with heat value of 19,510 B.t.u. per lb. of gas $\left(\frac{19,510\times\text{Efficiency}}{970.4\times100}\right)$
100	33.5	1.030	20.1
90	37.2	0.926	18.1
85	39.4	0.875	17.1
84	39.8	0.865	16.9
83	40.3	0.854	16.7
82	40 8	0.844	16.48
81	41.3	0.833	16.28
80	41.8	0.823	16.07
79	42.3	0.813	15.87
78	42.8	0.803	15.67
77	43.4	0.7925	15.47
76	44.0	0.782	15.27
75	44.6	0.772	15.07
74	45.2	0.762	14.87
73	45.8	0.752	14.67
72	46.4	0.741	14.47
71	47.1	0.731	14.27
70	47.8	0.721	14.06
69	48.5	0.711	13.86
68	49.3	0 700	13.68
67	49.9	0.690	13.47
66	50 7	0.679	13.26
65	51.5	0.669	13.06

TABLE 2. RESULTS OF BOILER TESTS WITH GAS FUEL

No. of test	Type of boiler	Steam pressure, pounds	Feed water, temp. deg. F.	Type and number of burners	Gas pressure at burner, ounces	Boiler rating, hp.	Boiler hp., developed	Cu.ft. of gas per boiler hp	Combined boiler and furnace efficiency	Test conducted by
1	Cook water tube	126.5	33	10 Gwynn	3.84	250	254.5	48.3	69.2	J. M. Witham
2	Cook water-tube	122	33.6	10 Kirkwood	4.5	250	253.7	48.0	69 7	J. M. Witham
3	6 Cook water-tube	112	172	60 Kirkwood	4.0	Each 250	247	41.0	81.6	J. M. Witham (with white flame)
4	6 Cook water-tube	88	178	60-6-in. Kirkwood	4.0	Each 250	213 1	41.0	8.16	J. M. Witham (with blue flame)
5	6 Cook water-tube	116	144	60–6–in. Kirkwood	6.0	250	296.6	45.2	74.0	J. M. Witham (white flame)
6	6 Cook water-tube	99	151	60 Kirkwood	6.0	250	271.1	42.1	79.0	J. M. Witham (blue flame)
7	6 Cook water-tube	128	157	42 Kirkwood	8.0	250	225.4	44 2	75.7	J. M. Witham (white flame)
8	6 Cook water-tube	128	184	46 Kirkwood	8.0	250	227.6	47 2	70.9	J. M. Witham (blue flame)
9	6 Cook water-tube	120.1	151.2	Kirkwood	6 9	250	273	44 9	74 6	J. M. Witham
10		132.7	185	Kirkwood	6.4	250	252	40.96	81.7	J. M. Witham
11	Heine water-tube	145.8	40.9	6 Kirkwood		200	154 9	48.6	68 8	G. I. Bouton
12	Heine water-tube	144.1	61.1	6 Kline		200	217.8	43 0	77.7	G. I. Bouton
13	Heine water-tube	149.6	74	6 Kirkwood		200	258.0	40.51	82 4	(Supplemental supply of preheared air used.) Bouton and Witham
14	Horizontal Cahall	95.0	46.5	6 Merrill	4.8	302	340.2	44 5	75.0	J. Rowland Brown
15	Horizontal Cahall	85.7	53.5	6 Merrill	7.0 to 30 0	302	260.0	37 4	89 2	J. Rowland Brown
16	Horizontal Cahall	87 2	54.3	12 Gwynn	16.0	302	532 8	44 2	75.7	J. Rowland Brown
17	Horizontal return tubular	90.0	176.5	Klein	2.21	100	55.0	48 8	68 7	E. G. Bailey
18	Horizontal return tubular	88.0	164.5	Klein	2.58	100	64 8	46.6	71.7	E. G. Bailey

efficiencies will be as shown in Table 1.

In a boiler fired with natural gas and having a properly designed furnace, everyday practice should show a combined efficiency of 70 per cent., according to one authority, and this seems quite reasonable. There should be no large amount of excess air, the combustion temperatures should be high, the furnace doors may remain closed and but little soot should be formed to deposit on the tubes.

The principal qualities derived from several boiler tests are given in Table 2. The figures given do not represent the best practice in all cases nor do all the figures admit of comparison on the same basis, since the boilers tested are different types, run under different conditions, and some with obvious inefficiency. The calorific value of the gas is not given for all the tests; in tests 1 to 10 inclusive, it was 1098 B.t.u. at 60 deg. F. and 29.92 in. barometer, and in tests 17 and 18 the value was 1199 B.t.u. at 32 deg. F. and 29.92 in. barometer.

Height of Chimney

The chimney height necessary to supply draft for natural gas is that necessary to overcome the frictional resistance of the gas passages from the furnace to the top of the chimney and seldom exceeds 0.20 in. of water unless an economizer is in the circuit. A stack 80 ft. high should meet any demand for intensity of draft, although local conditions may make it necessary to discharge the furnace gases at a much greater height. Dampers should be used in any case, for excessive draft will induce an excessive air supply and so decrease the efficiency.

Daly and Doolin Discuss Riveted Joints

By Danny Hogan

"Did ye see th' discussion goin' on be th' paaper," said Daly, "about th' pitch of rivets in b'iler seams? Sure 'tis a most insthructive set af articles, an' ivery b'iler maker should be radin' up on th' subject. But th' multichude of writers splashes a pint av good ink showin' th' importance av th' proper pitch an' how to find th' efficiency av th' seam, an' all that. Thin they sthop."

"Oi read thim," replied Doolin, "an' agree wid ye that they were hoighly insthructive."

"Engineers should pay more attintion t' b'iler design, Doolin, for th' funny part av it all is, thim high-brow fellers niver git anny further than analyzin' th' efficiency av th' seam."

"Oi admit 'tis important," said Daly, "but 'tis like speakin' only av th' pitcher in a baseball game an' forgettin' intoirely th' other eight min. A riveted seam is a chain wid siveral links, an' 'tis important t' know th' strength av each link, wan av which, be th' way, is th' straight pitch. And, moind ye, wan must choose a diameter for th' rivet hole for a certain thickness av plate an' not run away wid th' idea that anny diameter av rivet will do fer anny thickness av plate. Nayther should th' size av th' rivet head, th' height an' diameter at th' plate, an' th' shape be disraymimbered."

"Do ye think a button head, steeple, cone or conoidal head th' sthrongest?" asked Doolin, as he filled his "T. D."

"Th' heads must hold th' plates t'gither an' sthand up for calkin' th' seam," answered Daly, as he handed Doolin a brimstone match. "Always raymimber a rivet in shear is loike a bolt in tinsion, an' th' head has t' be considered, for all th' pitch writers neglicted this p'int.

"Thim chaps, Doolin, niver mintion whether th' rivet holes was punched full-

"Th' Funny Part Av It, Doolin, Is That Thim High-brow Fellers Niver Git Anny Further Than Analyzin' th' th' Efficiency av th' Seam"

soized, punched small an' reamed out in place, drilled with th' plates flat or drilled after rollin' th' plate up. Now all this constitoots a link in th' value ov a j'int whether th' ingineers think wid me or not, an' it makes th' difference in th' price of a b'iler as well. If ye are lookin' fer a cheap b'iler, just ignore th' question av makin' th' rivet holes, but if it's a good one ye want, have th' holes drilled out full-sized afther th' plate is rolled up, loike is done wid th' 16-ft. Scotch marines.

"Ye may know, Doolin, that English tests shows 18 per cent. loss in tinacity be reason ov th' punchin' ov ⅜-in. plates. Av coorse, th' higher in phosphorus th' metal, th' greater Oi believe would be th' loss. Consider, thin, this detail in findin' th' efficiency av th' rivet pitch. Say ye had a 70 per cint. joint wid holes punched full size, an' say th' mill test showed carbon 9.3 per cint., phosphorus 0.06 per cint., an' sulphur 0.05 per cint.—th' b'iler made wid a lap seam, an' 20 years old at that—what efficiency would ye allow?"

Here Doolin choked slightly as some of the sulphur fumes were inhaled. When he recovered his breath, Daly said: "Goin' further down th' pike, Doolin, let's look into th' mather av whether th' seam is a true arc av th' circle. Ye know th' usual flat spot on th' sheet is along th' longitudinal seam. Now tell me, is th' finished seam truly cylindrical or not? If not, thin flexure is prisint an' to be considered.

"There's siveral more points. Were th' rivets driven be hand, snapped by th' gun or be th' bull? Were they overheated an' carbonized in a coal fire or properly heated in a gas or oil furnace by a lad what was onto his job? Do they fill th' hole solidly from ind to ind or only at th' inds? If not, in th' latter case, th' shear diameter is much less an' that's a factor in yer analysis of th' p'int.

"Of course, Doolin, th' bull-drove rivet is th' best, anny an' all things considered, an' for why? Th' squeeze is exerted on sthraight loines an' th' prissure is therefore concintric an' not eccintric to th' rivet. The force exerted in drivin' is that powerful it is not nadeful to heat th' rivet as hot as whin workin' be other methods, an' this avoids burned rivets.

"Again, th' best b'iler shops are now usin' only $\frac{1}{32}$-in. clearance instid ov $\frac{1}{16}$ in., for th' rayson they can use a dull-

red rivet wid th' bull, when th' gun would need one near white hot, an' th' latter would, ov coorse, swell in diameter. It folers, ther'fore, that in th' first case th' hole is complately filled as there is less clearance to fill.

"Goin' back t' th' efficiency matter again, Doolin, let me obsarve that th' ink-slingers say nothin' av th' j'int failin' be crushin' in front av th' rivet at th' edge of th' plate. 'Tis just as much a matther t' be considered as that av th' sthraight pitch. They say nothin' nayther av th' back or distance bechune th' rows ov rivets. But at wan glance ye can see th' importance av analyzin' these features, as they are a link in th' chain an' an ilimint in th' question of safety.

"It's glad Oi am, Doolin, to note ye are readin' up on b'iler consthruction, an' Oi advance these few p'ints fer yer careful consideration, as th' politicians say. Whin ye land that job as examiner at th' city hall, if ye iver do, bear this dope in moind for questions t' put t' th' laads. Sure, they needs t' know all about th' subject as well as one part ov it.

"And now get onto yer job," concluded Daly, "fer Oi sint fer ye to replace some tubes in No. 4 b'iler an' Oi suppose ye'll be chargin' up th' time spint in discoorsin' av riveted seams."

"Only sthraight toime," answered Doolin, "fer 'tis daywork."

No-If Engine Stop

This engine stop depends upon gravity and a suspended weight which is released when the engine exceeds its safe speed limit.

Fig. 1 shows the speed-limit band *A* attached to the shaft. It is held in position on the blocks *B* by the bolt *C*. A steel spring 1¼ in. wide is held by these blocks, and the band and spring hold the adjusting blocks *E*. *F* is a weight fastened to the spring *D*. *G* is a steel lip contact. The spring is loose on the pin *H*, which prevents lateral motion.

Fig. 2 shows the location of the quick-closing valve. Fig. 3 shows the stop applied to a quick-closing valve. *A* is an adjustable arm, and centrifugal force, due to overspeed, causes the spring of the speed limit to elongate until it strikes the latch, releasing the vertical latch holding the rod *C*. This rod sustains the weight *F* suspended by a cable running through a ½-in. conduit *E* and over roller-bearing pulleys.

While the engine is running, the stop is operative. Should the stop go out of commission, the engine will stop, as the weight will close the throttle. To put the stop in commission, close the throttle and open the quick-closing valve by lifting the rod *C*. This opens the valve and relatches *B*. The engine cannot be run until this is done.

Another application of the stop is shown in Fig. 4. The amount of weight used to close the valve has no effect upon the latch *A*. The armature *B* must be pulled down to release the weight. When stopping is done from a distance, the armature is actuated by a magnet.

Fig. 5 shows a method of protecting an engine from running away when the American type of rope drive is used. Should the sheave crush, weight drop off, or the carriage work loose, the cable pulls the tripping latch and the throttle valve is closed before the rope can destroy the governor, thus saving the engine.

An electrical telltale device for applying to large belts and ropes is shown in Fig. 6. The parts are mounted on a slate base, the spring is held out of contact by a fine copper wire *B* stretched in front of the belt or rope. When the rope strands or belt tears, the wire is broken, contact is made at *C* and the alarm is continuous. Should the telltale be repaired with wire or string too strong to be broken by a rope strand, the spring *A* is pulled and makes contact at *D*, in which case the alarm is intermittent.

This engine stop is manufactured by the Metallic Packing & Manufacturing Co., Elyria, Ohio.

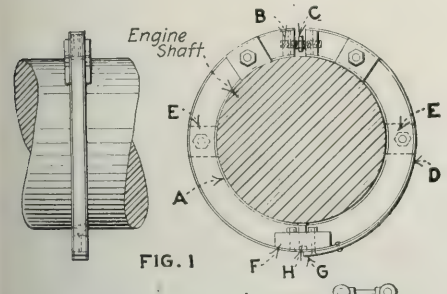


FIG. 1

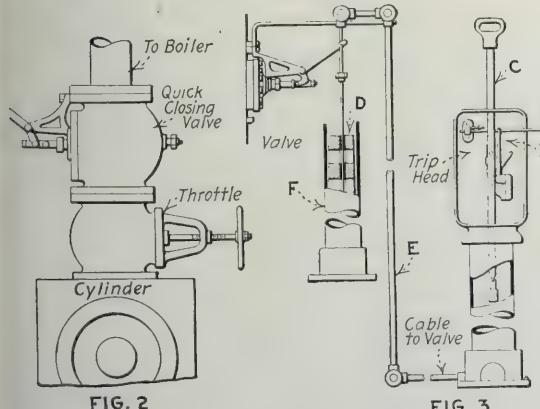


FIG. 2

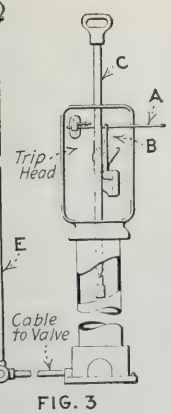


FIG. 3

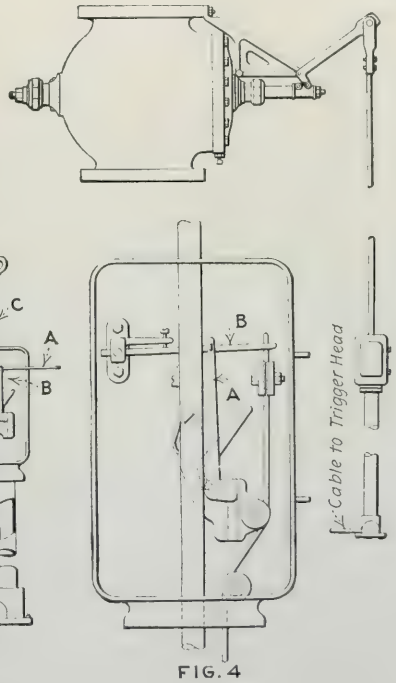


FIG. 4

DETAILS OF THE NO-IF ENGINE STOP

Bursting Tube in Water Tube Boiler

BY A. W. REENE

On Aug. 21 a tube burst in the lower row of a water-tube boiler in the plant of the Dixie Portland Cement Co., Richard City, Tenn., instantly killing a fireman who was enveloped by the terrific volume of steam and water which came out of the furnace doors.

This tube ruptured at a point just in front of the bridge-wall. The opening was about 13 in. long, and spread out almost flat at the center of the rupture, which gave two openings for the escape of the steam and water nearly the full size of the tube. The boiler pressure was 175 lb. at the time of the accident.

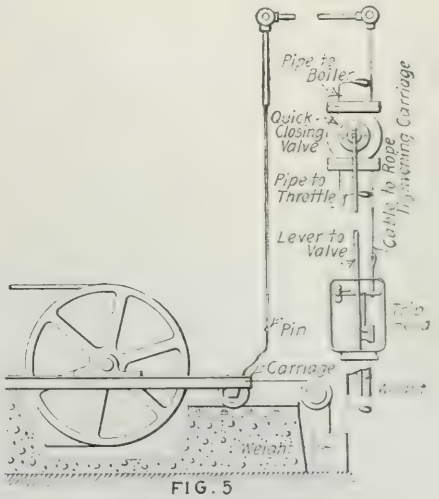


FIG. 5

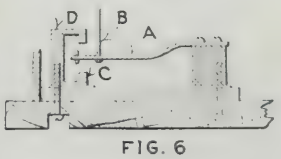


FIG. 6

An examination of the tube showed it had elongated 5/8 in. and was loose in both tube headers. A number of adjacent tubes were distorted, but were tight in the headers.

The tube was lap welded and of standard thickness. It split on the side opposite the weld. The boiler to which this accident occurred is fired with an underfeed stoker equipped with mechanical draft, the pressure being about 5 in. The fires were very heavy and the coal of a low grade. It is, therefore, quite possible that a hole was blown through the bed of fuel, allowing a jet of flame to impinge against the tube with such intensity as to overheat the tube and allow the pressure to do the rest.

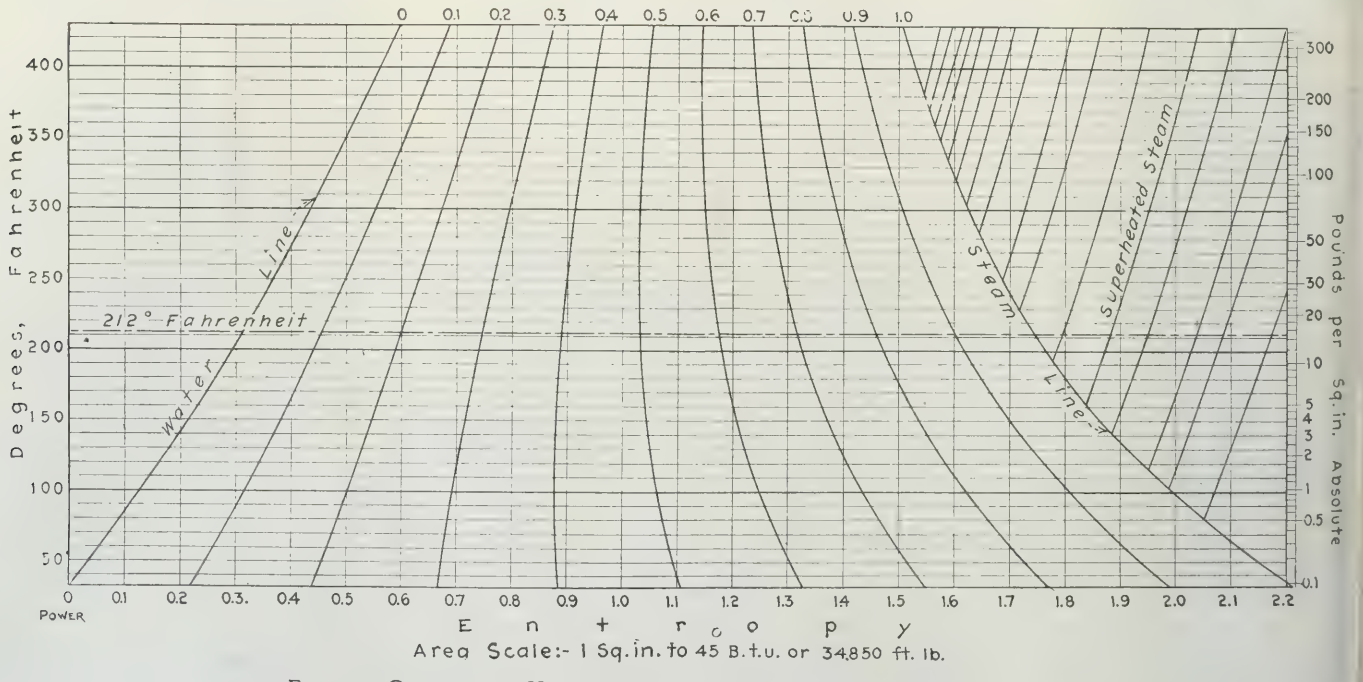
Simplicity of the Entropy Chart

By W. J. Crawford

The explanation of entropy ordinarily is perhaps rather confusing, but the use of the chart is simplicity itself. A number of steam problems are worked out with the chart. Try them and see how easy they are.

That the entropy-temperature chart is a wonderfully mysterious affair, requiring an immense amount of study to use it is a widespread idea among engineers and draftsmen. Those who have no trouble with the ordinary indicator diagram avoid the chart, because the textbooks, by their method of approaching the subject, have frightened them off. As a matter of fact, using the chart is simple and many have been exceedingly surprised when shown how easy it is to work out problems by its aid.

A general entropy-temperature chart is reproduced herewith and should be referred to, from time to time, as the explanation proceeds. For simplicity the volume and heat-unit lines found on the latest charts have been omitted. These lines are not necessary, but make using the chart more convenient. Fig. 1 represents the chart in its simplest form.



ENTROPY CHART WITH VOLUME AND B.T.U. LINES OMITTED FOR SIMPLICITY

The vertical line on the left indicates temperatures in Fahrenheit degrees and that on the right pressures in pounds per square inch, absolute, corresponding to these temperatures. The horizontal line represents entropy. The chart contains two main curves, *AB* and *CD*. The former is called the water line and *CD* the saturated steam line, or, more shortly, the steam line. The water line is nearly straight and the steam line is more curved. By tracing these two lines on the general diagram, it will be possible to work out many important examples.

EXAMPLE 1. A pound of water at 212 deg. F. is heated to 300 deg. F., and then converted into dry saturated steam at that temperature; it is then expanded adiabatically* to 212 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 2. Mark the point *A* on the water line at 212 deg. F., go up the water line to *B* opposite the 300 deg. F. mark. From *B* go horizontally across to the steam line at *C*. From *C* draw a vertical line downward to meet the horizontal line through *A*, at *D*. Then the dryness after the expansion is given by the ratio $\frac{AD}{AE}$. Measure *AD* and *AE* carefully with a rule reading to small decimal parts of an inch. Then dividing *AD* by *AE* will give 0.91. Therefore the percentage of dryness is 91, that is to say, of the one pound of steam originally dry and saturated at 300 deg. F., only 0.91 lb. is steam after the adiabatic expansion to 212 deg. F., and consequently 0.09 lb. is water.

The steam is dry and saturated at 300 deg. F., so it is necessary to go to the point *C* on the saturated steam line, before beginning the expansion. As the expansion is adiabatic, a vertical line must be drawn to represent it on the chart.

EXAMPLE 2. A pound of water at 160 deg. F. is heated to 300 deg. F.; it is then converted into steam 80 per cent. dry at that temperature, and then expanded adiabatically to 212 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 3. Mark the point *A* on the water line opposite 160 deg. F. Go up the water line to *B*, opposite 300 deg. F. Go along horizontally from *B*

*Adiabatic expansion is expansion of a fluid when no heat leaves it or is admitted to it during the expansion. This is approximately realized in the case of steam expanding in a well-lagged cylinder.

to C, which is $\frac{8}{10}$ the length of BK, the distance between the water and steam lines. The point C is easily found because sloping lines are drawn downward on the general chart, at each tenth between the water and steam lines. From C draw a vertical line downward to meet the horizontal line through E, drawn opposite the 212 deg. F. mark. EF is the length between the water and steam lines. Then the dryness fraction after expansion is found by dividing ED by EF. The answer will be $\frac{3}{4}$ or 0.75, and the dryness is therefore 75 per cent. By using the sloping lines labeled in tenths between the water and steam lines on the general chart the dryness could easily have been estimated after dropping down vertically to the 212 deg. line.

If the wetness after the expansion had been required, it could have been found by dividing DF by EF, which would give 0.25, or 25 per cent.

EXAMPLE 3. A pound of feed water at 175 deg. F. is heated and converted into steam at a pressure of 100 lb. per sq.in., absolute, or 85 lb. gage. The steam is wet. After it is formed it is expanded adiabatically to a pressure of 30 lb., absolute, or 15 lb. gage, when it is found to be 60 per cent. dry. Find the original dryness of the steam.

SOLUTION. See Fig. 4. Mark point A on the water line corresponding to 175 deg. F. Go up the water line to B, opposite the 100-lb. mark. Draw a horizontal line from B and opposite the 30-lb. mark draw a horizontal line DH between the water and steam lines. From D locate E at 0.6 of the length DH, to indicate a dryness of 60 per cent. after expansion. From E draw a vertical line upward to meet the top horizontal line at F. Then the original dryness of the steam formed at a pressure of 100 lb. is given by the ratio of BF to BK, which equals 0.59, or 59 per cent.

SUPERHEATED STEAM

To the right of the steam line on the general chart, sloping lines going upward may be noticed. These are the superheated steam lines.

EXAMPLE 4. One pound of steam is dry and saturated at 212 deg. F. It is then superheated to 300 deg. F., and then expanded adiabatically to 160 deg. F. Find its dryness after the expansion.

SOLUTION. See Fig. 5. Mark point A on the steam line corresponding to 1 lb. of steam dry and saturated at 212 deg. F. Go up the superheated-steam line to B, opposite 300 deg. F. From B draw a vertical line downward to meet the horizontal line drawn between the water and steam lines opposite 160 deg. F., at C. Then the dryness after the expansion is $DC \div DE = 0.98 = 98$ per cent.

EXAMPLE 5. One pound of dry saturated steam at 212 deg. F. is superheated and then expanded adiabatically to 160 deg. F. After the expansion it is again dry and saturated. How many degrees of superheat have been given to it?

SOLUTION. See Fig. 6. Mark A on the steam line corresponding to 1 lb. of steam dry and saturated at 212 deg. F. Mark B on the steam line corresponding to 1 lb. of steam dry and saturated at 160 deg. F. From B draw a vertical line upward, meeting at C the superheated steam line from A. Read the temperature opposite C. Then this temperature above 212 deg. F. is the number of degrees of superheat given the steam, or

$$335 - 212 = 123 \text{ deg. F.}$$

EXAMPLE 6. One pound of dry saturated steam at 212 deg. F. is superheated to 390 deg. F. It is then expanded adiabatically to 300 deg. F. Find the number of degrees of superheat it has after the expansion.

SOLUTION. See Fig. 7. Mark A corresponding to 1 lb. of dry saturated steam at 212 deg. F. Go up the superheated-steam line to B, opposite 390 deg. F. Draw a vertical line downward to C, opposite 300 deg. F. Through C draw a superheated-steam line CD (if there is such a line passing through C, use it; if not, draw a superheated-steam line as nearly parallel as possible to the superheated lines on either side of the point). Then D is the point on the steam line where the superheated-steam line C meets it. Read the temperature corresponding to D. The amount of superheat after the expansion is 300 deg. F., minus this temperature. Thus

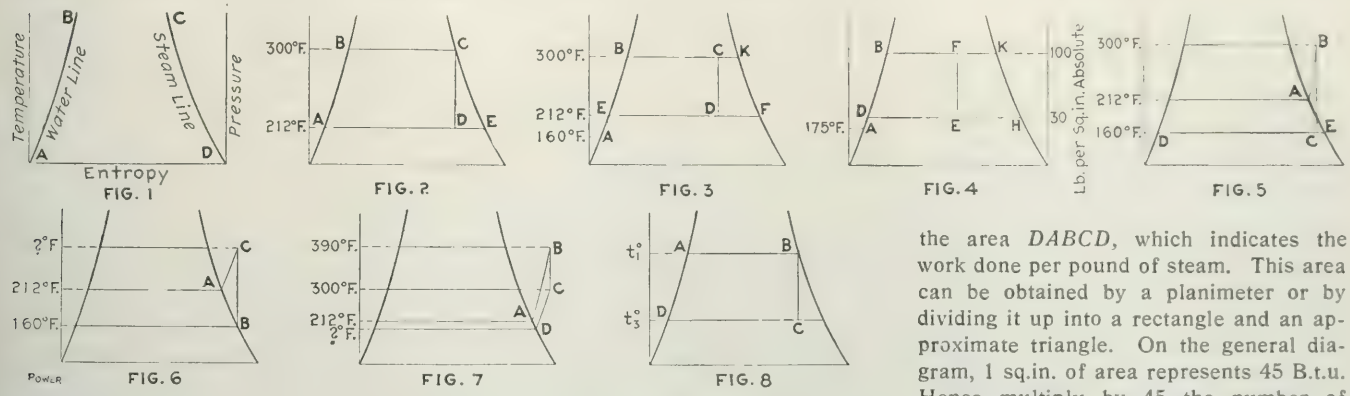
$$300 - 187 = 113 \text{ deg.}$$

of superheat.

The area of the entropy-temperature diagram, like the indicator diagram, represents work; but on the former, the work is obtained in British thermal units and not in foot-pounds, as with the indicator diagram. As an example of area, the Rankine cycle may be considered. This is the ideally perfect cycle with which the cycles of ordinary engines are compared. The Rankine cycle is as follows: Water is heated from a temperature t_2 to a temperature t_1; it is then converted into steam at t_1 and expanded adiabatically to t_2. At the temperature t_2, it is condensed and returned as water to the boiler.

EXAMPLE 7. Show the Rankine cycle between temperature t_2 and t_1. What is the work done per pound of steam?

SOLUTION. In Fig. 8, opposite temperature t_2, mark D on the water line. Go up the water line to A, opposite temperature t_1. Proceed horizontally to the steam line at B. From B drop a vertical line to meet the horizontal line, drawn opposite t_2, at C. Then the Rankine cycle is given by



the area $DABCD$, which indicates the work done per pound of steam. This area can be obtained by a planimeter or by dividing it up into a rectangle and an approximate triangle. On the general diagram, 1 sq.in. of area represents 45 B.t.u. Hence multiply by 45 the number of square inches obtained and the result will give the work done on the Rankine cycle in British thermal units.

Enough examples have been given to show that no one of ordinary intelligence need have difficulty in working out various thermal relations on the chart. After becoming familiar with the simple chart given, it is an easy step to the later charts, which include in addition heat units and volume lines and use absolute temperatures. Instead of measuring with a rule to obtain the ratios or using a planimeter for the areas, the answer to any of the problems given may be read directly from the chart. It is, therefore, more convenient to use, although at first glance rather confusing.

In discussing the utilization of waste heat from gas engines, K. Kutzbach, in *Stahl und Eisen*, emphasizes that it is impracticable to attempt to allow the jacket water to exceed, say, 205 deg. F., owing to the danger of preignition, and that the exhaust gases have a large volume with comparatively low heat content besides corrosive impurities. Utilization may be wise, but it is limited.

Mixed Pressure Turbine Installation

The figures and data presented below are from a typical case in which the application of a low-pressure turbo-electric unit has successfully met the demand for increased power in an industrial establishment without involving the throwing out of good and expensive steam plant, or unduly encroaching upon the space at command.

The adaptation of the purely low-pressure turbine to work in conjunction with an existing steam plant where the additional power required is between 40 and 60 per cent. of the normal rating of the reciprocating plant, and the mixed-pressure type when the increased power required exceeds this amount, effects gains in economy which naturally vary with the conditions obtaining; but there are few applications in which the saving will not discount the initial expense in the course of three or four years.

The application at present under review is that of a large flour mill in the north of England, the size of which has gradually increased so that an addition to the existing plant was rendered necessary about three years ago.

The existing plant comprised five 8x30-ft., and two 8½x30-ft. two-flue Lancashire boilers fitted with automatic sprinkling stokers, forced draft, superheaters, and economizers. These boilers supplied steam at 120 lb. pressure and 150 deg. superheat to three reciprocating steam engines: "No. 1," a 22 and 38 by 36-in. vertical compound Corliss; "No. 3," a 19 and 35 by 60-in. tandem-compound Corliss; "No. 4," a 16 and 28 by 60-in. tandem-compound Corliss. These three engines, all running condensing, drove the mill by means of ropes.

When it became necessary to increase the plant capacity, conditions rendered it advisable to install electric drives for the new machinery. A generator was therefore necessary and for its drive the consulting engineers had two options.

One was to install a high-pressure steam engine and more boilers; the other to provide a mixed-pressure turbine to work in conjunction with the existing reciprocating equipment. The choice fell upon the latter and the following were calculated as the new duties and performances of the engines already installed:

By John S. Leese

When it became necessary to increase the capacity of the power plant of a large English flour mill it was decided to install a mixed-pressure turbine operating on exhaust steam from the reciprocating engines already in use.

The conditions which prompted this decision are described and reports given of the operating results, which exceeded the guaranteed performance.

Since there is no equivalent to an indicated horsepower in a turbine, for purposes of comparision, 1 i.hp. on the engines is taken as being equal to 1 b.hp. at the turbine shaft, although in reality 1 b.hp. equals approximately 1.08 i.hp.

With the above figures in hand, two proposals were considered. The first was to run as follows:

No. 1 engine condensing giving approximately	500 hp.
No. 3 engine non-condensing giving approximately	600 hp.
No. 4 engine non-condensing giving approximately	450 hp.
and the turbine condensing giving approximately	850 hp. (582 kw.)
and a total of	2400 hp.

On this basis, the turbine maker guaranteed an overall steam consumption of the plant not to exceed 12.4 lb. per indicated horsepower per hour.

The alternative scheme was to operate as follows:

No. 1 engine non-condensing giving approximately	500 hp.
No. 3 engine non-condensing giving approximately	600 hp.
No. 4 engine non-condensing giving approximately	450 hp.
and the turbine condensing giving approximately	1150 hp. (790 kw.)
and a total of	2700 hp.

A guarantee of 11.3 lb. per indicated horsepower per hour overall consumption was promised for this arrangement.

The first proposal, with the higher consumption guarantee, was chosen, since the turbine maker's guarantee did not cover liability in the case of failure of the reciprocating sets to run thus constantly heavily loaded and noncondensing.

A. Steam consumption when running on exhaust steam only, with an initial pressure at the turbine steam chest of 16 lb. absolute, unity dryness, and a 28-in. vacuum (30 in. barometer).

Load	890-kw. max. on L.P. end	Max. with 22,000 lb. 585 kw.	500 kw.	300 kw.
Consumption in lb. per kw.-hr.	32.1	37.7	40.6	51.7

B. Steam consumption with high-pressure steam only, with an initial pressure of 180 lb. per sq.in. absolute, superheated to 500 deg. F., and with a 28-in. vacuum.

Load	1650 kw. (150% normal)	1375 kw. (125% normal)	1100 kw. (normal)	825 kw. (75% normal)	550 kw. (50% normal)
Consumption lb. per kw.-hr.	16.3	15.8	15.3	16.4	17.7

C. Steam consumption when running as a mixed-pressure turbine on 22,000 lb. of exhaust steam per hour.

Load	1100 kw. (normal)	825 kw. (75% normal)
Live steam required to make up power, in lb. per hour.	6500	4100

Engine No.	Same cut-off Pressure 120 lb. Superheat 150 deg. I. hp. Condensing	Same cut-off Pressure 120 lb. Superheat 150 deg. I. hp. Noncondensing.	Total dry steam in L. P. diagram lb. per hour	Total steam = dry steam + 15% moisture lb. per hour	Consumption lb. per i.hp. per hour. Condensing	Consumption lb. per i.hp. per hour. Noncondensing	Exhaust steam available per i.hp. per hour
1	656	454	7,390	8,500	13.0	18.7	15.9
3	911	685	12,320	14,200	15.6	20.7	17.6
4	527	368	6,130	7,050	13.4	19.1	16.2
Total	2094	1507		29,750	average		
					14.2	19.7	

In placing the order for the mixed-pressure turbine, the following figures were specified:

The turbine was supplied with a multiple-jet condenser, the turbine itself being of the impulse type with a velocity wheel at its high-pressure end.

The accompanying table gives the results of recent tests upon the plant, the guarantee figures tabulated above being placed in an adjacent column for easy reference.

(This table is given on the next page.)

From these results it will be seen that the steam consumption when corrected for superheat is but 0.03 lb. per indicated horsepower per hour in excess of the lower guarantee contained in the second proposal.

Remarkable similarity is shown between the tested engine horsepowers, as

given in the first table, and the last-test results, the respective figures being:

For No. 1 engine condensing 656 and 653 hp.
For No. 3 engine non-condensing 685 and 684 hp.
For No. 4 engine non-condensing 368 and 372 hp.

Since the installation of the turbine the mill owners have been able to cut out one of the seven boilers. The increased capacity of the milling plant, the facility with which the grain can be handled in the processes, the extreme cleanliness of the factory, and the reduced line shafting and transmission losses are also advantages which have been made very apparent since installing the electrical equipment.

	Test	Guarantee
Average steam pressure, lb. per sq.in. gage	123	120
" " temperature, deg. F.	473	500
Back-pressure on engines, lb. per sq.in. absolute	17	17
Low-pressure steam at turbine steam chest	16 17	16
Engine No. 1.		
Indicated horsepower condensing	653	500
Steam consumption, lb. per hour	8,500	6,500
Speed, r.p.m.	79	79
Engine No. 3.		
Indicated horsepower non-condensing	684	600
Steam consumption, lb. per hour	11,500	12,400
Speed, r.p.m.	72.7	71
Engine No. 4.		
Indicated horsepower non-condensing	372	450
Steam consumption, lb. per hour	6,600	8,600
Speed, r.p.m.	70 5	70
Turbine.		
Brake horsepower	735	850
High-pressure steam consumption, lb. per hour	1,750	
Low-pressure steam consumption, allowing 15 per cent. for moisture, lb. per hour	15,400	17,850
Three Engines and Turbine.		
Total output, b.hp. and i.hp.	2,444	2,400
Total high-pressure steam consumption, lb. per hour	28,350	
Test Results.		
Overall steam consumption of plant, lb. per i.hp.-hour	11 59	12 4
Correction for 27-deg. superheat, $27 \div 12 = 2\frac{1}{4}$-per cent.	0.26	
Corrected consumption, lb. per i.hp.-hour	11.33	
Difference between test and guaranteed consumptions, lb.	1 07	
Per cent. improvement on guarantee	8 5	

Load	150% 1650-kw.	100% 1100-kw.	600-kw.	50% 550-kw.
Power factor 0.8	96	95½	93	93
" " 1.0	96½	96	94	94

The mixed-pressure turbine is governed by a positive-acting centrifugal governor which, through pressure, irrespective of speed, automatically controls the admission of live steam to the turbine, if the supply of exhaust steam should fall short of the load demand. The governor also prevents excessive speed variations when the load is suddenly thrown on or off the turbine, the momentary figure being 6 per cent. and the permanent figure 3½ per cent. of the normal speed when the load is suddenly applied.

An oil separator in the exhaust lines of the engines in front of the turbine inlet allows the condenser water to be used for boiler feeding.

The multiple-jet condenser has a rotary valveless dry-vacuum pump and a centrifugal extraction pump, both driven by a single 55-hp. motor mounted on the same bedplate. The condensing water used per hour at full load is 1900 gal., and its temperature is 65 deg. F.

The turbo-generator is of the pipe-ventilated type, rated at 1100 kw. at 2400 r.p.m., and generates three-phase current at 500 volts and 40 cycles. The shunt-wound exciter, which has a radial commutator, is directly connected and has an outboard bearing. The alternator ventilating air is cleaned by a cheesecloth filter. The generator is designed to stand overloads of 25 per cent. for two hours, and 50 per cent. for three minutes, with a maximum temperature rise of 60 deg. above that of the surrounding atmosphere. The following guaranteed efficiencies were exceeded in the electrical test:

The generator supply is controlled from a 12-panel black enameled slate switchboard, from which the power and lighting circuits are taken. The power circuits are at 500 volts pressure, and three-phase, 40-cycle motors are installed throughout the electrically driven part of the mill. Two step-down static transformers, of 100 kv.-a. capacity each, reduce the pressure to 110 volts for the lighting circuits.

The British Admiralty has ordered from the four principal Scottish mineral-oil companies 200,000 tons of oil fuel. The price is about 4½c. per gal., and most of the oil will be delivered at Rosyth base. Many destroyers and torpedo boats are being fitted for burning oil fuel exclusively.

The Government as a Coal Purchaser

Under the old plan of purchasing coal, George S. Pope explains, in Bulletin No. 51 of the Bureau of Mines, when the consumer found fault with its quality, he was assured that it must be good because it came from a mine with an established reputation. But as each dealer claimed that his coal was equal in quality to that of the others, there was no standard and the contract could not be cancelled. Purchased under the Bureau of Mines specifications, the consumer gets what he pays for. The advantages are:

1. Bidders are placed on a strictly competitive basis as to quality and price. This simplifies the selection of the most desirable bid and minimizes controversy in making awards.

2. The field for both the government and the dealers is broadened, as trade names are ignored and comparatively unknown coals offered by responsible bidders may be accepted without detriment to the purchaser.

3. The government is insured against poor and dirty coal, and from condemnation disputes based on visual inspection.

4. Experience with the old contract shows that it is not always expedient to reject poor coal, because of the delay and cost of removal. Under the present system, rejectable coal may be accepted at a greatly reduced price.

5. A definite basis for the cancellation of the contract is provided.

6. Constant inspection and analysis furnishes a check on the practical results obtained in burning the coal.

7. Paying on the quality basis incites the contractor to prepare the coal more carefully.

The government purchases annually from $6,500,000 to $8,000,000 worth of fuel. This sum includes the cost of delivery and of stowage. The Navy, War, Treasury, Interior and Commerce and Labor Departments are large consumers. Much of the government coal must be delivered by wagon, limiting the business to dealers having hauling facilities, particularly in Washington, where coal is purchased principally for heating the public buildings.

For the fiscal year 1908-9 the Geological Survey made tests and analyses representing 611,400 tons of coal bought under contract at an estimated cost of $1,858,800; for 1909-10, 829,300 tons costing $2,286,800; for 1910-11 the Bureau of Mines made analyses covering 1,091,400 tons, costing $3,084,800; and during 1911-12 it is estimated that the bureau will make analyses covering 1,500,000 tons, and costing $4,750,000.

"They call it an electric light," Pat confided to his companion, "but it do beat me how they make the hairpin burn in that little bottle."

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Sizes and Capacities of Wires

By G. H. McKelway

In the early days of the wire industry each manufacturer had his own gage but, in the interest of uniformity, the gages have been made standard, and the one at first used by the Brown & Sharpe Co. is now accepted for all sizes of copper wire up to and including No. 0000. When referring to this gage it is generally known as the B. & S. gage, from the initials of the firm bringing it out.

With this gage the higher the number the smaller the wire, and each size has 26 per cent. greater cross-sectional area than that of the wire with the next higher number. For rough calculation it is easier to remember that each size is approximately 1.25 times that of the next smaller. This rate of increase makes the wire three numbers lower, about twice the size of the first wire, so that, for instance, No. 7 has twice the section of No. 10.

Another point, easy to remember, is that a wire ten sizes larger than another has ten times its area and therefore one-tenth of its resistance, as the resistance of a wire changes inversely as its section.

For wire larger than No. 0000 it is customary to give the number of circular mils making up their area. A mil is 0.001 in. and a circular mil is the area of a circle 0.001 in. in diameter, so that a wire 1 in. in diameter has a section of 1,000,000 circular mils.

It is very unusual to draw solid wire larger than No. 0000, which is equivalent to 211,600 circ. mils, or nearly ½ in. in diameter, as it is then so stiff that it is very hard to handle. In fact, sizes of No. 0 and above when solid are seldom insulated and their greatest use is for trolley wire. For interior wiring, such as is done for lighting or power in buildings, it is not often that a solid wire larger than No. 8 or No. 10 is used.

In order to get the necessary flexibility with the larger sizes of wires they are each made up of a number of smaller wires stranded together. These wires are wound tightly together yet it is evident that they can never be so placed that there will not be small openings in their section and that all of the surface of each wire will be touching some other wire. Therefore, a stranded wire of the same diameter as a solid wire will not contain as much copper as the latter. For this reason the size of the stranded wire is calculated, not from the diameter of the wire as a whole, but from the sum of the areas of each of the strands. It follows that a stranded wire has always a larger diameter than a solid one of the same nominal size.

Having learned the ratio between the different sizes of wire it is only necessary to remember the resistance of one size and the resistance of any other can be easily calculated. Probably the wire whose size and resistance can be most easily remembered is No. 10, which is also a size much used in interior wiring. The reason for the ease in remembering the facts concerning it lies in the number of tens involved. Not only is the number of the wire 10, but it also has a diameter of 0.1 in., an area of approximately 10,000 circ. mils, and a resistance of one ohm in 1000 ft. of length. These figures are not exact but are close enough for all calculations as they only differ from the exact dimensions and resistance by 2 or 3 per cent.

Where large wires are concerned it is unnecessary to remember their area and diameter as the former is given by the description of the wire and the latter, in thousandths of an inch, is the square root of the area. To obtain the approximate resistance per thousand feet it is only necessary to divide 10,000 by the number of circular mils of the wire and the result will be the resistance in ohms. For instance,

$$\frac{10,000}{1,000,000} = 0.01$$

which is the resistance in ohms per thousand feet of a 1,000,000 circ. mil wire.

Another way of stating the same rule is to say that the number of feet per ohm is equal to the result obtained by dropping one cipher from the number of circular mils of the wire. By cutting off the last cipher from 1,000,000 circ. mils 100,000 is left, which is the number of feet of a 1,000,000 circ. mil wire needed to have a resistance of one ohm.

On comparatively short runs it is probable that the size of the wire will be limited not by the drop caused by its resistance but on account of the fire underwriters' requirements as to the amount of current that may be carried by the wire without its becoming too hot for safety.

The underwriters divide all insulated wires into two classes, depending upon whether or not the insulation used on them is of rubber, and in the rule books supplied by the National Board of Fire Underwriters is a table giving the amount of current allowed for each size of wire. As copper wire is almost universally used in buildings the tables are made out for wire of that metal, but the statement is made that aluminum wires of the same size and having the same insulation are permitted to carry only 84 per cent. of the current allowed for copper. This is because of the greater resistance of aluminum wire.

The amount of current which a No. 10 rubber insulated wire is permitted to carry is limited to 24 amperes while with any other type of insulation the capacity of the same wire would be 32 amperes, or one-third more. This ratio between rubber and other insulating material holds good approximately for all of the B. & S. sizes, although it increases a little as the sizes increase.

When the wires are 200,000 circ. mils or larger the ratio between the different types of insulation is as 1: 1.5, rubber being always the lower, not because wire protected with rubber becomes hotter than with other insulations or even because the temperatures a little higher than those allowed by the underwriters would be dangerous to new wire, but because continued high temperature deteriorates the insulation and rubber is more easily affected than the other materials.

The amount of current that a wire can carry does not increase directly with its area, so that a wire twice as large as another can carry only about 1.66 times as much current as the smaller one. Therefore, while the size of the wire doubles with every three numbers, the current should not double until the fourth number is reached, when it can safely do so.

The current carried by the wires of the B. & S. gage and not smaller than No. 16 can be calculated from the amount permitted to be used on No. 10, as previously given.

When working with wires whose area is great enough to be given in circular mils, it should be remembered that a 200,000-circ. mil rubber-covered wire is permitted to carry 200 amperes, and that with other insulation 300 amperes is allowed. The capacities of other sizes can be readily found by allowing two-thirds more current for every time that the cross-section of the wire is doubled.

The Small Portable Storage Battery

By B. Dawson

Although battery manufacturers are constantly sending out instructive circulars regarding the care, charging rate, discharge rate, capacities, etc., of their respective products, yet the average user of the small battery keeps violating the instructions given.

A common mistake is to use the battery until the last bit of spark has gone out of it. Right here is where more or less damage is done to the plates by excessive sulphation. Unless proper precautions are used to rid the plates of the sulphate, the battery will more and more rapidly deteriorate.

In choosing a good portable storage battery, preference should be given to one having each cell terminal exposed so that it is easy to test the voltage of each cell during the charging period. It is also essential that the terminals be in good condition, free from the effects of the containing acid or charging acid gas in the form of creeping salts. Good-sized vent holes should be on the battery to allow easy inspection of the plates and the electrolyte in each cell; this will also allow the gases to escape freely during the charging.

The strap handle should be strong and durable and of the detachable kind. The strap handle should always be removed, except for carrying the battery from place to place.

Rubber jars in a battery have the advantage over glass jars in that the former will stand more shock; otherwise the glass jars are just as good. The positive and negative plates should amply comply with the manufacturer's rating; therefore it is essential to choose good reliable makes that will not disintegrate too easily on maximum discharge. Herein lies the difference between a cheaply constructed pasted plate, and a first-class one.

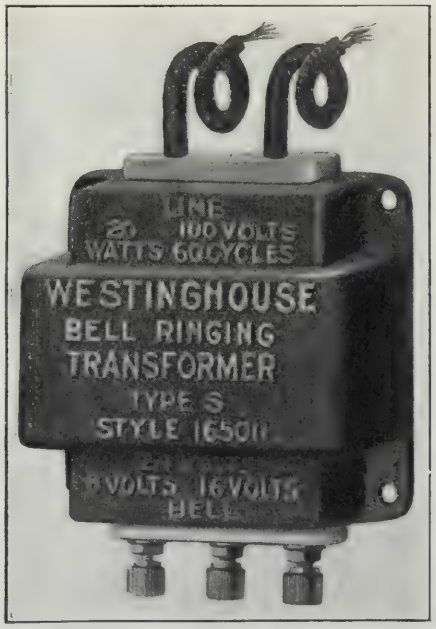
The writer has found the following to be common faults in the operation of the small portable storage battery: Dirty battery terminals; stranded and dirty wires connecting the leads; insufficient charge in battery; low capacity caused by sulphated plates, or low water in the cells; one or more short-circuited cells caused by sediment touching the plates, thereby lowering the terminal voltage; too small a battery for the required service; tampering with the electrolyte, such as spilling it or adding contaminated water, and one or more cracked jars, causing the electrolyte to slowly dribble away and ruin the outer casing.

Following are a few recommendations for the care of small portable batteries: Keep the batteries clean by washing all around the casing with pure water, taking care to keep water out of the cells. Keep the battery and terminals perfectly clean and make good connections. Never allow the battery to discharge below 1.65 volts per cell. Always recharge the battery immediately when the minimum (1.65 volts) has been reached. Never put the battery away for the winter in a discharged or low-water condition. When the battery is put away for the winter make sure it is placed in a dry, nonfreezing room, and is fully charged with sufficient distilled water added to cover the plates. Keep the vent plugs well sealed during this time to prevent undue evaporation; this will also keep out the dirt. If the batteries are needed in the spring a freshening charge should be given.

Do not let inexperienced or careless workmen charge or repair a battery. Avoid taking the batteries to repair shops where machine-shop bench-work and battery material are all handled from the same bench. Use only pure distilled water to replace loss by evaporation.

Bell Ringing Transformer

To supply electric bells from a lighting circuit, the Westinghouse Electric & Manufacturing Co. has placed on the market a new transformer, built according to the design of its well known type S distributing transformer. It is light, compact and absolutely fireproof, and will deliver on open circuit 8, 16 or 24 volts, or 16 volts fully loaded. It is particularly useful for apartment houses and public buildings.



Westinghouse Bell Ringing Transformer

The coils and magnetic circuit are vacuum dried and impregnated with a moisture-proof insulating compound. After this treatment the complete unit is fitted into its case. The interior of the case is then filled with a specially treated cement completely embedding the coils and iron in a fireproof and indestructible compound, which precludes any possibility of the bell-ringing circuit coming in contact with the 110-volt circuit.

Rubber-covered primary leads are brought out of the top of the case through a porcelain bushing and are left of sufficient length to permit connection to the lighting circuit. Binding posts are provided at the bottom as shown for the bell circuit.

CORRESPONDENCE

Station Inspection

The manager of the 2300-volt, three-phase, lighting system with which I am connected has inaugurated a system of tests and inspections which I believe to be indispensable to the upkeep of an electrical system.

Every three months the transformers, fuse boxes and lightning arresters are inspected. The height of oil in the transformers is noted and more added if necessary. The oil is tested for moisture by inserting a red-hot nail, moisture being indicated by a sizzling noise. Fuse boxes are inspected for loose contacts. When our system was first installed one set of secondary fuses blew quite often, and an inspection showed loose contacts which when tightened eliminated the trouble. The knurled cylinders of the high-tension lightning arresters had several burned spots, showing that they had been in action.

We test the machines and outside lines for insulation resistance with a megger. Starting at the power house we test the armature and fields of the exciter and alternator. With the machines exactly alike, one showed five megohms lower resistance than the other. The outside system, comprising four transformers and about 12,000 ft. of line, showed 300,000 ohms.

We next test each transformer and the lines radiating from it, removing the primary and secondary fuses. The transformer reading was infinity, the secondary lines testing from 40,000 to 100,000 ohms. The secondary lines included the interior wiring and fixtures.

The value of these tests is shown by the fact that one of our consumers claimed to have a ground on his system which caused his meter to record more kilowatt-hours than were used. Our test of his wiring showed 1½ megohms resistance, thereby disproving his contention.

The greatest value, of course, of systematic tests is the knowledge of defects, which in many cases can be remedied, and of impending breakdown.

Cecil R. Coile.

Fort Morgan, Ala.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

German Progress in Large Oil Engines

By F. E. Junge

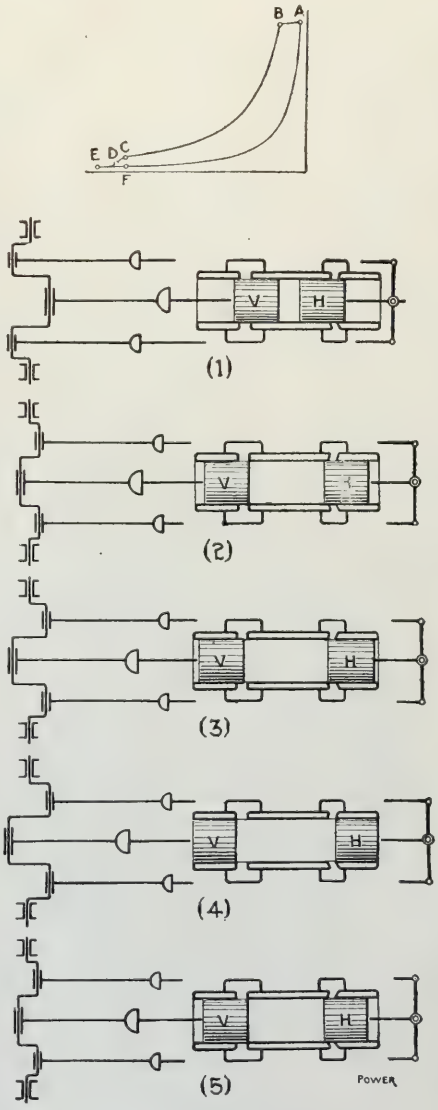
It is the purpose of this series of articles to bring American readers in close touch with the latest achievements of oil-power engineering in Europe, and especially in Germany, where, owing to the natural adversity of fuel conditions, and to the need of economizing in operating and other costs, power production has been developed to a high degree.

There seems to be a widespread impression in this country that the evolution of oil power in Germany has been and is associated exclusively with the development of the Diesel engine. While the Diesel principle has evolved a new era of power engineering, it would be unfair to slight the many individuals and firms who have taken an active part in the later development and have improved upon the original idea.

Of late much attention has been concentrated on the Junkers oil engine, claimed by the inventor, Professor Junkers, to be an improvement on the standard type of Diesel engine, an improvement in mechanical equipment, thermal efficiency, and capacity for overload. This engine operates on the Diesel cycle—using the term in the colloquial, not in the scientific sense—and was described by the writer in the Jan. 23, 1912, issue of Power. To make the present series complete, however, and for the benefit of those who have not read the former article, the general features of the Junkers engine will be reiterated, to be followed by details and a discussion of the design, leaving it to Professor Junkers to propound and to defend his scientific convictions.

The Junkers Engine

The Junkers engine works on the two-stroke-cycle principle, its method of operation being represented by Figs. 1 to 5. The cylinder is open at both ends and contains two pistons which are linked with a three-throw crankshaft, permitting them to move in opposite directions. In Fig. 1 the pistons are shown at their inner dead center, the space between them being filled with highly heated air, as a result of the previous compression stroke. In this position the fuel (oil) is injected into the compressed air in a fine spray, igniting and burning under constant pressure during the first part of the outward stroke from *A* to *B*, of the working diagram. From *B* to *C* the expansion of the burnt gases takes place, and at *C* the pistons have reached the position indicated by Fig. 2, in which the front piston *V* is just beginning to open a series of slots or ports, permitting the burnt gases to exhaust into the atmosphere. When the position shown in Fig. 3 is reached



Figs. 1 to 5

the equalization of the internal pressure with that of the atmosphere is almost completed. Simultaneously the rear piston *H* uncovers its series of slots, permitting fresh air at low pressure to enter the cylinder and expel the remaining burnt gases through the exhaust slots. This scavenging is continued until the pistons have passed the outer dead center (Fig. 4) and taken up the position shown in Fig. 5. At this instant both rows of slots are closed by the pistons traveling toward their inner dead center, a distance which is represented by *EF* in the diagram; at point *F* the cylinder is filled with fresh air, cut off from the atmosphere. As the pistons approach one another in traveling toward the inner dead center (*FA* in the diagram), the contents of the cylinder is compressed and heated to such a degree that the fuel injected shortly before point *A* is reached, ignites and the cycle is repeated.

For large work the tandem arrangement is used, which reduces the weight, cost and space of the engine per unit. In this arrangement a three-throw crank is used, the two outer pistons being linked with the central crank, and the two inner ones to the side cranks, which are placed at 180 deg. to the former. This is represented in plan and elevation by Fig. 6. By this arrangement each stroke is a working stroke. While the pair of pistons in one cylinder are executing an outward movement (power stroke), the pair in the other cylinder are traveling toward each other (compression stroke). When the pistons of one cylinder occupy the inner dead center, those of the other cylinder take up the outer dead-center position. In this way, while the fuel injection is taking place in one cylinder, scavenging is going on in the other. Scavenging is accomplished in the same way as in the single-cylinder engine.

The governing of the slots is performed by the pistons, complicated governing elements being thus dispensed with. Arranged symmetrically to the center line of the cylinder are the two scavenging pumps, which supply the air for scavenging, and the four-stage air compressor for the fuel spray air is similarly arranged; these auxiliaries are actuated from the yoke between the middle pair of pistons. The details of construction will be given later. For large work each cylinder has two fuel valves and one starting valve for compressed air. The fuel injection is performed in the ordinary way by means of highly compressed air, the atomized fuel being brought into close contact with the air provided for combustion.

As can be inferred from the schematic representations of the Junkers engine, the cylinders are simple castings, compared with the more complicated cylinders of the standard Diesel engine. The acting

forces balance one another in the power-transmitting mechanism, whereby an almost complete balance of the inertia of the reciprocating parts is attained. The bedplate need only extend up to the first cylinder, the cylinders themselves being carried on flexible supports and resulting in a comparatively light and inexpensive engine. The cylinders have no covers and no stuffing-boxes; hence on one side the pistons are always in contact with the atmosphere. Moreover, with every outward stroke they enter into well cooled parts of the cylinder, which are not touched by the products of combustion. This mode of operation secures a perfect lubrication of the pistons and cylinders, the oil being admitted at points, not in contact with the burnt gases.

Owing to the division of the total stroke between two pistons a high aggregate piston speed is attained, as well as a high rotative speed, and comparatively low separate piston speeds. Another feature is the long cylinder of relatively small diameter, offering exceptional scavenging opportunities and providing a favorable combustion chamber. A scientific discussion of these various features will be given in following articles.

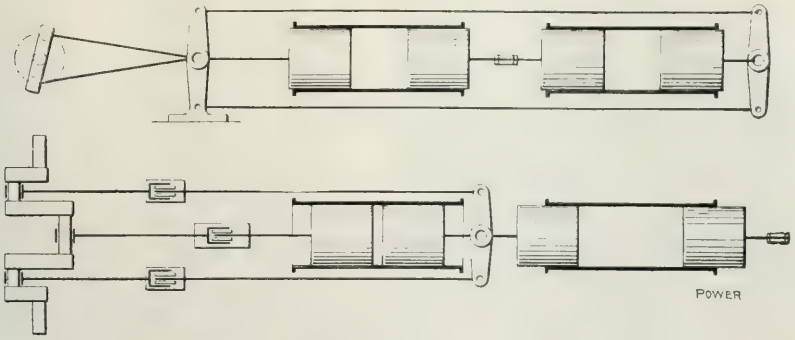


FIG. 6. PISTON AND CRANKSHAFT CONNECTIONS

Two-stroke Cycle Scavenging

BY ROBERT MILLER

All modern two-cycle engines operate on the Clerk cycle, or slight modifications of it, which are mostly mechanical.

The Clerk engine had a separate pump for handling the new charge which entered at the head end of the power cylinder. One modification uses the crank end of the power piston and incloses the crank space, so that on the outstroke this piston has combustion on the cylinder side of the piston and a slight compression of the fresh charge on the crank side. On the instroke, it has compression of the new charge on the cylinder side of the piston, and the suction of a fresh charge in the crank chamber. Other modifications use separate pumps for handling the air and the gas. Sometimes the new charge is delivered to the power cylinder at the head end; sometimes through ports uncovered by the piston head at the end of its outstroke.

The scavenging charge enters the power cylinder either as a blast or as a slow displacement. Where the power piston acts as both a power and a pump piston it is supposed to give the blast effect; where the pump pistons are timed half a stroke in advance of the power piston the other effect is desired. In all cases the expulsion of the residual gases is believed to be by the displacement due to the entering charge. And it is because this displacement is imperfect at its best, and unsatisfactory under varying conditions of speed, load and throttling, that the two-stroke-cycle has not, after 30 years, displaced the four-stroke-cycle or even held its own, except for certain classes of service.

When the piston of a four-stroke-cycle engine travels inward on its exhaust stroke, it pushes ahead of it a column of exhaust gas. At all speeds, loads and throttle openings, this piston travel is the same, so that the scavenging is positive, and always the same, neglecting the slight variation due to wire-drawing.

The simplest form of two-stroke-cycle is the crank-case compression with piston uncovered ports. Near the end of the explosion stroke, the piston uncovers the exhaust port; the excess pressure in the cylinder is relieved at this point until the further movement of the piston uncovers the fresh-mixture inlet port. Depending upon the size of the exhaust port, its lead over the inlet port, the pressure in the cylinder, and the speed of the engine, three conditions may exist. The pressure in the cylinder may be greater, in which case part of the cylinder contents will pass into the crank case until the two pressures are equalized. Moreover, the temperature of the cylinder contents may be high enough to ignite the fresh charge. While this equilibrium is being established, the exhaust port is still open, so that while the crank-chamber contents flow from the admission port into the cylinder, there is a flow from the cylinder through the exhaust port. It is usual to place a deflector on the piston head to direct the crank-chamber contents to the cylinder head; but with the pressures in the cylinder and crank chamber equal, there is no cause for the new charge to travel to the head when it is much easier to pass over the deflection to the open exhaust port.

When the pressure in the cylinder and the crank case are equal at the moment of inlet-port opening, the same action occurs as in the first case.

The third condition is entirely different, yet it is what is usually assumed to be true. The cylinder pressure being lower than the crank-case pressure the new charge passes out of the bypass port and strikes the deflector with considerable velocity. A small portion gets over the top and goes directly to the exhaust port; the major portion is directed upward. There is a volume of burned gas in the cylinder either quiescent, or moving slowly toward the exhaust port. The new charge moving rapidly meets this mass; so that the action instead of being a real displacement is a quick intermingling. In fact, this blast may pierce the old charge and displace very little of it.

Where the charge enters at the head end it is usual to provide valves which are opened by the greater pressure of the new charge, or mechanically by cams or eccentrics. A few employ piston valves.

Where the valves are in the head, the new charge does not directly displace the volume directly behind the valve (poppet type) and the stream is in the form of a hollow cylinder. The clearance-space regions remote from this stream are effected by it only by an eddy action and have no real displacement. In some cases the new charge has been given a swirling action, in an attempt to reach the distant regions of burnt gas.

Where a blast action is not desired, but rather a gradual displacement, the action is somewhat as follows: At the moment the inlet opens, the cylinder and displacing pressures equalize; then, depending upon the shape of the valves or other impediments in its path, the charge passes in straight lines toward the exhaust ports, the surrounding residual gas gradually passing into the column.

All of these actions will vary because of the four reasons stated. The size of the exhaust port and its lead over the inlet port, determine the cylinder pressure at the moment of inlet opening. The pressure in the cylinder being dependent upon the throttle, open throttle means high exhaust pressure. The speed affects the time available for the exhaust to pass out, as double the speed allows only one-half the time.

Thus far displacement by full charges only have been considered. When the charge is throttled, more complications result, as the pressure balance is again disturbed. In addition, small new charges displace small quantities of old gas, leaving more old gas to foul the new. In a four-stroke-cycle, with the smallest throttle opening at which the engine will run, the exhaust displacement is the same as at full throttle, so that the small fresh charge is only slightly fouled by the burnt gas in the clearance.

The Blanchard Oil Engine

Among the latest additions to American oil-engine practice is the Blanchard engine built by the Blanchard Machine Co., of Cambridge, Mass., in sizes of from 8 to 100 hp. This is a slow-speed heavy-duty engine operating on the two-stroke cycle with kerosene, distillate or fuel oils. Fig. 1 shows the general appearance of a single-cylinder Blanchard engine, although they are made with as many as four cylinders.

Only moderate compression is employed and as the oil is sprayed into the combustion chamber, is vaporized by the hot air and ignited by a hot tube, relatively slow combustion takes place; hence the engine is not subjected to such extreme pressure and shocks as prevails in the ordinary gas or gasoline engine.

Considering the cycle of operation of the Blanchard engine, Fig. 2 represents a vertical section through the cylinder with the piston at the bottom of its stroke. As it starts on the upstroke it covers the inlet port *A*. The crank case is air-tight and as the air-intake port *B* is always covered except when the piston is at the top of the stroke, a partial vacuum is formed within the crank case. When the piston reaches the top of the stroke and uncovers the air-intake port, air is sucked into the crank case through the pipe *C*. On the down-stroke this air is compressed to about 5 lb. Just after the exhaust port *F* has been uncovered the piston uncovers the transfer port *D* and the inlet port *A*, permitting the air confined in the crank case to rush through the passage *E* and, deflected by the piston, into the upper part of the cylinder. This serves to scavenge the cylinder. Upon the next up-stroke this fresh air is compressed to about 200 lb. (the exhaust and inlet ports being covered by the piston).

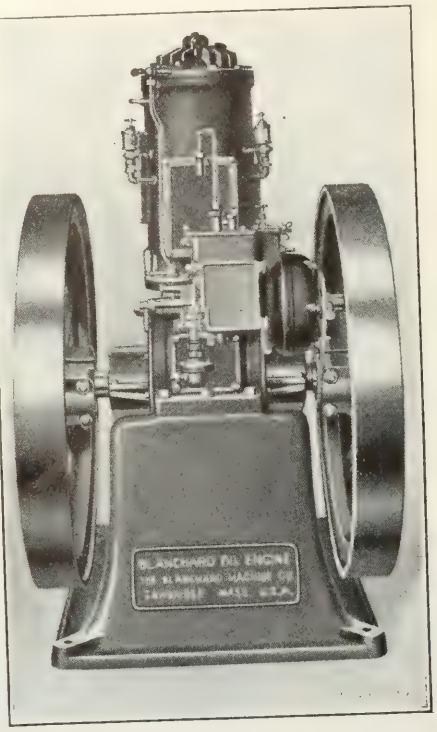


FIG. 1. SINGLE-CYLINDER BLANCHARD ENGINE

The heat of compression, together with that absorbed from the cylinder head, which is unjacketed, is sufficient to raise the temperature of the air to such a degree that the oil sprayed in is instantly gasified and is ignited by the hot plug *G*. In starting the engine this plug is first heated by the blow-torch *H*, but after the engine is in operation this is kept hot by the burning of each successive charge.

The injection of water into the combustion chamber forms a special feature of this engine.

Fig. 3 is a section through the fuel feed mechanism, the oil pump on the left and the water pump on the right, *I* and *J* being the oil and water plungers respectively. Both are driven by the camshaft *K*. By pushing in or pulling out the piece *L*, which is held by a set-screw, the stroke of the water plunger is varied. In this way any desired proportion of oil and water can be had. This proportion then remains fixed, although the quantity of fuel admitted is determined by the governor, which regulates the length of stroke of the pump plungers. The lever *M*, Fig. 2, is for priming.

The governor is of special design and is guaranteed to regulate within 3 per cent. from no load to full load.

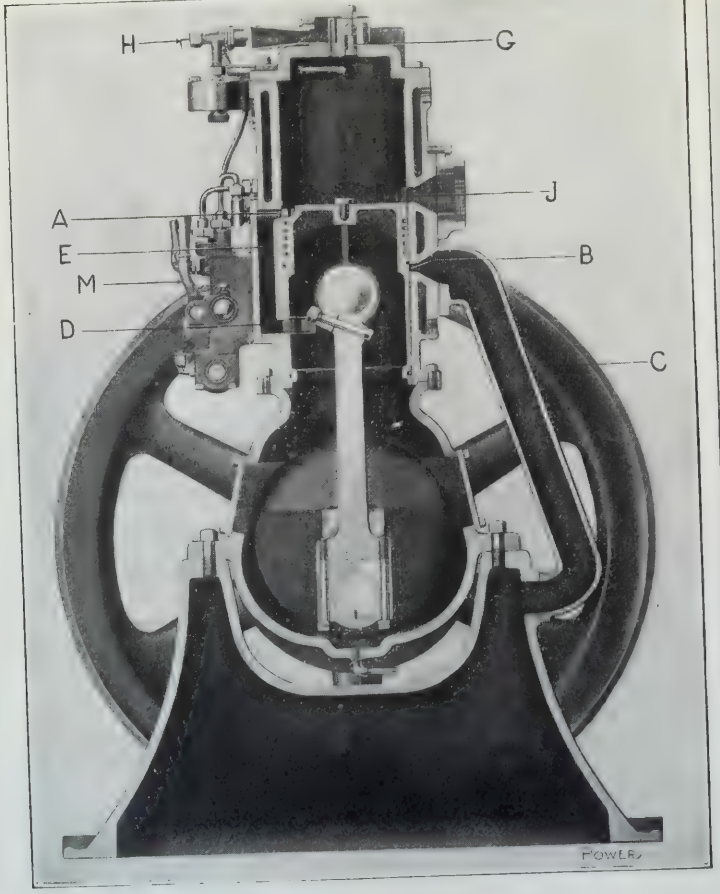


FIG. 2. SECTION THROUGH CYLINDER

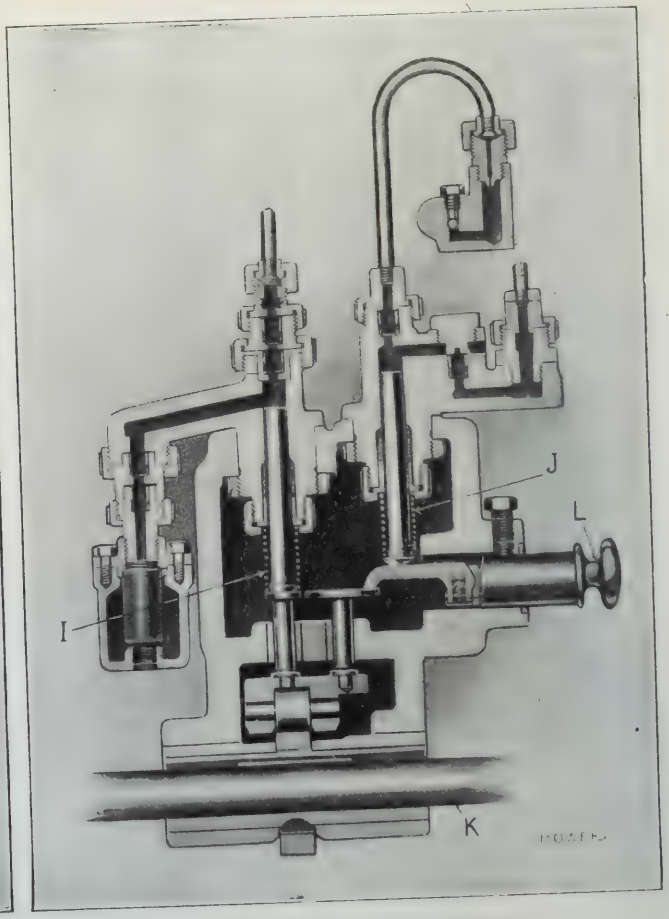


FIG. 3. FUEL AND WATER PUMPS

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Indicating the Ammonia Compressor

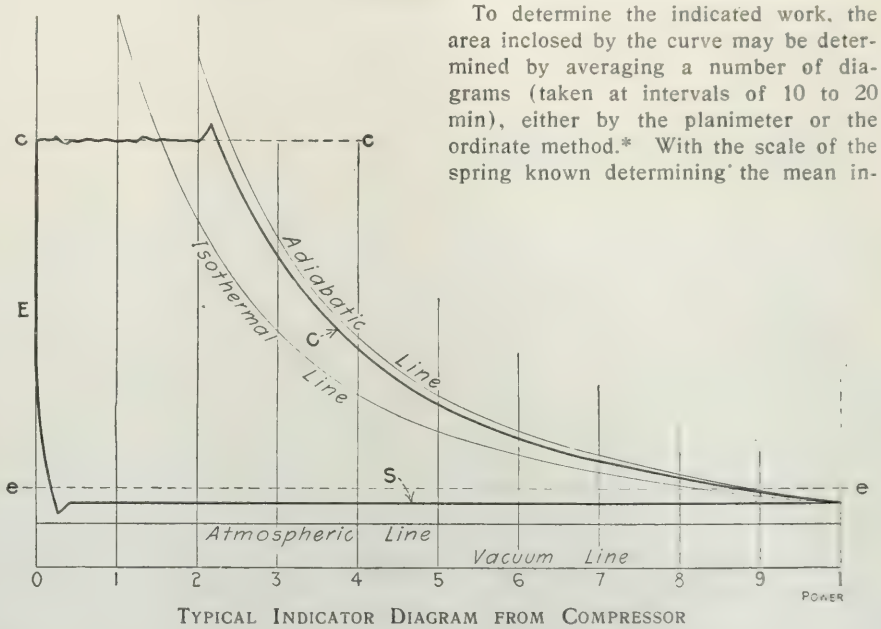
By John Creen

Regularly indicating a compressor and knowing how to correctly interpret the diagrams obtained is of great value. The indicator diagram is an important permanent record of the quality and quantity of work done by the compressor relative to its capacity. Diagrams should be taken every two weeks or so, and filed for future reference. For ammonia machines a special indicator entirely of iron and steel, fitted like the ordinary steam-engine indicator, is generally advisable. The ordinary bronze or composition indicator can be used without fear of damage, if the piston is removed, thoroughly cleaned, and the indicator cylinder kept well oiled with clean compressor oil after finishing the test. Diaphragm indicators, owing to the irregular bending of the diaphragm, have not given good results compared with piston instruments, although they let no ammonia escape.

The best compressor manufacturers provide their cylinders with indicator pieces bored along their whole length to the interior of the cylinder and generally sealed by a setscrew and flange besides having a thread for the indicator cock. Regrettably, up to the present, no definite standard for these threads and connections has been adopted. Very often intermediate pieces have to be procured to suit the threads before the indicator can be applied. This presents difficulties where, in order not to increase the clearance, the indicator opening is closed by a long pin not easy to remove. It would be better if the openings had a flange carrying both the thread receiving and indicator cock and the bored pin for filling the clearance.

With carbon-dioxide machines the pressure becomes so great that the ordinary indicator piston of ¾ in. diameter cannot be used. A smaller piston of 0.25 or 0.4 in. diameter, working in a lower narrow part of the indicator or in a box screwed therein, must be employed. These are commonly called Riedler pistons. The high pressure of the carbon dioxide also makes applying the normal indicator cocks very difficult; using small screw-down valves is better. These need not have a bore greater than ⅛ in., but should have a blow-through plug fitted to the cavity in connection with the indicator to restore atmospheric pressure under the indicator piston after taking a diagram.

Before the indicator cock is fixed to the compressor, the suction valve is closed and the machine allowed to make a few revolutions, so that most of the cylinder content is exhausted. The machine may then be stopped, the discharge valve closed and the indicator cocks or valves, as the case may be, screwed in position. The air which has entered the cylinder must finally be expelled by opening the cocks, and to a slight extent the suction valve, after which the discharge valve is opened again and the machine started. As soon as the suction valve is full open, a reading may be taken.



Typical Indicator Diagram from Compressor

The Indicator Diagram

On the diagram obtained the straight lines corresponding with the condenser and evaporator pressures must be drawn as soon as possible, for which purpose the manometer must be previously compared with the indicator. If all parts of the machine are in order, a diagram somewhat like the accompanying illustration will be obtained, the suction line *S* lying a little below the evaporator pressure *ee*, and the pressure line somewhat above the condenser pressure *cc*. The work required for opening the two valves is shown by the two little peaks, one on the suction line pointing downward and the other on the pressure line pointing upward.

The influence of clearance is seen from the back expansion curve *E*, which cuts the evaporator line *ee*, as shown, and diminishes the actual suction volume, quite apart from the fact that the vapors are not at the evaporator pressure, but at the somewhat lower suction pressure. The compression curve *C* in the diagram reaches the evaporator pressure only after the change of stroke, so that instead of the theoretical volume indicated by the length of *ee* the compressor has a smaller suction volume for vapors at the evaporator pressure.

Horsepower of Compressor

To determine the indicated work, the area inclosed by the curve may be determined by averaging a number of diagrams (taken at intervals of 10 to 20 min), either by the planimeter or the ordinate method.* With the scale of the spring known determining the mean indicated pressure is easy, and multiplying this by the area of the piston in square inches, gives the mean piston pressure in pounds. As the piston rod reduces the piston area on one side, this process must be carried out separately for each side. If then the mean value of the two piston pressures is multiplied by the stroke in feet and twice the number of revolutions per minute, the work in foot-pounds is obtained, and this divided by 33,000 gives the horsepower.

For example, take an ammonia compressor with a 10-in. diameter cylinder, a 2-in. piston rod and a stroke of 15 in. = 1.25 ft. The effective piston area is 75.4 sq.in. on the piston-rod side and 78.54 sq.in. on the other. If the mean indicated pressure is 41.54 and 42.80 lb. per sq.in. respectively, the total piston

*See Power, June 18, 1912, page 881, "Doing Without a Planimeter."

pressures will be 3132 lb. and 3361, the mean being 3246 lb. Then

$$\frac{2 \times 60 \times 1.25 \times 3246}{33{,}000} = 14.75 \; hp.$$

Defects Indicated by Diagram

The indicator diagram is useful in pointing out faults in a compressor. Too large a clearance flattens the back expansion curve, and means that the suction volume is being given insufficient use. A similar effect is produced by the jamming open of the discharge valve, caused by the spindle taking an oblique position in its guide. This defect frequently causes a late opening of the discharge valve and naturally a large expenditure of work, which is shown on the diagram by a large peak in the pressure curve. After opening, a loose valve immediately fixes itself wide open, and only when it shuts at the instant of the change of stroke does the defect come to light. With a long stroke the pressure valve may rebound from its seat; this is shown by a hook in the back-expansion curve, which runs very level from this point. Insufficient use of the stroke volume is the consequence. If the back-expansion curve proceeds normally after the hook, nothing is wrong with the play of the valves. The indicator piston clutching or its spring vibrating may have been responsible. Losses also arise from too high resistance in the suction and discharge pipes, too strong tension on the valve springs, faulty adjustment of the valves or the presence of an air cushion. These defects are indicated by too wide a variation from the evaporator or condenser pressures. In such cases the valve springs must first be replaced by weaker ones or more play be given to the air buffer; then if no appreciable improvement is obtained, constrictions should be looked for in the pipes and removed if any are found.

Jamming of the suction valve in the closed position will cause a considerable fall in pressure before the commencement of suction and the formation of a large peak on the suction curve. If the valve jams in the open position, compression is greatly delayed; that is, part of the stroke of the piston is wasted.

Much looseness of the valves is shown by the disappearance of the horns or peaks. The compression curve is steep and almost rectilinear when the pressure valve is the looser, and flat when more looseness exists in the suction valve.

When the piston does not fit tightly, the gas passes from one side to the other so that the pressure rises too slowly on the compression side and the compression curve becomes very flat. The late, and therefore sudden, opening of the discharge valve in this case is immediately recognizable by the vigorous banging of the valve on its seat. On the other hand, the pressure on the suction side, when the piston changes its direction, easily remains higher than that in the evaporator, and so prevents the suction valve opening. This lasts until the piston velocity has become so great that the leakage past the piston is no longer able to fill the suction space. The pressure then gradually falls and the suction valve begins to work. If several of the above defects are present, very involved diagrams may be produced; but their interpretation should not be difficult to the experienced engineer. Regular waviness of the various curves is generally due to indicator spring vibrations and not faulty valve action.

Most engineers are familiar with indicator diagrams and the meaning of their various lines; the admission line parallel to the atmospheric line, the compression line rising in an easy curve to the discharge line, and the latter usually wavy or peaked, due to vibration of the discharge valve. The admission and discharge lines should be joined by a nearly vertical line and the diagram should have a square heel, this representing the amount of clearance. But what the engineer needs is a guide, something he knows is correct with which to compare his diagram. The only true guide for comparison is the adiabatic line.

To plot the adiabatic line on the diagram one must have a table of adiabatic constants. The diagram is divided into ten equal parts. From the table of constants in the horizontal line is found the number corresponding to the absolute back pressure on the diagram; then, in the same vertical column which contains the absolute back pressure, are the values of p_1 to p_9. These quantities are laid off on the lines drawn on the diagram to the scale of the indicator spring; this gives a series of points through which a smooth adiabatic curve is drawn.

If the ammonia gas could be compressed from the beginning of the stroke to the condenser pressure in a perfectly tight and nonconducting cylinder, without loss or gain of heat, the indicator pencil would trace an adiabatic line. With no leakage past the piston or valves, the adiabatic line for all ammonia compressors should almost overlie the compression line of the card for its whole length. The water jacket seems not to affect the compression line materially, but, during the last quarter of the stroke when the gas is very hot, the adiabatic line may be slightly above the compression line. If the latter is not closely adiabatic, something is known to be wrong with the piston, valves or gaskets of the compressor, if the indicator has been properly connected, the pipe leading from the cylinder to the indicator is short and of small bore, say ⅛ in. diameter, and this pipe is well insulated from the cooling effect of the jacket water.

If an engineer periodically indicates his compressor as well as the steam end, and makes himself familiar with a simple and practical method of plotting the adiabatic line on the diagram, he will be able to get much better work from his machine.

CORRESPONDENCE

Drawing Charge from Double Pipe Condensers

As most refrigerating engineers know it is difficult to withdraw the charge of ammonia from a system using double-pipe condensers. The ammonia seems to lay up in the condenser and will not come down to the receiver.

In a 100-ton ice plant we had a 12-stand condenser 12 pipes high. Two pipes in each stand were 2½ in. in diameter with a 1¼-in. water pipe inside; the other 10 pipes were 2 in. in diameter with a 1¼-in. water pipe. The condenser was about 40 ft. above the liquid receiver which was located on the ice tanks close to the expansion valves.

On the first pump down I drew out all the liquid in the receiver, which amounted to about 35 lb. I knew that there was a great deal more liquid somewhere as there should be about 2000 lb. Upon investigation the liquid was found in the condenser coils.

To get the liquid down to the receiver, I closed the valve at the receiver on the liquid line feeding the expansion valves and then opened the expansion valves wide. I closed the gas and liquid lines on four of the 12 stands of pipe in the condenser and opened the auxiliary or pump outline to these four stands. The auxiliary line was connected to the suction side of the compressor, and I pumped down the system and these four stands of pipe to atmospheric pressure, discharging all the gas into the other eight stands of the condenser. Shutting down the compressor I let the gas condense and after a while drew off all the air and foul gas in the condenser through the purge valves, and the liquid into the receiver. Then I cut out four more stands of pipe on the condenser and proceeded in the same manner as before until the last pump down when all the stands of pipe on the condenser were cut out except one, which was used to condense the gas.

The system and the eleven stands of pipe were pumped down to 15 in. of vacuum and the remaining liquid was drawn out into the receiver.

C. E. Anderson.

Chicago, Ill.

Correction—In Mr. Anderson's article on "Setting the Expansion Valve," issue of Sept. 24, page 458, eighth line, an odd misprint occurred. The line should read: close the expansion valve and *pump down the coils*, etc.

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POWER

The Value of the Horsepower

When the steam engine began to assume industrial importance, it became necessary to have some unit by which to designate its capacity. The work to which it was initially put had been for the most part done by horses, and the capacity of the engines was naturally compared with that of the horses which they replaced. Savery in 1702 considered that an engine which would raise as much water as two horses working continuously would replace ten horses, and hence was a ten-horsepower engine, since relays of horses must be used to keep the work going continuously.

When Watt and Boulton got into the engine business they established a standard of their own. Some heavy horses from Barclay & Perkins brewery, London, were caused to raise a weight from the bottom of a deep well by pulling on a rope passing over a pulley. It was found that a horse could raise a weight of 100 pounds while walking at the rate of 2.5 miles per hour. This is 22,000 foot-pounds per minute. Watt added fifty per cent. for friction and good measure and established the value of 33,000 foot-pounds per minute or 550 foot-pounds per second.

But if Watt's well had been in New Orleans, and if his methods of measurement had been sufficiently accurate he would have found that with the same effort his horses could have lifted more than they could at London. The attraction of the earth upon the same mass is greater at the poles than at the equator, greater at the sea level than at an elevation.

This is due to the fact that on account of the greater diameter of the earth at the equator, a body there is further from the earth's center, as it is also at an elevation, and to the fact that at the equator the effect of centrifugal force which opposes that of gravity is greatest, becoming nothing at the poles.

Watt's definition of a horsepower is therefore not a fixed definite unit, but has a different value for every latitude and elevation. If we take it at 550 foot-pounds per second at 50 degrees of latitude (about that of London) it will become 549 foot-pounds per second at the poles and 552 at the equator. At 45 degrees it would be 550.48 at sea level, 550.45 at an elevation of 5000 feet and 550.61 at an elevation of 10,000 feet.

While these small differences are not likely to be serious in ordinary commercial transactions, it is not right nor scientific that so important a unit as that of power should have a changeable and unstable value.

A watt is the power developed by the action, with the velocity of one meter per second, of a force capable of giving to a mass of one kilogram in one second a velocity of one meter per second. Since the kilogram is a definite mass (that of a cubic decimeter, or liter, of pure water at the temperature of its maximum density) the force which will accelerate it at a given rate is also a fixed quantity, and the rate at which work is done when the force is exerted at a velocity of one meter per second is a definite, absolute unit, independent of latitude, altitude or any of the disturbing factors which complicate the unit determined by the effect of gravity.

The watt, or kilowatt is thus as much a mechanical as an electrical unit and may be used to indicate rates of power development and application even when no question of electricity is involved with more accuracy and as much propriety as the horsepower. Its use for this purpose should be encouraged, but since most mechanical determinations in English-speaking countries are made in foot-pounds, it is necessary that there should be a definite relation established between the gravitational unit, or horsepower, and the absolute unit or kilowatt.

The round number usually employed for this equivalent is 746 watts, which happens to equal 33,000 foot-pounds per minute for the latitude of 50 degrees, nearly that of London where Watt's experiments were made. By defining the horsepower, therefore, as 33,000 foot-

pounds per minute at 50 degrees latitude and sea level, its equivalence with the absolute system of units is established as 746 watts.

The United States Bureau of Standards has adopted this value and, in a recently issued circular entitled the "Relation of the Horsepower to the Kilowatt," discusses the question historically and otherwise giving tables of the value corrected for different latitudes and elevations and urging the use of 746 watts as the exact equivalent of the horsepower instead of the inconvenient 745.6494 relation which obtains when the standard 45 degrees of latitude is used.

Boiler Plate

The Massachusetts Board of Boiler Rules will hold its regular semiannual meeting on Thursday, Nov. 7, as provided by law, to consider changes in the boiler rules for that state. It would be a good time to propose a few changes in these rules regarding the requirements for boiler plate.

The rules specify that shells, drums and butt straps, shall be made of open-hearth "firebox" steel, and certain physical and chemical requirements are specified for this class of steel. There are two other grades of steel specified that may be used for other purposes, viz., "flange" and "extra soft" and the physical and chemical characteristics specified for the three overlap. Thus, if a steel of 55,000 to 62,000 pounds tensile strength has an elongation of 26 per cent. or more and a maximum of 0.03 per cent. phosphorus and 0.04 per cent. sulphur, and contains from 0.30 to 0.50 per cent. of manganese, it is either "flange" or "firebox," whichever the steel manufacturer chooses to stamp it. Similarly, a steel of 52,000 to 55,000 pounds tensile strength and 28 per cent. elongation and with the same maxima of phosphorus, sulphur and manganese is "firebox" or "extra soft," whichever the manufacturer prefers.

Since the chemical and physical qualities and not any process of manufacture determine the brand, why do not these qualities also determine the use that may be made of the material? A plate of either quality mentioned above, should be equally suitable under the law for use in a shell, regardless of the brand stamped upon it.

The law requires that the material that is to enter the shell of a boiler that is built for use in the state, be tested and the test certified to. Why should not these tests determine the fitness for use of the material tested? Considerable money has been spent by the steel and boiler manufacturers in restamping plates so that they might conform to the wording of the law after a boiler had been built and reached its destination. Surely such stamping added nothing to the safety of these boilers, and safety is the real and only object in the enforcement of the law.

Another point about boiler plate that might well be considered, is the present relation of its tensile strength to the pressure allowed on a boiler, which places a premium on the boiler manufacturer using material of the maximum allowed tensile strength. The rules allow a range of tensile strength for shell material from 52,000 to 62,000 pounds per square inch; consequently the boiler maker endeavors to obtain from the plate manufacturer the highest tensile strength material that he is willing to guarantee will not exceed the maximum allowed, because this will produce a boiler for a given pressure with a minimum of material and cost. As a consequence, practically all the plate that enters boilers built for the state of Massachusetts, has between 57,000 and 62,000 pounds per square inch tensile strength, this being as small a range as the steel manufacturer cares to keep within.

Almost all authorities agree that a lower range, say, 50,000 to 55,000 pounds would produce a material better suited to boiler purposes. If the method of fixing pressures on new boilers was arranged so that the actual tensile strength of the material for the shell did not enter the calculation, but that the lowest tensile strength allowed for this class of material would be used for this purpose, there would be no incentive to the boiler manufacturer to secure material of high tensile strength, and consequently boiler plate better suited for the purpose would be obtained.

One good reason for trying to keep near the lower part of the range allowed by the law is that the larger part of a plate is usually of considerably higher tensile strength than that of the coupon used for testing its strength. The test coupons are necessarily cut from the edges of the plate, where the impurities and the carbon content are less, due to segregation in the ingot, and consequently the tensile strength lower than for points further in from the edge. Where a coupon cut from the edge of a plate tests 55,000 pounds per square inch, the bulk of the plate is likely to range anywhere from 60,000 to 65,000 pounds per square inch in tensile strength.

Forehandedness

A man seriously ill calls a doctor or one in legal difficulties a lawyer, but very few are wise enough to seek doctor or lawyer for advice before they are actually in trouble. They forget that "an ounce of prevention is worth a pound of cure."

So it is with engineering. Many owners attempt to make changes in their plants or even erect new ones, without first obtaining expert advice. Then when they begin to have trouble, they call in the consulting engineer, and often much of the work, done wrong in the first place, has to be done over.

An example in mind is that of a small manufacturer who wanted to enlarge his roofing plant. Wishing to save money he decided not to call in an engineer to design it for him, but secured a millwright to do the work. As soon as the plant was started, trouble began from the tearing of the roofing. After wasting about a month trying to fix it, the owner sent for a consulting engineer. On checking up the speeds of the several drives, he found they were not running in unison, and it was necessary to redesign the plant, and order new sprockets, etc. So besides having to pay for the services of the expert, it cost the owner much lost time and added material.

Countless similar illustrations could be given of that foolish economy, which in the end costs many times what it saves. The man who will effect an ultimate saving and obtain the best economy is the one who does not begrudge a little additional first cost. Let the owner engage the engineer in the beginning; for he is the one best fitted by experience and education to do the work right, and will save his client many times his fee in future worry and expense. Happy is he who has this expert in the man in charge of his power plant; if he is a real engineer it is not necessary to go outside.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Indicator Diagrams from Engine and Steam Pipe

The two indicator diagrams, Fig. 1, were taken simultaneously, one with an indicator attached to the cylinder and the other with an indicator connected to the steam pipe supplying the engine. Both indicators were connected to the same reducing motion. The engine was a 20x42-in. Corliss, running at 74 r.p.m. connected by a 6-in. steam pipe 50 ft. long with two 72-in. by 18-ft. horizontal return-tubular boilers. A 60-lb. indicator spring was used. When these diagrams were taken the steam gage read 103 lb., but the steam line drawn by the indicator shows 101 lb. The two diagrams have been traced below each other so that the abscissæ of both correspond. The line diagram shows a sharp drop to 94 lb. as soon as the steam valve opens, but starts to rise just before the point of cutoff is reached.

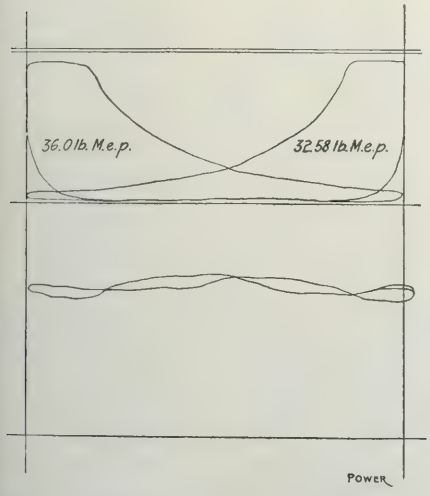


FIG. 1. INDICATOR DIAGRAMS FROM ENGINE AND ITS STEAM PIPE

The flow of steam from the boiler then raises the pressure line up to 110 lb., a return wave toward the boilers then causes the line pressure to fall to 96 lb. and the flow from the boilers then commences to raise the line pressure which reaches 102 lb. just as the steam valve at the opposite end opens. The line pressure now drops to 94.5 lb. but commences to rise near the point of cutoff until it reaches 108 lb. dropping, as a return wave starts to the boiler, to 97.5 lb. The two diagrams show a complete cycle of the pipe-line waves and their relation to the stroke of the engine. A number of other diagrams taken showed similar cyclic changes in the pipe-line pressures.

This succession of waves in the pipe line is a good illustration of why some pipe lines vibrate and likewise why a receiver close to the engine is advantageous. In this case the engine received 148 steam impulses per minute (two per revolution) and the waves in the pipe line have an approximate frequency of 296 per min., so that the live steam is "packed" into the cylinder at the beginning of the stroke by the wave impulse apart. By plotting the position of the crosshead at equal time intervals on the indicator-wave diagram, the abscissæ of the wave are obtained. The wave can then be replotted upon a base line having time abscissæ of equal values. This has been done in Fig. 3 in which the amplitude of the wave has been multiplied by five. Where the length of this wave is a harmonic of the distance between the boiler and the engine in the course in

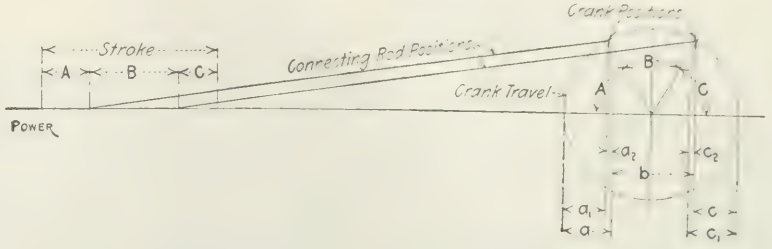


FIG. 2. SHOWING EFFECT OF ANGULARITY OF CONNECTING ROD

as the line steam wave is in phase with the speed of the engine. On some of the diagrams the pipe-line wave was slightly out of phase with the engine though in none of the diagrams taken did the steam valve open with the pipe-line pressure at a minimum.

The diagram showing the wave in the pipe line is distorted as the indicator was actuated by the crosshead of the engine which does not travel at a uniform speed. The average piston speed in this case is 8.63 ft. per second, the maximum being twice this, or 17.26 ft. per second, while the steam wave has a practically uniform velocity. To obtain the true shape of the wave it is necessary to re-plot it upon a base having a uniform time increment. The angular velocity of the engine crank is practically uniform; hence the distance the crosshead moves when the crank moves through any angle A or A_1 is equal to the algebraic sums of the versed sines of the angles through which the connecting-rod and the crank move. See Fig. 2, in which A, B and C represent the distance through which the crosshead moves while the crank moves through equal angles at different portions of its travel. Similar letters at the crank end show why this occurs.

As the crosshead starts toward the shaft its motion is the sum of the versed sines of the crank and connecting-rod angles ($a_1 + a_2 = a$). After the crank has passed over an angle of 90 deg., the motion of the crosshead equals $c_1 - c_2 = c$. In the diagram $a_1 = c_1$ and $a_2 = c_2$ and the crank positions are taken 60 deg. which the steam travels, or where it is a harmonic of the distance between sharp changes in the direction of flow, there will be periodic vibration set up in the pipe line.

This wave action is entirely independent of the flow of the steam in the pipe line, that is, the wave travels through the steam at a velocity which is sufficient to

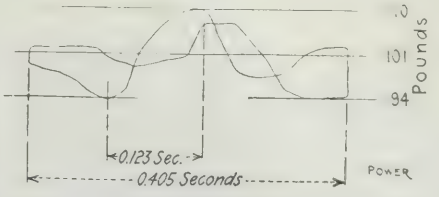


FIG. 3. AMPLITUDE OF WAVES IN STEAM PIPE DIAGRAM EXAGGERATED

cause an increase in pressure of 16 lb. The velocity of the wave can be computed by the following formula:

$$v = \sqrt{182 \frac{Pg}{w}}$$

in which,

v = Velocity with which the wave travels in feet per second;
P = Pressure rise in pounds per square inch caused by the wave;
g = Gravity = 32.2;
w = Weight of the flowing substance in pounds per cubic foot;
182 = A constant.

For the case in hand the following numerical values can be assigned. P = 16 and w = 0.2681, the weight of a cubic

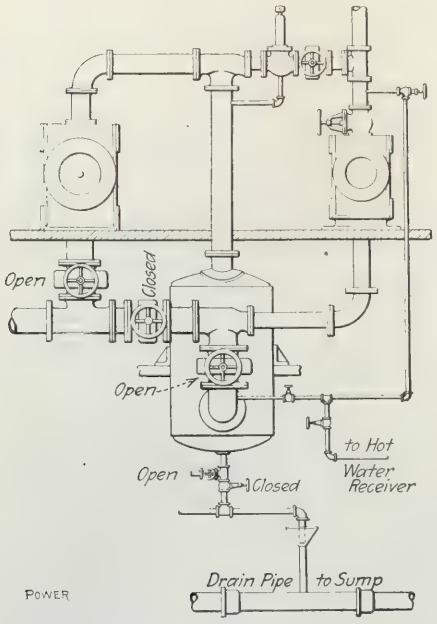
foot of steam at 103 lb. per sq.in. Substituting these values in the equation gives the value of v as 591.4 ft. per sec., which agrees quite closely with the periodic flow shown on the diagrams. The steam pipe is about 50 ft. long and the wave crests are less than 0.2 sec. apart, showing that the wave velocity exceeds 500 ft. per second. In some of the diagrams taken, the periodic rise in pressure was slightly less than that shown and in others slightly more. The diagram illustrated was selected because the wave was more clearly shown than in the others.

A. E. DIXON.

Cleveland, Ohio.

Rising Tide Fills Receiver

When the warming-up valve to the receiver of a cross-compound condensing engine was opened preparatory to starting the engine, a series of water-hammer blows gave warning that the receiver contained water, presumably above the height the warming-up pipe entered the receiver. The discovery led to considerable discussion as to how the water could have entered, and the possible damage which might have resulted if the engine had been started without preliminary warming, as sometimes happens when an unexpected call is made to meet a sudden increase of the load, or other emergency.



CONNECTION TO RECEIVER FILLED WITH TIDE WATER

All sources from which water might have entered the receiver were considered to determine the most likely one by a process of elimination. It was thought at first that the throttle might have leaked and the steam condensed in the receiver, or it might be the warming-up valve that was leaking. These possibilities were rejected as the receiver and the cylinders were found to be practically cold, and the receiver gage registered a partial vacuum of about 10 in., which was accounted for by the air and circulating pumps having been running for several hours.

It was then decided that this partial vacuum must have been responsible for the water being drawn into the receiver, though from what source was a problem yet to be solved, as the receiver drip emptied to the drain from an open-end pipe, the outlet being nearly 4 ft. above the water in the sump-well.

A few hours later the incoming tide solved the problem, it being unusually high and flooding the sump, it backed up in the drain until the end of the drip was submerged and the vacuum in the receiver caused it to be partially filled. Removing the nipple leading into the funnel made a recurrence of the flooding from this source impossible as there was no water seal as before.

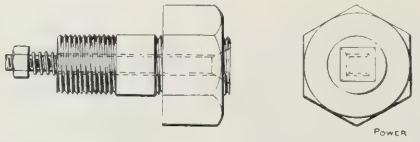
What damage might have resulted if the engine had been started with the receiver partially filled with water can only be conjectured, and I would appreciate learning the opinions of others as to what would have happened.

C. T. ADAMS.

Lynn, Mass.

Safely Follower Stud

The illustration shows an arrangement that has proved effective in preventing a broken stud and nut from dropping between the cylinder head and the moving piston.



CONSTRUCTION OF SAFETY FOLLOWER STUD

The center pin is a loose fit in the stud; and is made square, to prevent its turning around if the stud should break. The stub end should be drilled out as shown. The spring is to take the shock of a break, and also to hold the broken ends from vibrating with the motion of the piston.

R. DALE.

Joliet, Ill.

Insufficiency of Hydrostatic Boiler Tests

From time to time I, and probably many of the readers of POWER, have heard remarks which were not flattering to hydrostatic boiler testing. I have never had much faith in this means of determining the strength or condition of a boiler and some experiences which I have had recently have completely eliminated what little faith I may have possessed.

I am now convinced that cold-water pressure is satisfactory for determining the tightness of new work or for finding leaks which are known to exist, but as a means of determining the strength of an old boiler or what steam pressure a new boiler will stand, or whether or not an old boiler, or a new one for that matter, will be tight or stay tight under steam, I believe this so-called testing is about as useful as a fifth wheel on a wagon.

A short time ago I had occasion to put into commission a locomotive which had not been used for about three years. Shortly before it was laid up it had been retubed. Every "get-at-able" place was examined and sounded with a light hammer. Some corrosion was found, especially around the water leg; the tubes looked fairly good. I then connected up a hand pump and pumped up 200 lb. pressure, which was held while another inspection was made. Everything appeared to be satisfactory and the boiler was fired up, and, with the safety valve set at 125 lb., started to work on comparatively light hauling. The second day a tube developed a hole somewhere between the heads and leaked so badly that it was necessary to quit work and plug it. After this other tubes began to leak and it was ascertained that the tubes were in an unsafe condition.

On another occasion I did some experimenting on a 16-in. tubular steel automobile boiler having ½-in. tubes. This boiler appeared to be in good condition. Cold-water pressure was pumped up to 500 lb. on two occasions several days apart and the boiler was as tight as a bottle, but when 200 lb. steam pressure had been raised a superheater tube let go, and as the rear wheels were jacked up, the engine cut loose at a pace that drove all the "kids" and chickens to cover that were in the vicinity. With this boiler the steam after passing the throttle is carried down through a boiler tube to the firebox, then passes through a section of the superheater and up through another boiler tube, or so-called superheater tube, and then repeats this operation through another set of tubes and section of superheater. That makes it possible for any one of four tubes to fail and discharge the contents of the boiler into the engine with no means of shutting it off except possibly by putting the link motion in the central position. I cut the damaged tube out and found it corroded nearly through on top and also a small pit hole through it which had started the trouble.

A new tube was put in and a few trips made with the car, always expecting a little extra fun at any minute, of course, but nothing happened until one day the old "kettle" was fired up and when the gage showed about 50 lb. pressure the

bottom apparently dropped out of it. An examination showed water spurting from holes in four tubes.

These were plugged in approved automobile style and the experiments continued by pumping the pressure up to 500 lb. and holding it a short while for two or three times a day for several days, the object being to give the cold-water test all the chance there was to find any weak places which might exist. However, there were none for it remained as tight as a new boiler could possibly be. A slow fire was then started and the pressure brought up very gradually and when the gage showed about 225 lb., whish! went another tube. I am not exactly a novice in matters pertaining to steam and boiler inspection, and I should not recommend hydrostatic tests to reveal what an engineer wants to know of the condition of a boiler.

H. L. Strong.

Portland, Maine.

Another Experience with the Central Station

Recently I visited an engineer and he confessed his fear of the central station, but knew he could compete with it if the owner would coöperate with him.

On examination I became convinced that the solicitor was figuring him out of a job. The two boilers were fed by injectors. The engine which developed about 80 hp. exhausted direct to the atmosphere. Live steam was used for drying purposes and also for heating when it was required. The engineer said he had been "turned down" every time he mentioned using the exhaust steam. The owner came into the engine room and I was introduced to him and soon led the conversation to central-station service, which he said he contemplated using. I inquired as to the gain he expected and he replied that it would do away with one boiler and the other could be fired with sawdust and shavings (a byproduct) and he could sell the butts and slabs for wood; that he would not have to pay an engineer, and would save the price of about 125 tons of coal a year which was then required.

I pointed out to him that by using valves and piping he could heat the kiln and factory with the exhaust steam, and with a pump and heater to use the returns for boiler feed, could save the coal bill just the same; that he would have to pay an oiler to look after shafting, and that I thought he could do without one boiler.

"Why!" said he, "the engineer has been telling me that, too!"

I referred him to an engineer whose specialty is steam heating and got him to promise he would consult him. I thought little more about it, until this owner wrote me saying the new system was installed, that they were using but one boiler through the summer and might be able to run with one all the time.

V. C. Wood.

Copenhagen, N. Y.

Common Trouble of Siphon Condensers

Whenever a valve breaks in a water pump supplying a siphon condenser, it usually follows the water up and lodges between the cone and the casting, as shown at *A*, Fig. 1. Impurities, small eels, clam shells, etc., have also been found blocking the space as to seriously impair the vacuum.

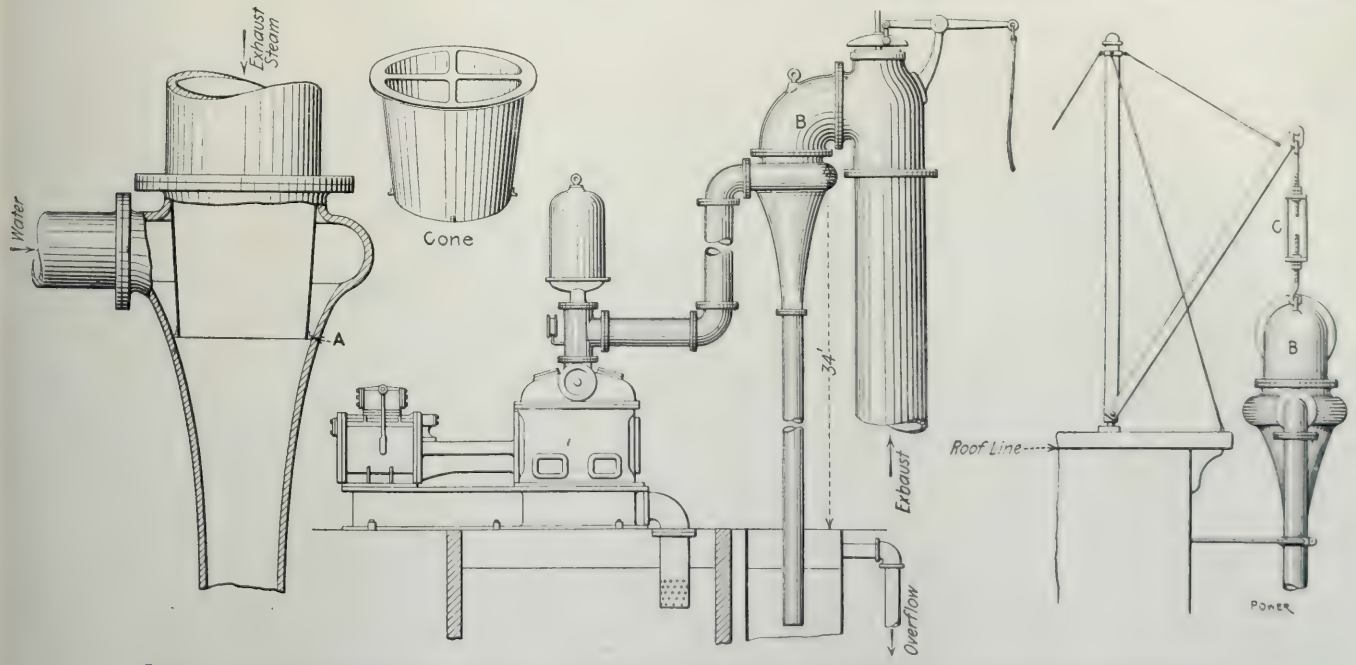
Under this condition of stoppage, a pump becomes overworked to a point above safety. Generally 3 or 4 lb. is registered on the pump gage, under normal conditions, but when the opening is badly plugged, 16 to 18 and sometimes 20 lb. is required to force water through the condenser, and consequently the strain is plainly visible on the pump.

In some places running but 12 hours, the day can pass without interruption, and at 6 p.m. the condenser may be easily cleaned. This is done by removing the bolts and raising the elbow *B*, Fig. 2, then pulling up the cone shown in Fig. 1. If the roof of the mill or shop is higher than the condenser, the elbow can be raised with block and tackle. If lower, the arrangement shown in Fig. 3 will do very well. The elbow is lifted by turning the buckle *C* with a short bar; it is then swung aside. The cleaning consists of hauling the cone out with both hands. At the instant it is raised, the obstructions will be seen chasing down the water-leg and into the well. The cone may be set aside and, if any stuff remains it can be brushed away with the hand. Extreme care, however, should be exercised in replacing gaskets and making joints. Gaskets should be painted with graphited oil and then the bolts tightened cautiously.

An improvement has been made of late years which facilitates this whole operation. It consists of a geared mechanism on the outside of the elbow. And by turning a wheel the cone is raised. The cleaning can thus be performed with greater ease and at any time when running.

Luke Marier.

Fall River, Mass



Parts and Arrangement of Syphon Condenser and Means for Removing Elbow to Clean Cone

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

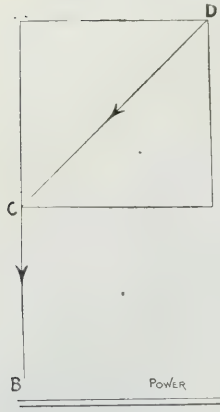
Test of Axial Flow Centrifugal Fan

In the June 25 issue a letter appeared by me relative to an axial flow turbine fan. Those interested in this fan will probably find the following test results of value:

TEST OF AXIAL FLOW CENTRIFUGAL FAN

Intake Side						Outlet Side						Input					
Stat. suction at center	Velocity pressure at center	Air velocity from reading	Average velocity over pipe	Capacity cu.ft. min.	Hp. air	Velocity pressure	Air velocity from reading from reading	Hp. air	R.p.m.	Rim speed	Ratio $\frac{\text{vel. rim}}{\text{vel. entry}}$	Ratio $\frac{\text{vel. rim}}{\text{vel. delivery}}$	Volts	Amps.	E.Hp.	B. Hp.	Mech. Efficiency as an Exhauster, Per cent.	Mech. Efficiency as a Blower, Per cent.
$\frac{1}{8}$	$\frac{1}{8}$	1394	1270	725	0.0142	$\frac{1}{4}$	1971	0.0284	1100	2735	1.96	1.39	46	1.4	0.087	0.057	25	50
$\frac{3}{16}$	$\frac{3}{16}$	1707	1560	890	0.0262	$\frac{3}{8}$	2415	0.0525	1320	3283	1.92	1.36	58	1.65	0.128	0.084	$31\frac{1}{4}$	$62\frac{1}{2}$
$\frac{1}{4}$	$\frac{1}{4}$	1971	1790	1020	0.040	$\frac{1}{2}$	2789	0.08	1540	3830	1.95	1.38	75	2.05	0.206	0.136	$29\frac{1}{2}$	59
$\frac{5}{16}$	$\frac{5}{16}$	2204	2000	1140	0.056	$\frac{5}{8}$	3118	0.112	1760	4377	1.99	1.41	90	2.35	0.284	0.188	$29\frac{3}{4}$	$59\frac{1}{2}$
$\frac{3}{8}$	$\frac{3}{8}$	2415	2200	1250	0.0745	$\frac{3}{4}$	3417	0.148	*1870	....	....	....	102	2.7	0.369	0.243	$30\frac{3}{4}$	$61\frac{1}{2}$
$\frac{7}{16}$	$\frac{7}{16}$	2590	2350	1340	0.0925	$\frac{7}{8}$	3690	0.184	*1980	....	....	....	111	2.8	0.417	0.275	$33\frac{1}{2}$	67
																	Average 30%	60%

*These r.p.m. are evidently too low.



PARALELLOGRAM OF FORCES

AB = Rim velocity of wheel

AC = ½AB = speed of vel. center

AD = AC = ½AB = vel. of entry of axial flow.

CD = resultant or velocity of delivery = $\frac{AB}{1.414}$ since paralellogram is a square.

Description

Wheel 9½ in. diameter. Inlet pipe 10¼ in. diameter, six ft. long with flared mouth giving effective inlet equal to that of wheel (62 sq.in.) Area pipe 82 sq.in. or 0.57 sq.ft. Average velocities in pipe taken as 0.91 of that of center. Readings taken with two Pitot tubes at center of pipe, the static and dynamic tubes being connected to each leg of a *u*-tube, thus reading velocity pressure. Readings are in inches per sq.in. of water.

Motor ¼ hp. 110 volts series wound, direct connected with flexible coupling. Prony brake test showed motor efficiency to be 66 per cent.

Mech. efficiency of wheel given above includes two babbitt bearings. At 5500 ft. rim speed it is silent running.

About 5200 ft. was the highest peripheral speed obtainable with the motor used; at 8000 ft., which is not excessive with this type of wheel, the air velocities on each side of the wheel would correspond to 1 in. and 2 in. of water, respectively, while handling 2000 cu.ft. of air per min.

C. O. BRETHERICK.

Seattle, Wash.

Priming Pipes

R. T. Rezniem's letter in the Sept. 10 issue describes a practical arrangement for priming a pump. I would have appreciated a longer treatment of the subject than given by Mr. Rezniem.

When pumping liquids from a well or tank below the pump, the liquid will run back into the well when the pump is stopped if air leaks into the suction or the discharge chambers. If the discharge valves seat improperly the liquid will leak through them and also through the suction valves if they are badly worn or cracked. Air then fills the space formerly occupied by the water.

If there is a column of water in the discharge pipe and the discharge valve is opened the discharge chamber is immediately filled with water and the pressure due to the head holds the discharge valves so tight against their seats that no water gets to the pistons and suction valves to keep them water-sealed and air-tight. The piston creates a very low vacuum by its displacement in moving away from the end of the cylinder; it will also compress the air in front of it against the discharge valves. The air must be compressed to a pressure equal to and above that in the discharge pipe to open the discharge valves, but it will follow the piston and expand and again be compressed as the piston moves back and forth. This low air pressure in the cylinder only aids to keep the suction valves closed and the pump becomes air-bound.

The water is surging back and forth through the priming pipe as it is discharged from the cylinder, but with the air cock open, the air is liberated. Water passing from the discharge pipe to the suction-valve chamber would keep the valves sealed air-tight and thus exclude the air until a sufficient vacuum had been created in the pump for the atmospheric pressure on the water to force the later up into the pump. Then the liquid, being a solid body would, when pushed by the piston, force open the discharge valves and the pump work properly.

R. A. CULTRA.

Cambridge, Mass.

Power for Salt Works

The article on this subject by Mr. Saxe, page 219, Aug. 13 issue, is very interesting and recalls my experiences with such plants.

From his descriptions of the grainers, I believe he could have returned the condensation to the boilers without such a great temperature loss. Three or four per cent. CO_2 is not unusual in such plants, in fact, it is quite common in plants where the firemen are greatly underpaid. If Mr. Saxe could have instituted a bonus system for the firemen it might have brought excellent results.

Mr. Saxe said induced draft would have been more economical and produce less smoke than the forced draft used. In my opinion forced draft would prove better than induced draft. With forced draft there should be a higher percentage of CO_2 than with induced, and a lower stack temperature, if the inside of the boiler is free from scale and the outside free from soot.

Plant designers seem to have a ten-

dency to provide too large a grate area. Often I have found the grates so large that it was only possible to burn about 10 lb. of coal per hour per square foot of grate. Mr. Saxe could have dispensed with a number of boilers, burned more coal per hour under those in use and obtained a high percentage of CO_2, and undoubtedly a higher efficiency, if the settings were good, and the boilers clean inside and out.

Another factor in boiler efficiency often overlooked is the relation between the chief engineer and the fire-room force. The fireman should be taken into the confidence of the chief. Let the chief teach the fireman how to fire, then pay him what he is worth.

G. F. DUEMLER.

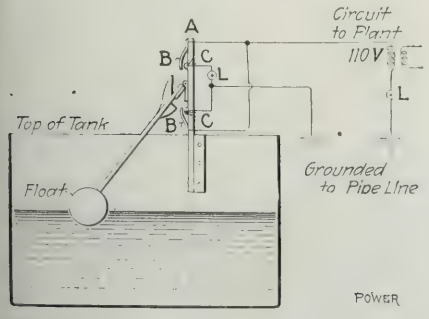
Philadelphia, Penn.

Water Tank Signal System

The recent letters and illustrations on this subject have interested me very much. We have had a signal system in operation for several years that has given good service.

The present water-system reservoir is not high enough to supply a section of the city, so a pump was placed in the power station and takes water from the mains and forces it to a tank about 1½ miles distant and high enough to supply the high section. As the pump was only required to run a part of the time a signal system was required to tell when the tank was empty or full. The system shown was worked out and installed.

A is a piece of timber placed in the tank in a vertical position. *BB* are strips of spring brass, and are forced in contact with plates *CC* by the float lever. A single wire is strung from one side of a 110-volt circuit at the station and connected as shown. The other side is connected near the pump to the discharge line with a 110-volt 16-cp. lamp in circuit, and contacts *CC* are connected to the same pipe line at the tank with a lamp near the upper contact.



MECHANISM OF HIGH- AND LOW-WATER ALARM FOR TANK

When the tank is almost empty the float drops and the lever causes the lower spring *B* to make the contact with *C*, completing the circuit, and the lamp *L* will burn at full brilliancy. When the tank is full the float lever causes the upper spring *B* to make contact and complete the circuit through the lamps *L* and *L* and the lamps burn at half brilliancy.

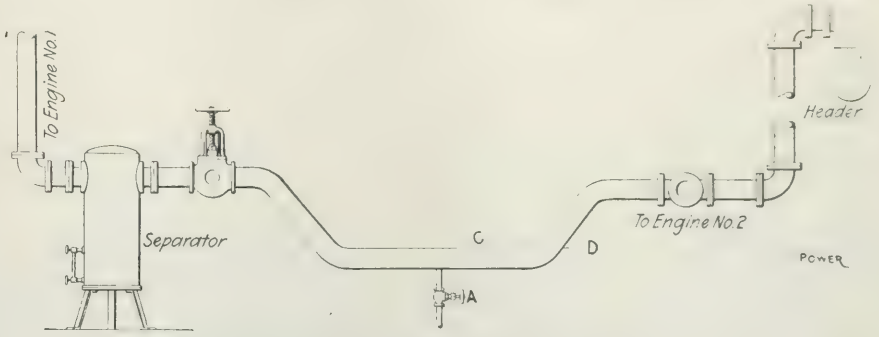
The float lever is made fast to piece *A* by a hinge. The system has given very little trouble and has the advantage of giving two different signals without switches.

C. E. NIGH.

Morgantown, W. Va.

Faulty Pipe Design

J. W. Parker's letter on this subject in the July 30 issue reminds me of a piping layout arranged similar to that shown below. The drop in the pipe was put in to avoid a large exhaust main. The drain *A* had not been connected to a trap. The installation was a new one and engine No. 1 had not yet been started, and the stop valve at the separator had not yet been opened. Engine No. 2 had been running for a few days, and the pipe was therefore full of condensation up to the stop valve. Apparently no water-hammer had occurred to warn anybody, although the straight run at *C* must have measured at least 10 ft.



SHOWING WATER POCKET IN STEAM MAIN

One night, steamfitters were working near the drain and for some reason opened the valve. Violent water-hammer occurred soon after the valve had been opened. I should say it occurred when the water reached the level *D*, and began to run back from the stop valve shown. The pipe was broken and twisted, and the separator upset. One steamfitter was badly burned; the others escaped through the nearest exit.

R. MCLAREN.

Govan, Sask., Canada.

Water-Sealed Flue-Gas Collector

In the water-sealed flue-gas collector shown on page 269 of the Aug. 20 issue, it would appear that the collecting cylinder is a little too large for the water holder. With the various dimensions given the water in the apparatus now stands almost in equilibrium, which means that all the lower cone is of no value as a part of the collecting apparatus. An overflow vent might be placed about 5 in. from the bottom of the outer vessel, which would allow the water to come finally to rest at that level, making all the inside of the collector available.

Mr. Barnum does not speak of one point I have found important. The water used in a collecting system of this sort should not be fresh. Water has the power of absorbing nearly its own volume of CO_2, and if fresh water is used in an apparatus of this sort, when the sample is to be tested later for CO_2, the results are invariably low. The reverse of this also seems unfortunately true, i.e., if water saturated with CO_2 is used to collect gas, which is low in CO_2, the water will give up a little of the carbon dioxide to the collected gas.

However, most flue-gas work is done on gases which do not vary greatly, and I have found it gave the best results to use the same water for collecting the gases over and over, until it became too physically dirty to use any longer. This applies chiefly when analyzing for CO_2, but slightly to CO, the latter being comparatively insoluble. The power engineer rarely is concerned with the oxides of sulphur. If it becomes necessary to investigate their proportions, he may as well give up any idea of attempting to collect the sample over water; some absorption process must be used.

DONALD M. LIDDELL.

Elizabeth, N. J.

Confessions of an Engineer

Mr. Rogers' article, "Confessions of an Engineer" in the Aug. 20 issue reminds me of an experience I had with a superintendent.

The building was an old one and badly in need of repairs and the shafting was not unlike that mentioned in the article. It had not been lined up since it was installed. One could stand anywhere in the room and see that it was way out of line. We used automatic grease stick lubricators and some of the bearings ran so warm that a stick would not last half a day.

The superintendent believed in letting

things run as long as they would, regardless of their conditions, and when I asked for a helper to align the shafting he replied that I could not do anything with it, that the building had settled and as long as it ran, not to bother with it.

I was not satisfied to think that way, so tackled the job alone that afternoon, using only a spirit level and line. In some places I had to raise the shaft about ⅜ in. and nearly as much sidewise in other places. The work was not perfect, but it helped so much that I was not bothered with hot bearings during the remaining six months I stayed there.

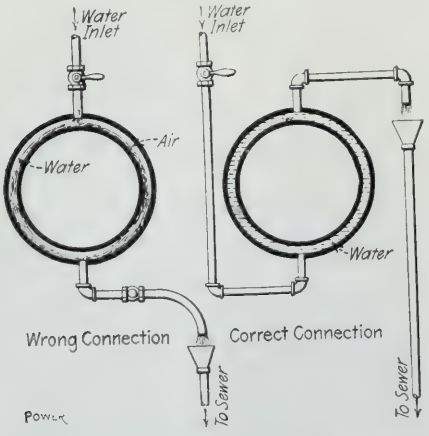
Another experience was with a millwright when I wanted to change the belt on a small dynamo. The one in use was about worn out and I had selected as its successor a belt that had been run in a dry place with no attention and had cracked considerably. He claimed it would never do the work and I might as well get a new belt first as last. Nevertheless, I had him make the joint on Saturday and I left the clamps on until Sunday and gave it a coat of good dressing. I followed this up with a lighter coat when the belt had absorbed the previous ones, and was never bothered with the belt not doing its work properly. He was quite surprised at the condition of the belt when we removed it a few months later, having no more use for that dynamo.

G. B. Longstreet.

Somerville, Mass.

Value of a Sketch

Soon after arriving in this country, I secured a position as general mechanic with a Canadian railroad. I could not speak English very well so experienced trouble in connection with my work.



Jacket Connections to Air Compressor

As mentioned in the editorial and also Mr. Dixon's letter under the above headings in the Aug. 13 issue, I found my ability to sketch very valuable in conveying ideas.

On one occasion I was sent to correct the trouble in an air-compressor cylinder which was always running hot. The foreman was very upset over the case as a man had been working on it for three days and left the job with the cylinder in the condition he found it. The water connections were wrong and after a half hour's work I changed them and the trouble ceased. I had considerable difficulty in explaining how the job was done, so I drew the sketch shown so he could understand what I tried to explain. He was so pleased he told me to take the rest of the week off and go fishing.

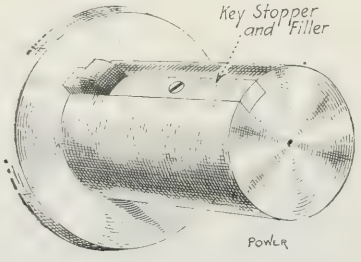
J. G. Koppel.

Montreal, Que., Canada.

Safety Pin for Flywheel Keys

A. L. Haas' letter under this heading in the Sept. 3 issue interests me. While the job illustrated and described will undoubtedly prevent the key from coming out, it does not appeal to me as a good job. It is, as many old mechanics express it, "painful to the eye." It looks clumsy and awkward, but appearance is not always considered so long as solidity is obtained.

A solid job can often be had and at the same time the appearance made desirable. It does not seem reasonable to drill a comparatively large hole clear through the end of a shaft, and fit a correspondingly large taper pin in it to prevent the key from backing out. A better plan would be to fill the unused part of the keyseat with a neatly fitted false key, as shown. This will at once act as a stopper and filler, and preserve the symmetry of the shaft end. I have seen this done on large engines, and no trouble was ever experienced from the key coming out.



A "Mechanical" Way of Fitting a False Key

Charles J. Mason.

Scranton, Penn.

Paint for Coils

In answer to B. A. Warren, who asks for a paint for coils in the Aug. 13 issue, I would advise that a good, serviceable paint for coils used in refrigerating work, smokestacks or any iron work can be made from 5 gal. coal tar, 1½ gal. paraffin oil and 3 pints of Japan dryer. Mix and stir until the coal tar has cut the paraffin. This will take from 5 to 10 min. This paint will outlast asphaltum when it comes in contact with water or brine.

E. F. Tracy.

Mechanicsville, N. Y.

Shorter Cutoff Greater Load

Mr. Stewart, writing on this subject in the Sept. 10 issue, has not made his argument as clear as he might, although he is correct in most of his statements.

Relative to a shorter cutoff in the low-pressure cylinder causing a loop in the high-pressure diagram, he might get a negative loop in this diagram on no load and still have the valves set so as to cut out the loop at half load. All compound engines are not governed alike, but as Mr. Stewart says, the load can be shifted from one cylinder to the other by changing the low-pressure cutoff, but he neglected to state that the power of the engine remained the same. The indicator will show that a change in the receiver pressure, due to manipulation of the low-pressure cutoff will change materially the amount of work done in each cylinder, although it will leave the total work done by the two cylinders the same as before.

So if the receiver pressure is lowered by making the low-pressure cutoff later the back pressure obviously will be less in the high-pressure cylinder and the mean effective pressure will be more, so that the high-pressure cylinder will now be doing more work. Since the total work done remains the same, the low-pressure cylinder in spite of its later cutoff will now be doing less work.

Conversely if the receiver pressure is increased by making the low-pressure cutoff earlier, less work will be done in the high- and more in the low-pressure cylinder. To change the power of the engine the high-pressure cutoff must be changed, while to distribute the work in the two cylinders the low-pressure cutoff must be altered. The governing mechanism of many compound engines operates on the high-pressure cutoff, but this is open to the serious objection, that when the governor makes the high-pressure cutoff later in responding to an increase of load, the larger share of the work will be done in the low-pressure cylinder, and when the governor causes cutoff in the high-pressure cylinder earlier, the larger share of the work will be done in the high-pressure cylinder and at very early cutoffs the low-pressure side may then actually be a drag on the engine. To overcome this many compound engines have the governor operate the cutoff on both cylinders. Then the governor operates the high-pressure cutoff to change the power of the engine and the low-pressure cutoff to equalize the work done in the cylinders.

C. F. Robinson.

Iowa City, Iowa.

Making a Concrete Engine Foundation

By H. L. Strong

Directions for making the form and proper mixture for a concrete engine foundation. Engineers have to do this work sometimes and the article is worth filing for such an emergency.

Matters pertaining to concrete may not be considered as within the province of the steam engineer, but it is well to be able to do a little concrete work as occasion may require.

For example, the boss inquires as to whether the engineer can install a new direct-connected set, and also the material he considers best for the foundation. Concrete is recommended as it is generally used and costs about $5 per yard in small quantities, and makes a very firm foundation, whereas brickwork costs from $1 to $2 more and is no better, if it is as good. The employer is pleased to have a man who can care for the installation of an engine from the foundation up. He can give this man the foundation plans and it may be completed when the engine arrives from the makers.

As there is usually a heavy concrete floor at the place where the foundation is to be, a few steel bars like that shown in Fig. 1 should be made. A couple of 6- or 8-lb. striking hammers, and four men should now be put to work cutting out the concrete floor, one man holding a "point" and another striking it. After a hole is started the floor may be torn out quite rapidly. When the hole is cut to the required size it should be trimmed up to the lines, so as to make a smooth job of patching in after the foundation is completed. These tools are also used for cutting mass concrete, and should be kept sharp for easy and rapid work. Made round as shown, they are much superior to the square or chisel-shaped point commonly used, as there is less breakage and sharpening needed and they penetrate much easier.

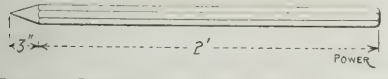


FIG. 1. POINT FOR CUTTING CONCRETE

Having cut out the floor and excavated to rock, it is now in order to compute the cubical volume of the foundation, which it will be assumed is 300 cu.ft. This divided by 27 gives 11⅑ cu.yd. The mixture should consist of, in masonry parlance, 1, 2½ and 5, which means 1 part cement, 2½ parts sand and 5 parts crushed stone. As the cement and sand make about mortar enough to fill the voids in the stone, 12 yd. of stone will do (allow something for loss in handling, etc.) Half as much sand, or 6 yd., is needed and as this mixture takes about 1½ bbl. of cement per yard, 18 bbl. are required. If the cement comes in bags, four bags equal a barrel and 18 × 4 = 72 bags. A bag holds 1 cu.ft.

A mixture of 1, 3 and 6 also makes good concrete for mass work and does not take quite as much cement. For very particular work 1, 2½ and 4 or 1, 2 and 4 is frequently used. "Fine stuff" for finished surfaces is made of 1 part cement and 2 parts sand, the sand should not be too coarse. As the cement fills the voids in the sand, a yard of sand makes about a yard of mortar. The engineer should now estimate the material needed by weight, as that is the way it is usually sold. Sand weighs 100 lb. per cu.ft.; therefore

$$27 \times 100 \times 6 = 16{,}200 \text{ lb.}$$

or 8⅛ tons is required. Crushed stone weighs 90 lb. per cu.ft., and

$$27 \times 90 \times 12 = 29{,}160 \text{ lb.}$$

or 14.58 tons will be needed.

A finished surface is not necessary on the foundation as it will be out of sight; otherwise the form should be made of matched and planed spruce boards. Good square-edged boards will do and they are cheaper. The number of square feet of 1-in. boards and linear feet of 2x4-in. joist required to build the form should be determined. Add enough to this amount to construct a 10x10-ft. double-boarded platform for a mixing-board.

Old or stale cement should not be used. To test for its quality, mix up a sample and if it does not set firm enough in 24 hours to be handled without crumbling, and so a knife blade cannot be pushed into it, it is not fresh. The test is rather crude, but after a little experience one can judge correctly as to whether or not concrete is setting properly.

Cement sets best in a moderate temperature. Too much heat is not desirable and any cement sets slow in cold weather. Clear, "sharp" sand and clean, crushed stone which has been properly screened should be demanded, as good concrete cannot be made with poor cement or soft, dirty sand and "any old thing" for stone. Soft sand or even a small amount of earth in the material will prevent the cement from setting properly. The writer has seen concrete made from such material that could be kicked to pieces after it had set several days. Such concrete is worthless for any purpose and is the cause of much adverse criticism of concrete in general. If proper materials were always used in making concrete, there would be few cases of its failing to prove satisfactory, unless from improperly designed structures or removing the false work too soon.

Make up the broad sides of the form by cutting the 2x4-in. joists into lengths an inch or two shorter than the height of the foundation. There should be enough pieces to allow for spacing them 18 or 20 in. on centers. To do this, lay them down on edge, mark the spacing on a board and nail it to the 2x4 joists with 6-penny wire nails, and continue to nail on boards until the required width is attained. The ends of the boards are then marked and sawed to the required length, and as the bottom of the foundation is wider than the top, the cut has to be made at an angle. Make up these pieces until there are enough to form a box just the size and shape of the foundation, the boards on the top extending an inch or two above the joists. Now cut two pieces of 2x4 for each side of the box to be used for "purlins" and nail them across the studding. One of these sides is shown in Fig. 2. The pieces are then assembled in their proper places and the form is squared. The next step is to solidly brace the form from the outside by cutting 2x4 shores which fit tightly between the purlins and boards laid against the sides of the excavation.

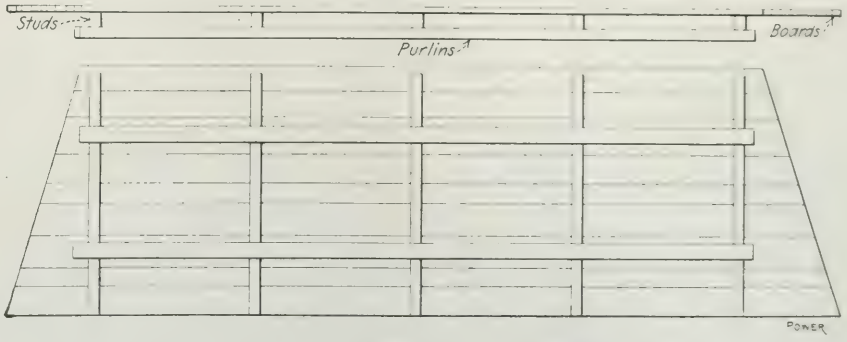


FIG. 2. DETAILS OF A SIDE FOR A FOUNDATION FORM

The form can now be lined up by driving wooden wedges around the shores. It should next be accurately leveled, as it is both troublesome and expensive to

patch up defects in the concrete due to inaccuracy of the form.

Fasten pieces of board across the top of the form, on the lines of the foundation or holding down bolts. Carefully locate the positions of the bolts on the boards and bore holes in the boards just large enough to take the bolts. Bolts of the required diameter and 2 or 3 ft. long, having square heads and heavy washers or a few inches of the end turned at right angles are then hung through the holes, the nuts on them being screwed down enough to keep the ends above the false work an amount equal to the thickness of the nuts, washers, bedplate and to allow for grouting under the bedplate. Pieces of pipe, larger than the bolts and a foot or more long, should be slipped over the bolts and fastened up under the boards or template instead of cementing the whole length of the bolts in solid, as this will prevent trouble if the bolts do not enter fairly when the machine is set in place.

Next make a bottomless box 3 ft. by 3 ft. 4 in. by 1 ft. = 10 cu.ft. This box is placed on the mixing board and filled level full of sand and then lifted up, leaving the sand on the board. One barrel (4 bags) of cement is then dumped on the sand and the two thoroughly mixed together while dry by turning over two or three times with square-end shovels. The mixture is then roughly formed into a basin and three or four pails of water poured in and the dry material worked over into the water, and spread out. The bottomless box is then filled twice with crushed stone which is dumped on the mortar and the whole mixed thoroughly by shoveling it over three or four times and adding water enough to make it slightly sloppy or so that it will run off a shovel or out of a wheelbarrow without much scraping. The old method of barely wetting the mixture and then pounding it into the form is obsolete, as concrete mixed in that manner cannot be as dense or strong as by the new method of running it into the form. An excess of water is also detrimental as it washes the cement out of the mixture. Dump the mixture into the form, but remove the inside braces as the form fills. A flat spade is also worked up and down next to the form to push the stone back and let the mortar fill in smoothly along the face. This is especially necessary if a smooth surface is desired.

This operation is continued until the form is filled to within 2 or 3 in. of the top. The concrete is then allowed to set enough so the template can be removed without throwing the bolts out of line. Some 1-and-2 mortar is then prepared and the form filled full and "struck off" by working a straight-edge across the top, after which the surface should be smoothed with a wooden or steel "float" (large, flat trowel). In three or four days the false work can be removed. The space around the foundation is then filled in, the filling tamped down solid, and the floor patched in. It is well to allow the block to set for about a week before setting the engine on it. After the engine is on the block, it may be shimmed up with pieces of sheet iron or wedges, if this is necessary to level it.

Now mix some cement and make a dam 2 or 3 in. high all around the base, after which mix some "half and half" grout and pour it in around the base. Work the space full with pieces of thin, flat iron. Pouring is continued until the dam is also full, which gives head enough to overcome the shrinkage as the grout sets. Before the grout and dam have set hard the cement is neatly trimmed off with a trowel, and in three or four days the nuts on the holding-down bolts may be set up.

Design of Surface Condensers

By W. H. Booth

A suggestion that the tubes of a condenser be spaced wide in proportion to their nearness to the cylinder, the passageway for the steam diminishing as the steam diminishes in quantity. This would tend to produce a uniform vacuum throughout the condenser.

In designing surface condensers the main object generally seems to be to put the largest possible surface in the direct path of the steam irrespective of any consideration for the passage of steam beyond the first row of tubes. About the only concession in this direction is the omission of a group of tubes so as to produce a gap of V-form in the general mass of tubes. These gaps serve to present more tube surface on the one hand, while on the other they provide more passageways by which the steam can reach the interior of the nest of tubes.

Yet the modern steam turbine is found to be so arranged as to provide a pipeway to such a blocked condenser, the full diameter of the turbine rotor. If so great a diameter of this detail be desirable, it is not consistent that the passageway between the condenser tubes should be so much reduced. Experience also points to the error, for the vacuum differs by as much as an inch between the steam entry and the base of the condenser. Except for considerations of cost and space, there is no reason why a condenser should not be many times its ordinary cubical capacity with widely spaced tubes dispersed throughout the space. If steam be regarded in the light of the kinetic theory of gases, the molecules of which are darting about at a velocity of perhaps 1800 feet per second, is it not obvious that an extra few feet of length or breadth will have little effect upon the time required by each molecule to find its way to a cold tube surface? The ideal condition would be that each molecule touch a cold surface which would kill its self-translatory activity and drop it to the condenser base as so much dead water.

Every tube carrying a stream of water possesses a certain capacity for absorbing heat from the molecules which pass upon it. But if too many molecules strike the tube in a given time the tube surface will become hot for there is a limit to the rate of heat conduction through the walls of the tube. From the hot surface the steam molecules will rebound without having given up much of their heat energy and in the rebound they will strike against the oncoming stream of molecules from the cylinder, driving these back to some extent or at least hindering their flow. Undoubtedly the major portion of the work of a surface condenser is accomplished in the first row of tubes, and the result is this reflex action of many of the molecules which ought never to have hit any of the first row of tubes but should have passed forward to the second, third and subsequent rows. What seems to be needful is that only one tube in three of the first row should remain, and this tube should be fitted with a light casing of thin metal with lips placed vertically toward the flow of steam. In cross-section this casing would resemble the letter *A* with an open top. The tube would occupy the lower division of the letter which would rest upon the tube by means of an occasional cross bar. The top slot would have a breadth of, say, a fourth or a third of the tube diameter, thus reducing the number of molecules which reach the tube surface and therefore keeping down the temperature of that surface so that it ceases to be a surface from which molecules can rebound with any great velocity. In this way the amount of steam dealt with by the first row of tubes is reduced to about one-ninth or one-twelfth of the old arrangement. The next row of tubes would be similarly spaced and fitted and so also the third or even the fourth rows; beyond that the spacing would become closer and

the covers made with wider slots or even omitted, and so on to the base of the condenser, where the tubes would be closely spaced.

The provision for wider spacing is particularly desirable where the contraflow principle is employed, for if this is fulfilled the first tubes touched by the steam are the hottest and create the maximum of reaction. Since the temperature of saturated steam is fixed for any given pressure there can be no temperature difference without a pressure difference, and the contraflow principle presupposes such a pressure difference and indicates a loss of vacuum in the cylinder. It seems more rational that the first rows of tubes should be as cold as possible and that the last rows should be cold also. Any marked rise of temperature of the circulating water should be in the middle tubes where most of the steam has disappeared and the passageway is ample for what is left. There appears to be no difficulty in designing a condenser on these lines to provide for intercepting the water condensed in each division so that it shall not drip on the tubes below. Summarized briefly, the suggestion is that the spacing of the tubes should be wide in proportion to their nearness to the cylinder and that the passageway should diminish as the steam diminishes in quantity.

If a 2-inch vacuum is attainable under certain conditions and there is an absolute pressure of 3 inches at the top row of tubes, the steam volume per unit weight will be 50 per cent. greater at the lower tubes and should be provided for accordingly. But such a loss as 1 inch is what the suggested design aims to avoid by the provision of ample space for the passage of the steam. In practice it appears probable that such a design would culminate in condensers having a maximum of passage area transverse to the path of the steam, so that the condenser would be long and wide in order that the first rows of tubes should still possess considerable surface even though the tubes were not allowed to exceed a given fraction of their full duty, as provided for by the cover sheds.

Steam is perhaps best distributed over the upper row of tubes by providing for its entry above the tubes and parallel with them in a wedge-shaped chamber extending the length of the tubes. If admitted to the central part of the area of the upper row of tubes it should be by a tapering entrance or a reversed hopper springing from both ends of the tubes and fitted with diverging plates to spread the steam throughout the length of the upper row. Baffle plates across the path of the steam are bad practice and serve to cause obstruction. It is better to guide the steam than to compel it to spread by the interposition of a baffle.

The ideal condenser is, of course, a large open chamber with cold walls, into which steam is admitted and speedily finds one of the six cold walls. A large cubical room with about 45 feet square on each wall would give only the 12,000 feet of surface regarded as necessary for a tubular condenser occupying a mere fraction of this volume. Even cellular, flat internal walls would diminish the volume very little and it is not easy to escape from the tubular design. But this has been overdone to the extent that tubes have been spaced too closely to permit the vacuum formed at the base to act upon the cylinder.

The Sulphur Contents of Petroleum

The Bureau of Mines has just issued a bulletin describing different "Methods of Determining the Sulphur Contents of Fuels, Especially Petroleum Products," by Irving C. Allen and I. W. Robertson. The paper describes in detail eight methods of sulphur determination and the conclusion of the authors is that the best method is that of burning the fuel in a calorimetric bomb with pure oxygen under a pressure of 30 to 40 atmospheres. This method is accurate, practicable and rapid and is recommended in preference to all of the other methods described. The calorimetric determination if desired can be made at the same time. Copies of the paper can be obtained by addressing the Bureau of Mines at Washington, D. C.

Automatically Operated Valves

The Golden-Anderson Valve Specialty Co., Pittsburgh, Penn., manufactures a line of automatically operated valves. That shown in Fig. 1 is a double-cushioned, triple-acting, nonreturn valve, designed to automatically protect boilers in case of a bursting tube. It will also act as a safety stop and prevent steam entering a cold boiler.

The dashpot area of the upper part of the body insures a cushion in the opening and closing of the valves and the alignment with the seat regardless of position.

The dashpot *A* is attached to the valve spindle; the outside of the dashpot acts as a bronze cylinder and is held firmly in place by cap bolts.

A bypass, admitting full boiler pressure above and beneath the inside dashpot, shown at *C*, balances it. The steam connection from the main steam line to the automatic pilot valve holds it closed, because of the large area above it.

Between the two dashpots is a steam-pipe connection leading to the pilot valve and if rupture occurs in the steam lines, pressure is removed from the top of the pilot valve, allowing it to open instantly, exhausting the pressure from between the dashpots *A* and *B*. The full boiler pressure always being above the dashpot, the valve will close at the same time being cushioned in the operation.

This valve can be opened or closed by hand, by raising the small lever *F*,

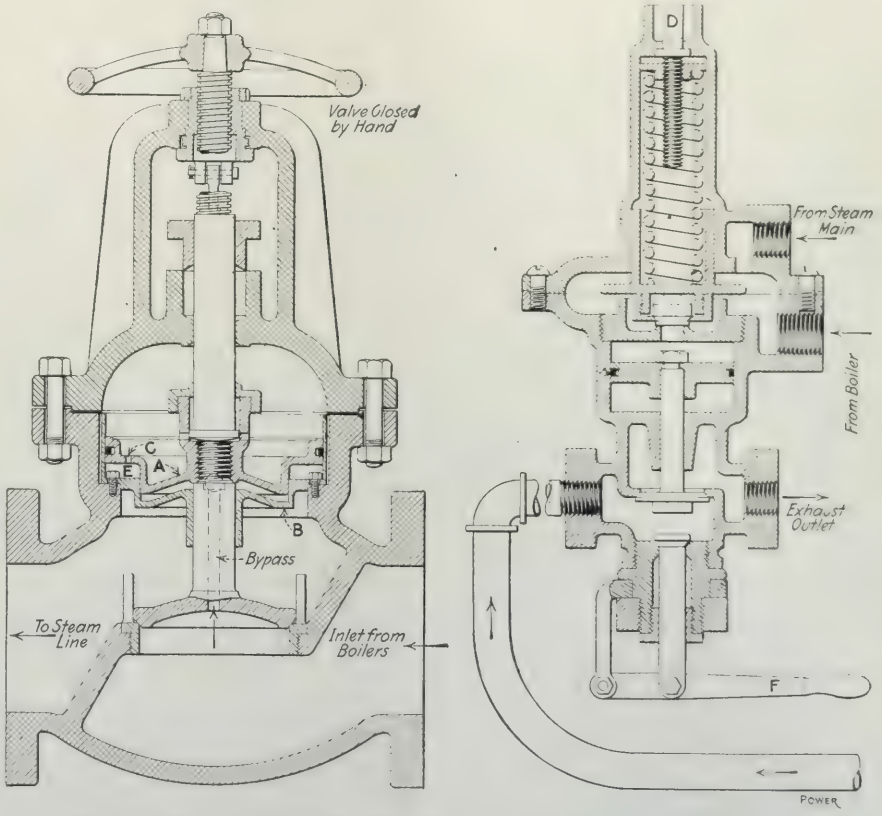


FIG. 1. DOUBLE CUSHIONED NON-RETURN VALVE AND PILOT VALVE

which releases it from the automatic position. The pilot valve, which is piped to the chamber *E*, can be set for any variation of pressure by adjusting the screw *D*.

Fig. 2 shows a double-cushioned, triple-acting, nonreturn globe valve with sealed bonnet. It is designed for the same purpose as the valve shown in Fig. 1. It also operates in connection with a pilot valve, similar to the sketch with an angle valve connected in the steam main.

A partial view of the pilot valve shows the diaphragm, adjustable spring and trigger pin. Full header pressure is carried by the pipe connection to the top of the pilot valve, where it compresses the spring in the central chamber, beneath the diaphragm, and so holds down the trigger pin extending downward outside to the pilot valve.

If a break occurs in the header the pressure above the diaphragm in the pilot is reduced, allowing the spring to lift the trigger pin and release the pilot valve. The boiler pressure from the cushion chamber of each triple-acting valve then has free vent to the exhaust opening in the pilot.

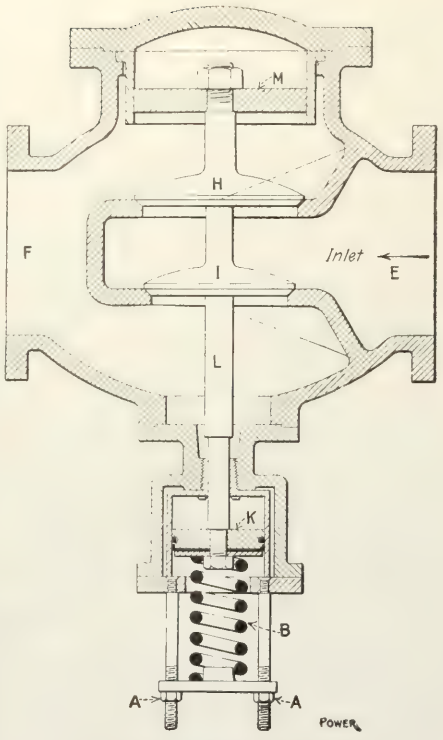


FIG. 3. REDUCING VALVE

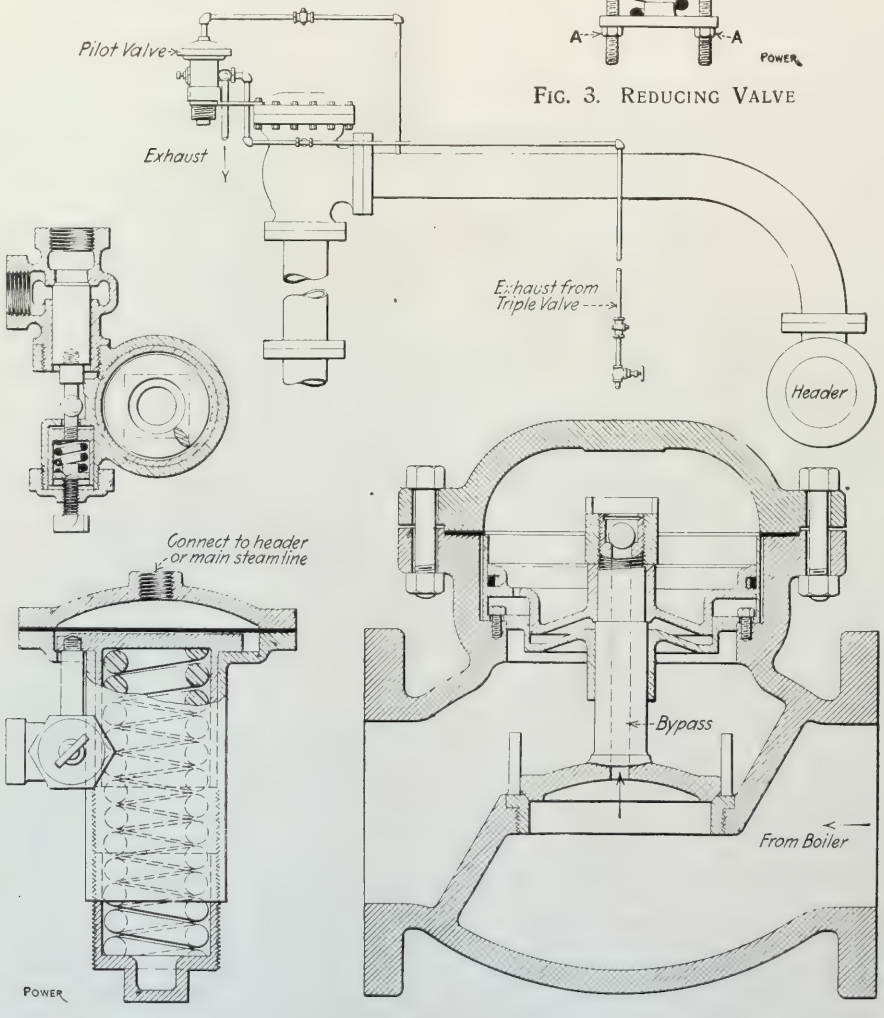


FIG. 2. TRIPLE ACTING NON-RETURN GLOBE VALVE

An automatic, double-seated, reducing valve for high and low pressure is shown in Fig. 3, in a closed position.

When the spring *B* is adjusted by the nuts *A* to the required pressure, the valve disks *H* and *I* are forced open. The high pressure or inlet is at *E*. The low pressure or outlet is at *F*. When the pressure on the low side has reached the pressure at which the spring *B* is set, the pressure still increasing, is exerted on top of the valve *H*, causing the valves *H* and *I* to close. The steam above and below the piston *M* and above the piston *K* prevents any chattering or hammering, and cushions the valve in opening and closing.

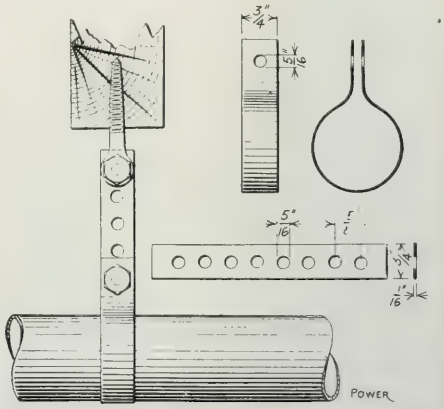
These valves are made by the Golden-Anderson Valve Specialty Co., Fulton Building, Pittsburgh, Penn.

Pipe Hanger for Light Piping

BY H. M. NICHOLS

The accompanying illustration shows the general details of a convenient pipe hanger for supporting light pipe, say from 1 in. up to 3 in. in diameter. It is made up of a V-shaped ¾-in. band-iron $\frac{1}{16}$ in. thick, and bent so that the inside diameter is the same as the outside diameter of the pipe it is to support. The flat ends of this piece are punched with $\frac{5}{16}$-in. holes.

The second part of the hanger is a flat strip of band-iron in which a number of $\frac{5}{16}$-in. holes have been punched about ⅝ in. apart. The strips can be made up in random lengths and when used broken off to the desired length.



PIPE HANGER FOR LIGHT-WEIGHT PIPE LINES

Eye lag-screws are used to fasten the pipe hangers to the ceiling. These lag-screws are most easily put up before the pipe is put in place. The bands can be put on the pipe and clamped loosely to the straight strips with ¼-in. bolts. After the pipe is lifted into place it is only necessary to bolt the straight strips to the lag-screws with ¼-in. bolts.

This type of pipe hanger is particularly convenient when it is necessary to run a line of piping slanting.

Some tubes in one of the boilers of the French battleship "Verite" exploded recently, causing a panic among the sailors. Fortunately, nobody was injured.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Power Required for Pumping

How many horsepower must be developed to pump 200 gal. of water per minute against a pressure of 80 lb. per sq.in.?

D. P.

The weight of one gallon of water being 8⅓ lb., the weight of water pumped per minute would be

$$200 \times 8\tfrac{1}{3} = 1666.66 \text{ lb.}$$

A pressure of 1 lb. per sq.in. is equivalent to the pressure created by a column of water 2.309 ft. high, and pumping against 80 lb. pressure would be equivalent to pumping against a head of 2.309 × 80 = 184.72 ft. or raising the water to that height. The energy developed would be

$$1666.66 \times 184.72 = 307{,}865.43 \text{ ft.-lb. per min.}$$

or

$$\frac{307{,}865.43}{33{,}000} = 9.33 \text{ hp.}$$

Siphoning Feed Water Unreliable

What rate of flow can be relied upon for boiler feeding by piping water over a hill which is about 600 ft. distant and about 8 ft. lower than the source of supply, using 1¼-in. wrought-iron pipe, and is siphoning a reliable means of increasing flow?

E. W. H.

Using clean iron pipe 1¼ in. inside diameter laid without contractions or obstructions, a head of 8 ft. would overcome 600 ft. of pipe friction and deliver a flow of about 5½ gal. per min. at atmospheric pressure. A greater rate of flow could be temporarily secured by siphoning, but partial vacuum would be produced at the highest part of the siphon, causing air to become liberated from the water, and the pipe to become air-bound. Flow increased by siphoning should not be depended upon for direct boiler feeding.

Compression Space of Duplex Pump

What is compression space of the steam end of a duplex pump and how can it be measured?

G. D.

The steam ends of most duplex pumps have separate steam and exhaust passages, the latter being nearer the middle of the cylinder. When the piston passes over the exhaust port no more steam can escape, and whatever space then remains in the cylinder and steam passage will be compression space, in which any steam contained will be compressed and act as a cushion, opposing further motion of the piston until the end of the stroke. The volume of the compression space can be measured by moving the piston until the leading piston ring just covers the exhaust passage, and with the steam-chest cover and valve removed, finding the weight of water required to fill the compression space. The number of ounces of water so required divided by 0.5774 will be the volume of the compression space in cubic inches.

Pressure for Drying Kilns

What is the relative quantity of heat which can be transferred by coils of a drying kiln to be raised to 150 deg. supplied with steam at 100 lb. and when supplied at 130 lb. pressure?

J. J. G.

The temperature of dry saturated steam at 100 lb. boiler pressure (115 lb. abs.) is 338 deg. F. and at 130 lb. boiler pressure (145 lb. abs.) is 355.8, or practically 356 deg. F. The heat transmission of pipe coils with active circulation when supplied with steam at the pressures stated, would be at the rate of about 2 B.t.u. per hr. per sq.ft. of radiating surface for each degree difference between the temperature of the steam and the atmosphere to be heated.

Using steam at 100 lb. boiler pressure the transfer of heat per square foot of surface per hour therefore would be about (338 — 150) × 2 = 376 B.t.u. and on the same basis for steam at 130 lb. boiler pressure the transfer would be

$$(356 - 150) \times 2 = 412 \text{ B.t.u.}$$

or about 9½ per cent. more heat.

Weight and Thicknesses of Flywheel Rim

What should be the thickness of a rim for a cast-iron bandwheel 14 ft. in diameter with 24-in. face for a 20x42-in. Corliss engine running 90 r.p.m.?

E. R.

The formula for weight of flywheel is

$$W = 700{,}000\, d^2 s \div D^2 R^2$$

in which

W = Weight in pounds;

d = Diameter of cylinder = 20 in.;

s = Stroke = 42 in.;

D = Diameter of flywheel = 14 ft.;

R = Revolutions per minute = 90;

and substituting the values given, the weight would be

$$W = \frac{700{,}000 \times 20 \times 20 \times 42}{14 \times 14 \times 90 \times 90} = 7407.4 \text{ lb.}$$

As cast iron weighs 0.26 lb. per cu.in., the rim would contain 7407.4 ÷ 0.26 = 28,490 cu.in. As the face of the wheel is to be 24 in., if the section of the rim is rectangular and of uniform thickness, the area of one side of the rim would be 28,490 ÷ 24 = 1187 sq.in., and this latter quantity will be the difference in areas of two concentric circles, viz., one having its radius equal to the radius of the wheel, and the other having its radius equal to the inner radius of the rim. The radius of the wheel being 84 in., the area of the outer circle would be 22,167 sq.in.; the area of the inner circle would therefore be 22,167 — 1187 = 20,980 sq.in. and the inner radius of the rim would be

$$\sqrt{\frac{20{,}980}{3.1416}} = 81.72$$

or practically 81¾ in., and the thickness of rim would be

$$84 - 81\tfrac{3}{4} = 2\tfrac{1}{4} \text{ in.}$$

Combined Steam and Water Power

When the main jackshaft of a mill is driven by power developed by a waterwheel and an engine at the same time, what is the best way of throwing as much of the load on the water power as it will carry?

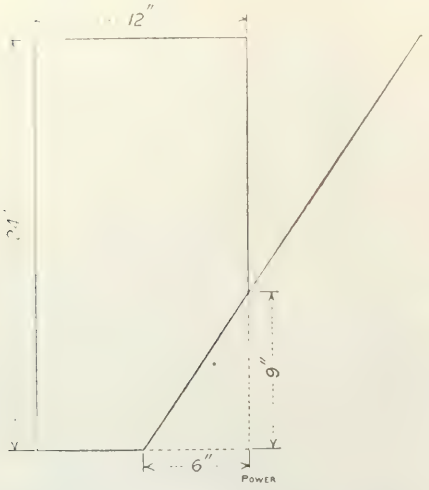
J. S.

To make the fullest use of the water power available, the engine governor should regulate the engine to a slower speed of jackshaft than the governor of the waterwheel, thereby calling on the engine only for power required in excess of the power developed by the waterwheel. In starting up, the load should be moved off by the power of the engine alone to better insure that the engine cylinder is warmed up and all water of condensation is removed before throwing on the power of the waterwheel. Neglect of the latter precaution has resulted in many serious engine smashes, especially during cold weather, when cylinder condensation is greatest.

Study Questions

This Week's Questions
Last Week's Answers

(116) What would be the volume of a cylindrical tank 12 in. in diameter and



24 in. high, made to fit against a sloping wall, as shown in the sketch?

(117) A steam-fitter went to purchase 1½ lb. of sheet packing and offered the correct change. The salesman cut a piece and weighed it; then remarked, "This runs 10c. over." The steam-fitter then told him to cut it in two, that he would take one-half of it and with the extra 5c. he would buy some putty. How much was the packing per pound?

(118) If 5½ lb. of water is heated from 50 to 75 deg. F., how many units of heat is absorbed and what is the equivalent in units of work?

(119) A pound of heated cast iron is plunged into 2 gal. of water and raises the temperature of the latter from 60 to 69 deg. F. What was the original temperature of the iron if its specific heat is 0.13?

(120) The thrust of a screw propeller is 10 tons, and acts against a ring having a mean radius of 6 in. The coefficient of friction being given as 0.06 and the shaft making three revolutions per second, how much power is consumed in friction?

Answers to the above will appear in the next issue. Answers to last week's questiions follow:

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(111) The increase in resistance due to an increase of temperature is 0.42 per cent. for each degree Centigrade above zero, the resistance at 0 deg. C. being taken as the base. Let

R_0 = Resistance at 0 deg. C.;
R_1 = Resistance at temperature t_1;
R_2 = Resistance at temperature t_2.

Then

$$R = R_0 (1 + 0.0042 t_1)$$

In the present case $R_1 = 100$ ohms; $t_1 = 90$ deg. F. $= 32.22$ deg. C.; and $t_2 = 75$ deg. F. $= 23.89$ deg. C. Substituting

$$100 = R_0 (1 + 0.0042 \times 32.22) = 1.14\ R_0$$

whence

$$R_0 = \frac{100}{1.14} = 87.72$$

again

$$R_2 = R_0 (1 + 0.042\ t_2)$$
$$R_2 = 87.72\ (1 + 0.0042 \times 23.89) = 96.52$$

or the resistance per foot at 75 deg. F. equals

$$\frac{96.52}{1000} = 0.09652\ ohms$$

(112) Let

x = Engineer's age when he took charge of the plant.

Then

$\frac{4x}{3}$ = Engineer's age now;

and

$\frac{x}{3}$ = Number of years since engineer took charge;

so that

$$\frac{x}{3} + 5 = \text{Oiler's age now} \qquad (1)$$

Let

y = Fireman's age when engineer took charge;

then

$$\frac{y}{2} + 2 = \text{Oiler's age now} \qquad (2)$$

and

$$y + \frac{x}{3} = \text{Fireman's age now} \qquad (3)$$

From (1) and (2)

$$\frac{x}{3} + 5 = \frac{y}{2} + 2$$
$$2x + 30 = 3y + 12$$
$$3y = 2x + 18$$
$$y = \frac{2x}{3} + 6$$

From this value of y in (3)

$$\frac{2x}{3} + 6 + \frac{x}{3} = x + 6 = \textit{Fireman's age now}$$

Let

z = Number of years before the oiler will be as old as the engineer was when he took charge.

Adding this to the oiler's age will equal the engineer's age or

$$\frac{x}{3} + 5 + z = x$$
$$x + 15 + 3z = 3x$$
$$3z = 2x - 15$$
$$z = \frac{2x}{3} - 5 \qquad (4)$$

In z years the sum of their ages is to be 150. Then

$$\left(\frac{4x}{3} + z\right) + \left(\frac{x}{3} + 5 + z\right) + (x + 6 + z) = 150$$

Substituting the value of z from (4)

$$\left(\frac{4x}{3} + \frac{2x}{3} - 5\right) + \left(\frac{x}{3} + 5 + \frac{2x}{3} - 5\right) + \left(x + 6 + \frac{2x}{3} - 5\right) = 150$$

Simplifying

$$\frac{14x}{3} = 154$$
$$14x = 462$$

$x = 33$ = Engineer's age when he took charge.

Then the oiler's present age is

$$\frac{x}{3} + 5 = \frac{33}{3} + 5 = 16\ yr.$$

and the fireman's age is

$$x + 6 = 33 + 6 = 39\ yr.$$

and the engineer's age now is

$$\frac{4x}{3} = \frac{132}{3} = 44\ yr.$$

(113) The total heat (above 32 deg. F.) in a pound of steam at 16 lb. abs. (from steam tables) is 1152 B.t.u. A pound of water at 115 deg. F. contains

$$115 - 32 = 83\ B.t.u.$$

A pound of steam condensed has therefore lost

$$1152 - 83 = 1069\ B.t.u.$$

Since

$$115 - 48 = 67\ B.t.u.$$

will raise 1 lb. of injection water from intake to discharge temperature it will require

$$\frac{1069}{67} = 15.955\ lb.$$

or nearly 16 lb. of injection water for each pound of steam condensed.

(114) The maximum unit stress developed upon the outer fiber of the shaft will be

$$s = \frac{16\ C\ d}{\pi\ (d^4 - d_1^4)}$$

where

C = Twisting moment;
d = Outside diameter;
d_1 = Inside diameter;
$\pi = 3.1416$.

The twisting moment is the force P applied at a distance p from the center of the shaft or $C = Pp$. The problem gives $P = 3000$ lb., $p = 12$ in., hence

$$C = 3000 \times 12 = 36{,}000\ in.\text{-}lb.$$

and

$$d = 3\ in. \text{ and } d_1 = 2\ in.$$

Substituting in the formula

$$s = \frac{16 \times 36{,}000 \times 3}{3.1416\ (3^4 - 2^4)} = \frac{1{,}728{,}000}{204.204}$$
$$= 8462.126\ lb.\ per\ sq.in.$$

(115) By the Law of Charles the pressure of a gas at constant volume varies as the absolute temperature

350 *deg. F.* + 461 = 811 *deg. abs.*
32 *deg. F.* + 461 = 493 *deg. abs.*

Then

$$811 : 60 :: 493 : x$$
$$x = \frac{60 \times 493}{811} = \frac{29{,}580}{811} = 36.47\ lb.$$

is the pressure of the air when the temperature is reduced to 32 deg. F.

Electric Railway Convention

The American Electric Railway Association and its affiliated associations held their thirty-first annual conventions in Chicago, Oct. 7, 8, 9, 10 and 11.

The report of the American Electric Railway Engineering Association committee on power generation possesses greatest interest for the majority of our readers. The first of the three subjects treated was "Method of Selecting Faulty High-tension Cables at the Power House." The replies to a letter sent to a number of companies operating large power plants, requesting information on their methods, indicated that some companies have not taken advantage of the development of the art and have no satisfactory method of selection. Others have methods fairly satisfactory to the local conditions. The replies in general showed a real need of further development of new ideas and apparatus to meet the requirements of satisfactory selection suited to their operating needs.

The second subject, "Chemical Laboratory Practice in Connection with Power Work," will be abstracted in a later issue.

The third, a paper by L. P. Crecelius, entitled "Boiler Settings," brought out that furnace arrangement has undergone considerable change recently and that it is now not uncommon to see a furnace projected out to a distance greater than the length of the stoker, to increase as much as possible the distance to the tubes from the point where the fuel is introduced.

In very recent installations the furnaces are installed entirely under the boiler instead of being extended, but the distance between the grate and the tubes is made very large to provide ample room for combustion.

In one case where horizontal water-tube boilers are employed, this distance is fully 12 ft.; in another, Stirling boilers provide an enormous combustion chamber. Therefore, in both cases the necessity of maintaining large and expensive arches is dispensed with. Almost every smoke-inspection bureau proposes, and in some cases insists upon, designs, including a horizontal baffle or arch, directly over the furnace, to correct improper existing conditions. It is believed, however, that if the boilers are set high enough in the first instance more freedom in the choice of arrangements for gas passes and baffling will be permitted.

In some recent tests upon archless furnaces operated continuously to produce 200 per cent. of the boiler rating, the tests being carried out for the purpose of determining limitations to successful operation at heavy overloads, Mr. Crecelius found that the ordinary firebrick linings of the furnace were subjected to enormous deterioration. Another difficulty developed in the fusing of ashes to the side and bridge-walls, just over the grate surface. A mass of fused ashes would accumulate until it extended all along the faces of the walls parallel with the grate, sticking out in some cases over 24 in., interfering with the forward movement of the fuel and proving extremely difficult to remove.

The rapidity with which such accumulations form depends upon the furnace temperature and the character of the fuel. A fuel having ash comparatively free of iron gives less trouble than one containing more iron; again, when boilers are operated at low rates and consequent low-furnace temperatures, the poorer fuel may be burned without serious trouble. However, because extremely high furnace temperatures are required, and in fact desired for smokeless combustion when boilers must respond to heavy overloads, all furnace linings are more or less subject to heavy deterioration.

The paper states the following conclusions: It is highly essential to provide large combustion chambers over furnaces intended for overloading, not alone for supplying space for flame, because with forced-draft furnaces this is unnecessary, but for the more important purpose of securing high initial temperature. A high temperature makes possible a minimum air supply to the furnace. Therefore, the cooling influence of the boiler surface or tubes must be sufficiently removed to prevent the rapid absorption of radiant heat, as only by this means can high furnace temperatures be secured.

Since the critical temperatures for ordinary furnace linings appear to have been reached already, it seems that future developments in the successful forcing of boilers must be accomplished by means of the archless type of furnace. This becomes more apparent when it is considered that vertical furnace linings are more easily and cheaply maintained than the large horizontal, overhanging surfaces or arches.

The following officers of the American Electric Railway Association were chosen on Oct. 10: President, George H. Harries; first vice-president, Charles N. Black; second vice-president, C. Loomis Allen; third vice-president, C. L. Henry; fourth vice-president, John A. Beeler.

On Oct. 11 the American Electric Railway Engineering Association elected as president, Martin Schreiber; first vice-president, L. B. Crecelius; second vice-president, John Lindall; third vice-president, J. H. Hanna; secretary-treasurer, H. C. Donecker.

The large German cruiser "Goeben," which was launched last year, is now undergoing her trials. It is reported that she has attained a speed of 32 knots.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Already, we are ready to place R. Reddie, of Newmarket, Tenn., in a choice niche of our Hall of Fame. Who is next?

Extensive improvements are shortly to be made at the paper mill, prominent among which will be the installation of a 700,000-hp. boiler—Steubenville (Ohio) "Herald-Star."

Our friend Frazier sends us this clipping, and adds: "Where would Steubenville be if the thing exploded?" We know no one able to answer this question, unless it be Danny Hogan or possibly the angel Gabriel.

Whatever may be said of our water-power possibilities, our power on water is a grim reality. The fleet recently in New York waters is capable of developing 427,000 hp. Uncle Sam's only near rival seems to be that 700,000-hp. boiler over at Steubenville.

Don't be misled by that wily refrigeration engineer who claims that half an hour spent in a brewery refrigerating room will cure a stubborn case of hay fever. As your Uncle Spill diagnoses the case, there's small choice between a sneeze and a hiccough—and he's tried 'em both.

A Hamburger has staked his reputation on a tide-power scheme to provide the German consumer with two-cent current by utilizing the North Sea as a power plant. This sprightly little H_2O body ought to make a fair ice-plated plant in winter time.

The House of Hill is to have a home of its own. The family has so greatly outgrown the old homestead on the Pearl St. farm that our Old Man, seeing his chance on Tenth Ave., just dug down into his old woollen stocking and closed the dicker. Soon we will start clearing away the brush for a home that will make our neighbors stand bolt upright and throw down their hoes. We will have all the fixings that this village's skill can provide. Along about February, 1914, when the sledding is good and the last shingle is nailed on the roof, fish out your woollen mittens, wrap yourself and the missis in the buffalo robe, and start old Dobbin jingling the bells right up to our dooryard. This is you-all's "invite" to the housewarming. If interested, read about it on page 626.

Minneapolis N. A. S. E. "Booster" Meeting

Minnesota Association No. 2 of the National Association of Stationary Engineers held a "booster" meeting and banquet at the Hotel Radison, Minneapolis, on Saturday evening, Sept. 28. It was the largest and most enthusiastic gathering in the association's history.

Minnesota No. 7, of St. Paul, came over in a body, and employers and many leading citizens helped to make the attendance nearly 300. The banquet was in the hotel ball room.

Prof. J. J. Flather, of the University of Minnesota, delivered an instructive and witty discourse on engineering matters in general.

National President J. F. McGrath, of Chicago, told of the aims and objects of the association and what it has accomplished in the past and hopes to do in the future. He assured No. 2 of his hearty support at all times during his administration.

J. A. A. Burnquist, a member of Minnesota No. 7, a candidate for lieutenant-governor of the state, and one of the association's best friends, discussed license laws, convincing all present that he knew his subject and is working along the proper lines.

John W. Lane, Chicago, editor of the *National Engineer*, satisfied his hearers that he is the right man to manage the national organ, and his advice and counsel made a deep impression.

W. L. Harris, president of the New England Furniture & Carpet Co., a director of the Civil and Commerce Association, and one of Minneapolis' most successful citizens, gave a characteristically vigorous and spicy talk. The toastmaster then called for applicants for membership and 40 could not resist the temptation.

Alderman C. D. Gould, a candidate for mayor, spoke of the great good accomplished by the association and offered his services whenever he could be of assistance.

Other speakers were J. B. Crane, Duluth, state president; Herman Mueller, Minneapolis, state vice-president; James McCreary, St. Paul, state secretary; T. S. F. Hayes, St. Paul, state deputy and president of No. 2; Fred Camitsch, St. Paul, and J. M. Williams, Minneapolis.

The meeting and banquet, conducted by the entertainment committee, J. F. Gould and Frank Rosenthal, was such a complete success that it is planned to make it an annual event. This is the beginning of an active campaign for new members and No. 2 expects to have every engineer in the city enrolled in the near future. The educational and entertainment committees are already planning some very interesting things for this winter.

In the afternoon the national officers were driven about the Twin Cities in automobiles accompanied by members of No. 2 and shown some of the large power plants and the many points of interest.

Boosters and Banqueters at the Minneapolis N. A. S. E. Meeting

New York Electrical Show

The usual annual electrical exposition and automobile show was held at the Grand Central Palace, New York City, from Oct. 9 to 19. There were about 100 exhibits on the three floors of the building and the large daily attendance showed that interest in this kind of exposition has not lagged.

A luncheon to Thomas A. Edison, commemorating the completion of 30 years of central-station service, opened the exposition. J. W. Lieb, Jr., acted as toastmaster and Samuel Insull, of the Chicago Commonwealth Edison Co., was among the speakers.

The central stations in and around New York City were well represented, and the U. S. Government contributed several instructive exhibits, that of the electric class at the Brooklyn Navy Yard being especially interesting.

Institute of Operating Engineers

At the first meeting of the new council of the Institute of Operating Engineers, L. Houmiller was appointed executive secretary in place of H. E. Collins, whose term had expired and who is too busy to give the necessary time this year. Mr. Houmiller has been with the institute since its inception and is ready to do all he can to advance the work and help accomplish the aim of the institute.

The constitution, as revised at the annual meeting in September, will be printed and circulated the latter part of this month. The *Bulletin* will be out the early part of November. A copy will be sent to those interested by writing to the secretary, 29 West Thirty-ninth St., New York City.

Our New Home

The Hill Publishing Co., John A. Hill, president, will erect a 14- to 16-story building at the northeast corner of Tenth Ave. and Thirty-sixth St., New York City. The property comprises half the block on Tenth Ave., and extends 175 ft. in Thirty-sixth St., and is covered with tenements.

It is Mr. Hill's object to have the most complete publishing and commercial building in the city. Every known convenience for cleanliness, comfort, safety, low insurance and economical conduct of business will be installed. The publishing company will occupy several floors. A special feature of the building will be a roof garden for the recreation of the occupants of the building, both tenants and employees. It is expected the building will be ready to occupy February, 1914.

Boiler Accident in Poughkeepsie Plant

At 10 p.m., Sunday, Nov. 13, a tube pulled out of the mud drum of a Stirling boiler operating under 160 lb. pressure at the plant of the Central Hudson Gas & Electric Co., Poughkeepsie, N. Y. Four men were badly injured, but Fireman Doyle, who was reported dying, is rapidly recovering. Due to a loss of steam pressure caused by the ruptured tube, a part of the service was interrupted until another boiler was cut in on the line. The interruption was trivial, however, as the load carried at the time of the accident was rather light.

The boilers are equipped with Murphy furnaces, and Fireman Doyle had the firedoor open breaking a clinker when the tube let go. Another man was in the front ashpit of the boiler directly opposite and both were deluged with incandescent fuel and steam and water. An attendant who rushed to the top of the boiler to shut the stop-valve was severely burned about the hands, and the engineer, in an effort to rescue the injured, lost his way in the steam-filled room and fell into an open outside ashpit, receiving bruises about the limbs and back.

The tube that let go was almost directly in front of the fire-door, and in pulling out of the mud drum it assumed an angle directly opposite to that at which it entered the drum, or, instead of the open end facing the drum it pointed away from it, straight toward the fire-door. The doors of a Murphy furnace are very heavy, and if they had been closed when the tube pulled out, it is possible no one would have been injured as the steam pressure might have dissipated itself in the furnace, the volume of which is very great, without blowing the doors open immediately, thus giving the attendants time to get out of the path of the hot fuel and steam and water.

It is not definitely known whether the cause of the accident was corrosion of the tube or faulty workmanship in securing it to the drum.

Hot Water Tank Explosion

Shortly after 3 a.m., Oct. 2, a hot-water tank exploded in a café in New Bedford, Mass. The interior of the building was badly wrecked and the plate-glass front blown across the street. The damage amounted to about $5000.

Ten-inch floor timbers were split from end to end, a mezzanine floor laid on 3 in. of reinforced concrete was torn up, marble-topped tables were thrown to the ceiling and smashed. While all this occurred over the tank and heater, bottles and glasses remained undisturbed on the bar 15 ft. away.

The tank which exploded was of 75 gal. capacity, of heavy copper with soldered and riveted seams and marked with a 500-lb. test mark. It furnished hot water for the kitchen and was suspended from the basement ceiling. Below the tank was a cast-iron heater which was uninjured. The tank was connected to the city mains, the pressure in which automatically kept the tank full.

The tank had a lever-safety valve which an employee stated had never been in working order. After the explosion, the cold-water supply pipe was found to contain a check valve which prevented relief of the excess pressure from expansion into the street mains when the safety valve failed to operate.

NEW PUBLICATION

PRACTICAL AERONAUTICS. By Charles B. Hayward; with introduction by Orville Wright. American School of Correspondence, Chicago, Ill. Cloth; 6½x9¾ in.; 769 pages; 310 text figures. Price, $3.50, net.

While this work is necessarily somewhat bulky, it is a well digested and readable compilation on aëronautics. Sufficient theory is simply treated to aid the reader in readily understanding the various types of construction. Very little space has been given to dirigible balloons, which are for the most part confined to the Wellman, Vaniman and Zeppelin ships.

Under "Types of Aëroplanes" are described the "standard" biplanes of Wright, Curtiss, Voisin, Farman and Cody; the "standard-type" monoplanes include the Antoinette, Santos Dumont, Bleriot, Grade and Pfitzner types.

The section devoted to building and flying an aëroplane will be of undoubted interest to many readers. It includes the construction of models, a glider, Curtiss biplane, Bleriot monoplane, the art of flying, accidents and their lessons, and amateur aviators.

While it will be some time before aëronautics becomes a popular pastime of the operating engineer, to those who are interested, and can afford to fly, this book contains much valuable information.

(Locomotive)—In the notice of the bound volumes of *Locomotive* for 1910-1911, which was printed in the Oct. 8 issue, the mistake was made of saying that these volumes are complimentary. The price of the bound volumes is $1 each.

OBITUARY

On Sept. 18, Frederic Austin Warren, for 11 years chief electrician of the Colorado Fuel & Iron Co., died at Canon City, Colo. Prior to his connection with this company, he was employed by the General Electric Co., and later served as electrician at the Fremont and Coal Creek mines.

PERSONAL

J. O. Westberg has assumed charge of the mechanical department of the Felber Engineering Works, New York City. Mr. Westberg was formerly with the Sibley Construction & Supply Co., New York City.

Arthur Hadley, managing director, and Bernard Price, chief engineer, of the Victoria Falls & Transvaal Power Co., South Africa, recently arrived in America, and will visit some of the large electric-power plants.

Truman D. Hayes has resigned as New England sales engineer of the Triumph Electric Co., Cincinnati, Ohio, to accept a similar position with the International Steam Pump Co., New York City., which he will represent in eastern New York.

Kern Dodge, a well and favorably known Philadelphia engineer, has returned from a year's travel abroad and will open an office in the Morris Building, Philadelphia. He will devote himself henceforth to the engineering and financing of public-service properties. Mr. Dodge was a founder and for many years a partner of Dodge & Day, engineers.

Frank Koester, consulting engineer, has removed his office from 115 Broadway to larger quarters at 50 Church St. (Hudson Terminal Bldg.), New York City. Mr. Koester is principally engaged in steam and hydro-electric power-plant work, electric transmission and traction. He was recently retained as an expert by the Borough of Manhattan, New York City.

Wentworth P. Johnson has been appointed New York manager for H. M. Byllesby & Co. For the last nine years he was the European financial agent of the International Harvester Co., residing in London. Mr. Johnson is an old friend and business associate of Mr. Byllesby, having been connected with him in the Northwest General Electric Co. at St. Paul.

Carlisle Mason, vice-president and general manager of the Nelson Valve Co., Philadelphia, Penn., has tendered his resignation, to take effect Nov. 1, 1912. He assumed charge of this company five years ago and built up the business to its present standing, establishing the model plant at Chestnut Hill, Penn. Mr. Mason is well known in the trade. On his return from a trip of several months abroad, he will probably reënter the valve and fitting business.

The Ridgway Dynamo & Engine Co. is building the Rateau turbine, under arrangement with the American representatives of the inventor. A 1000-kv.-a. unit built at the shop has been in successful operation for several months and a number of other units are now in hand.

Moments with the Ad. Editor

You've been hearing a lot lately about "conservation of energy."

The difference between a great and an insignificant man lies in the amount of *reserve power* he has back of him.

* * *

Napoleon, probably the world's greatest modern soldier, never let his "old guard" go into a fight unless the exigency was most urgent—he held them in *reserve* and then, at the critical moment, they swept all before them. His plan succeeded admirably until he met a man with *more reserve force* than he had.

The lungs contain a *reserve* of air that is never used—the lymphatic system is nature's reserve of *all* the forces and elements used in the human body. Men have gone without food for long periods of time simply existing on the *reserve* stored up in that lymphatic system.

The government *reserves* great forests and waterways for use by a future generation—wise men put money in *reserve* against a "rainy day." It is the *energy* we have *conserved*, lying dormant, that allows us to meet a crisis and *overcome* it.

Now consider conservation of thought and *you*. Sometime in your life an opportunity has come to you. If you've grasped it, it was only because you had the "know how"—the *reserve knowledge* to bring to bear. Another opportunity will arise probably at no far distant date, have you a reserve to meet *that*?

What is the answer? Study. And what you learn, *conserve*. Books, lectures, periodicals, all are excellent, but in *your* case, you men whose future positions depend on the knowledge you have of your work and its equipment—you should study *advertisements*.

In them you receive first news of the newer, the greater improvements in cost-cutting, result-pulling appliances. First hand information of the fight that's continually going on to improve *conditions*. And you can *depend* on the news you get in an advertisement absolutely for, through experience, manufacturers have learned that it is ruinous to them to advertise dishonestly.

* * *

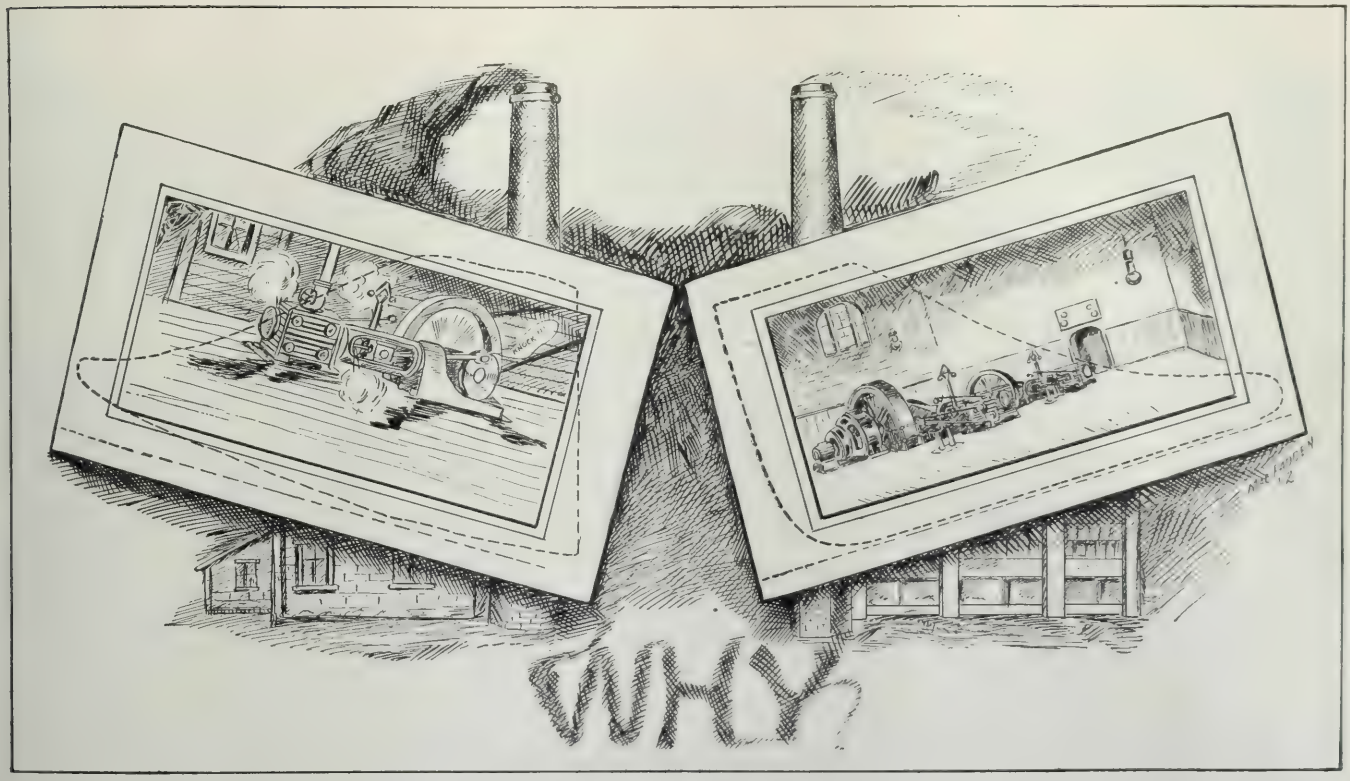
So study advertising, think how it applies to *your* case—consider the products—find out if they'll *help* you—and *remember* what you read.

Some day, when the *need* arises, you'll have *conserved* knowledge—enough to *meet* the need.

POWER

Vol. 36 NEW YORK, OCTOBER 29, 1912 No. 18

A Study in Contrasts



[The above is more in the nature of a true cartoon than any we have yet printed, because the picture is complete in itself, needing no text to explain it. We wish we might have more as good.

Any who feel they cannot make drawings good enough for direct reproduction, may send rough sketches or even only descriptions of their "ideas" and accepted ones will be paid for in proportion.—EDITOR.]

Plant of Landers, Frary & Clark Co.

By Warren O. Rogers

A power plant in which many economical features have been incorporated. Upright hand-fired boilers and compound reciprocating engines comprise the main generating units.

One of the interesting power plants in New Britain, Conn., is the new one recently completed by the Landers, Frary & Clark Co., manufacturers of silverware, etc.

Boiler Room

Obtaining permission to visit the power plant, the writer found his way to the boiler room, where Chief Engineer Arthur H. Hill was superintending the washing out of several upright boilers, of which there are 11, each of 235 hp. capacity. As he had been caught working, something for which chief engineers are often not given credit, Mr. Hill explained that when his boilers are cleaned, he wants to personally know their condition, first, not second hand. "We hand-fire run-of-mine coal and handle the fires according to the quality of the fuel," said he.

"Usually the fires are run at a thickness of 10 in. They are not sliced during a run, but are shaken down occasionally. The fires, however, are leveled by evenly spreading the coal in light quantities, very rarely having to use the hoe, when the boilers are operating at their normal rating.

"The flue gases pass to the economizers at a temperature of 420 deg. F., leaving at 240 deg. F." Just then Mr. Hill was called to another part of the plant and the writer took the opportunity to investigate the boiler room.

Coal is delivered to the firemen from the yard supply in a 1½-ton car running on a commercial track, the rails of which are placed level with the brick floor. The track is made with a cross-over at the far end of the room which permits loaded cars being delivered to that end and empty cars run out at the near end. By this arrangement a supply of coal is always awaiting the firemen. The gases of combustion after passing through the two economizers go to a 225-ft. brick stack which creates a 1¼ in. draft.

A view of the piping in the boiler room is shown in Fig. 1. The main feed pipe runs along the front of each battery with globe valves in the vertical pipe leading to each individual boiler. There are two steam mains, one leading to the main generating units, the other supplying steam to the auxiliaries and also for factory use. The steam pressure is carried at 175 lb. The feed water passes through an open heater to this water weigher, it is then pumped through a closed heater and thence through the two economizers. All clean drips and the auxiliary turbine exhausts are returned to the open heater. Other exhaust steam passes through only the closed heater, and all oily drips are piped to the sewer.

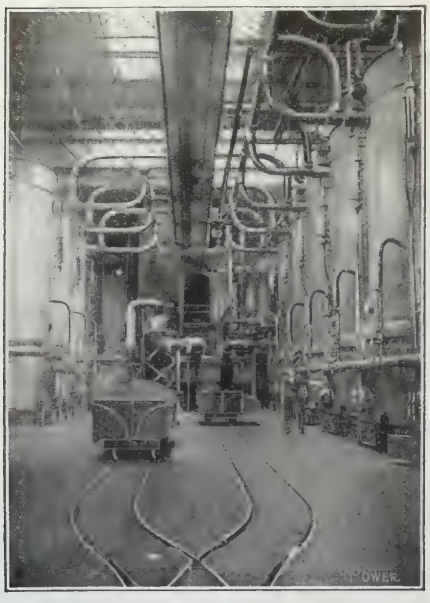


Fig. 1. Showing the Vertical Boilers and Car Tracks

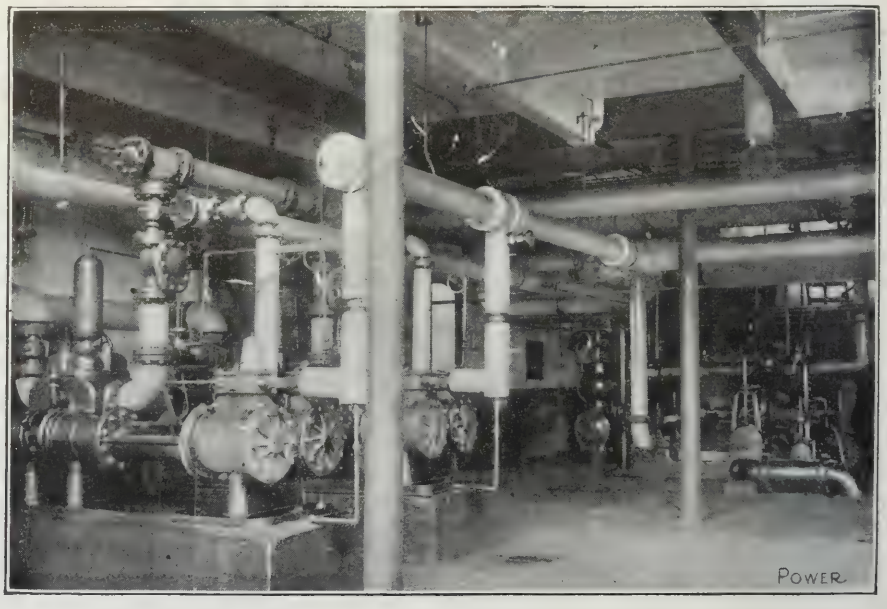


Fig. 2. One End of the Auxiliary Room

Auxiliaries

In passing from the boiler room to the engine room, which is one story above the former, one passes through the pump room. There is sufficient room here because the entire basement outside of the engine foundation is given up to the auxiliary units, hence there has been no crowding. Fig. 2 shows a view of one side of the auxiliary room. There are two 12 and 7 by 12-in. boiler feed pumps, each mounted on a concrete base, bringing them to a convenient height for making repairs, etc.

The generating units are served by a twin vertical air pump and jet condenser, which maintains a vacuum of 27½ in. The condenser is set next to the head end of the engine foundation in the basement so that the connection between the engine and condenser is as short as it can be conveniently made. The condensing water is pumped from the hotwell through a cooling tower, having a capacity of 2170 gal. of water per minute and cooling from 120 to 90 deg., the atmospheric temperature being 70 deg. F. This water is handled by a turbine-driven 10-in. volute pump. The discharge is controlled by a poppet valve in the steam line, operated by a float in the hotwell.

Engine Room

There are two main generating units, one with 24 and 48 by 48-in. cylinders, and one with 15¼ and 32½ by 22-in. cylinders. The larger cross-compound engine has a normal rating of 1550 hp. The smaller unit of 550 hp. is used as an auxiliary.

A motor-driven exciter-generator set of 60 kw. capacity, and a 75-hp. turbine-driven unit, used for the same purpose, are in an open space beside the smaller

main generating unit. The turbine runs at 6000 r.p.m., and is directly coupled to two 25-kw. direct-current generators running at 1500 r.p.m.

The factory lighting circuits are of 110 volts. The switchboard, located at one side of the engine room, is elevated as shown in Fig. 3. A gravity oil feed system is connected to the two engines, the supply tank of which is elevated. Returns from the engine bearings are filtered and pumped into the elevated supply tank.

The factory buildings are heated by a hot-water system, the water of which is heated by exhaust steam and in extreme cold weather by additional live steam. Circulation is obtained through a 60-hp. turbine-driven 10-in. centrifugal pump. There is also a motor-driven unit of the same size.

The engine-room floor is of concrete and the boiler-room and basement floors are brick; the entire plant is roomy and well lighted.

By the time the writer's inspection was well under way, Mr. Hill returned. The method of keeping power-plant data was discussed, which resulted in the writer obtaining a copy of the report sheet, shown in Fig. 4.

One feature of this report form is that all necessary information regarding

EQUIPMENT OF THE LANDERS, FRARY & CLARK POWER PLANT

No.	Equipment	Make	Use	Kw.	Volts	Amp.	Phase	r.p.m.	Size	Pres. lb.	Hp.	Manufacturers
11	Boilers	Bigelow-Manning	Steam generator							175	235	Bigelow Co.
4	Economizers	Sturtevant	Heating feed water									B. F. Sturtevant Co.
1	Heater	Open	Heating feed water									Hoppes Mfg. Co.,
1	Heater	Closed	Heating feed water									Whitlock Coil Pipe Co.
1	Water weigher	Kennicott	Water meter									Kennicott Water Softener Co.
2	Duplex pumps	Outside packet	Boiler feed						12x7x12″	175		Blake Pump & Condenser Co.
1	Condenser	Jet	Main unit						20x36x21″	175		Blake Pump & Condenser Co.
1	Cooling tower	Wheeler	Cooling water									Wheeler Condenser & EngineeringCo.
1	Turbine	Terry	Hot well pump					4200		175	40	Terry Steam Turbine Co.
1	Pump	Volute	Hot well					4200	10″	175		Worthington, Henry R.
1	Engine	McIntosh, Seymour	Main unit					100	24x48x48″	175	1550	McIntosh, Seymour & Co.
1	Engine	Phoenix	Main unit					171	15¼x32½x22″	175	550	Phoenix Iron Works Co.
1	Generator	Alternating current	Main unit	800	550 600	1440	3	100				Crocker-Wheeler Co.
1	Generator	Alternating current	Main unit	350	600	575	3	171				Crocker-Wheeler Co.
1	Motor	Induction	Exciter set		550	131	3	900			100	Crocker-Wheeler Co.
1	Generator	Direct current	Exciter set	60	125	480		900				Crocker-Wheeler Co.
1	Turbine	De Laval	Exciter set					6000		175	75	Le Laval Steam Turbine Co.
2	Generators	Direct current	Exciter set	25	125	200		1500				Crocker-Wheeler Co.
2	Filters	White star	Engine oil									Pittsburgh Gage & Supply Co.
1	Turbine	DeLaval	Heating system							175	60	Le Laval Steam Turbine Co.
1	Pump	Centrifugal	Heating system						10″			Westinghouse Machine Co.
1	Motor	Induction	Heating system		550		3	1140			60	Westinghouse Electric & Mfg. Co.
1	Pump	Centrifugal	Heating system						10″			Westinghouse Machine Co.
1	Stack	Brick							225′			
1	Heating system	Hot water	Factory									Evans-Almiral & Co.
1	Switchboard	Marble	Electrical control									Electrical & Mfg. Co.

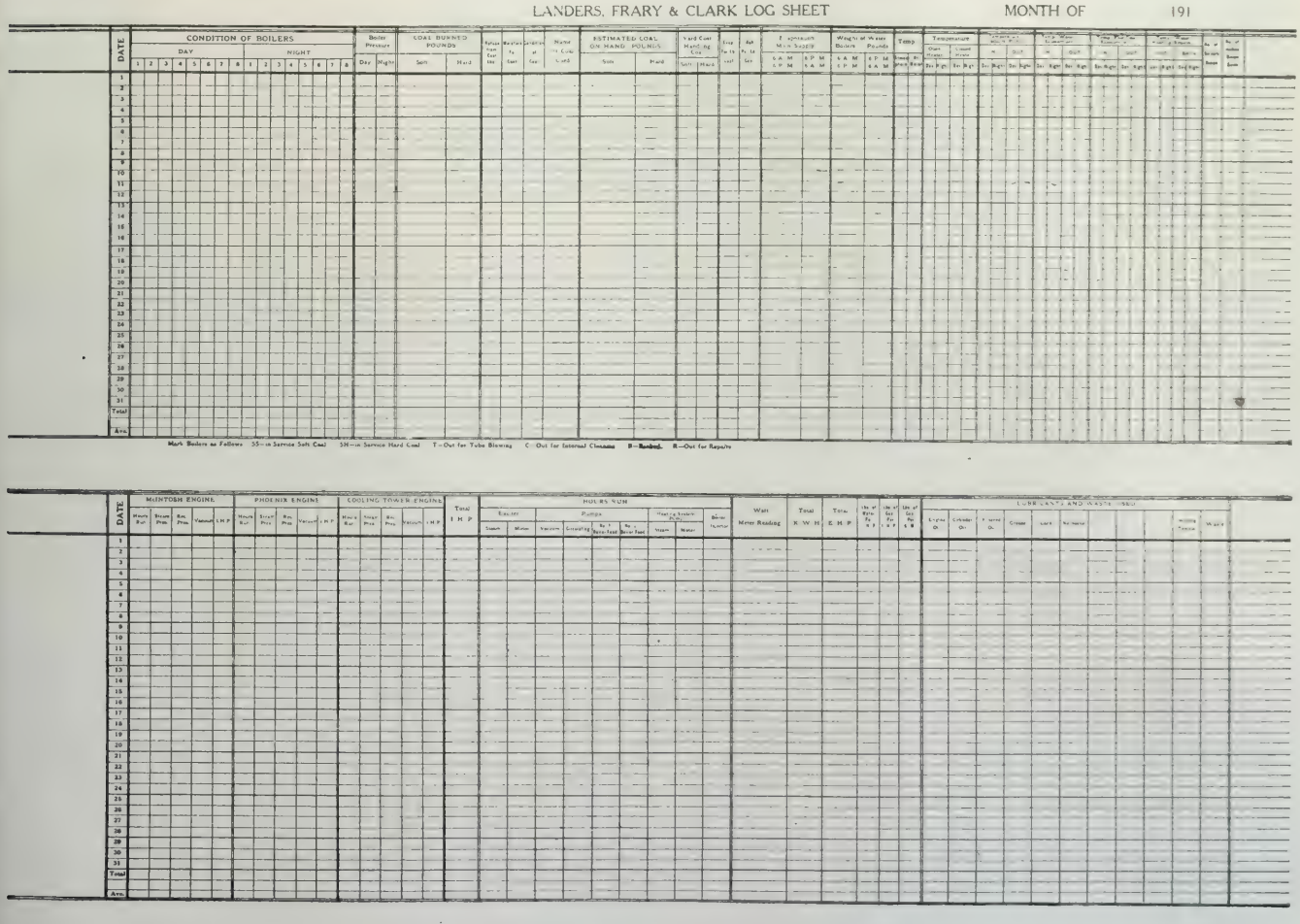


FIG. 4. FORM OF LOG SHEET USED AT THE PLANT

power-plant operation is confined to one sheet instead of a number of smaller ones. This enables obtaining all necessary data without looking over several reports and makes a very compact and satisfactory log.

The engineer or auditor can tell at a glance when variations in coal consumption appear, and if there is not a corresponding increase in the kilowatt-hour readings, it is seen that a leak is occurring at some point.

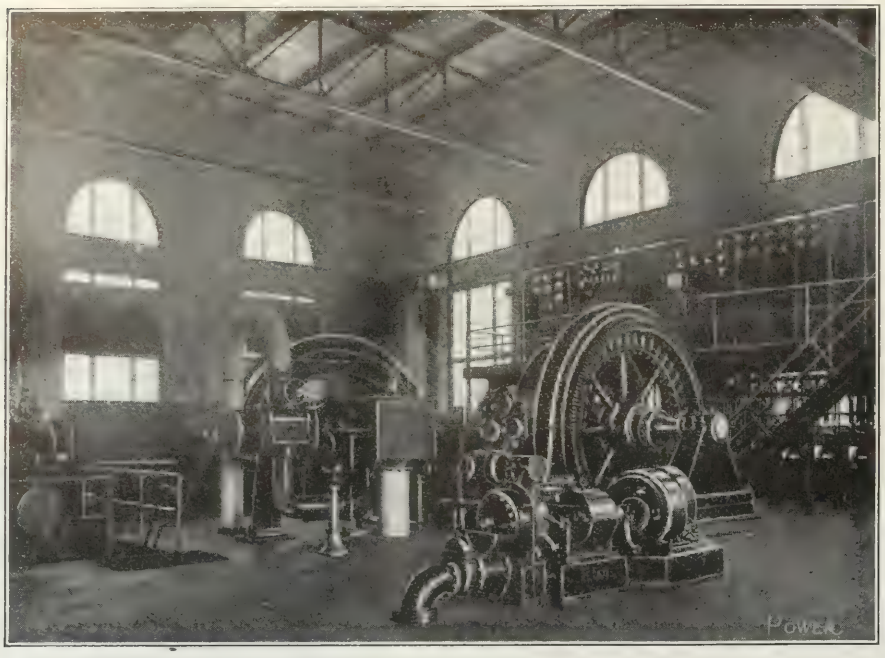


FIG. 3. PARTIAL VIEW OF THE ENGINE ROOM

Records are given of the hours the boilers and engines are in service, temperatures of the circulating water to and from the cooling tower, and the temperature of the feed water passing through the heater and economizer, as well as a list of the minor power-plant supplies used.

A Simple Problem in Horsepower

BY WALLACE S. FOWLER

This subject is as familiar to every engineer as everyday arithmetic to the college man. Nevertheless, the same cause that has led many a technical graduate to dig out his textbook when suddenly confronted by a problem in cube root, might some day overtake the engineer who has been asked to compute the brake horsepower of a proposed engine and finds himself a little "rusty" in first principles. To be sure he might have a dozen catalogs of reliable concerns manufacturing engines, and yet fail to find in their tables just the conditions he is looking for. The following formulas and table may be of service in such a case:

$$B.hp. = \frac{m.e.p. \times L \times A \times 2N}{33{,}000} \times E \quad (1)$$

where

$B.hp.$ = Brake horsepower;
$m.e.p.$ = Mean effective pressure;
L = Length of stroke in feet;
A = Area of the cylinder (area of the rod neglected);
N = Revolutions per minute;
E = Mechanical efficiency of the engine.

The mean effective pressure may be obtained from the following simple formula:

$$m.e.p. = F\ (V - B.P.) \quad (2)$$

where

F = Assumed card factor;
V = Mean forward absolute pressure given in table;
$B.P.$ = Absolute back pressure in the exhaust pipe

Example: Required the brake horsepower of an engine with 16x18-in. cylinder, running at 250 r.p.m. under 150 lb. gage steam pressure, cutting off at one-third stroke; when the vacuum in the condenser is 27 in., the assumed card factor 80 per cent., and the assumed mechanical efficiency 90 per cent. For 150 lb. pressure and one-third cutoff V from the table equals 115.21. The back pressure in the exhaust pipe includes the absolute pressure in the condenser (which is approximately one-half the difference between the given vacuum and perfect vacuum) plus an allowance for friction in the exhaust passages. In this problem assume the friction loss to be 2½ lb. Then the back pressure would equal

$$\frac{30 - 27}{2} + 2\tfrac{1}{2} = 4\ lb.\ absolute$$

Substituting in formula (2) gives

$m.e.p. = 0.80\ (115.21 - 4) = 88.968 = 89$

The area of a 16-in. cylinder is 201 sq.in. Then from formula (1)

$$B.hp. = \frac{89 \times 1.5 \times 201 \times 500 \times 0.90}{33{,}000} = 365.9$$

If the same engine were to operate noncondensing the back pressure would be 14.7 lb. abs. plus 2½ lb. friction, or 17.2 lb. abs., making

$m.e.p. = 0.80\ (115.21 - 17.2) = 78.4\ lb.$

and

$$B.hp. = \frac{78.4 \times 1.5 \times 201 \times 500 \times 0.90}{33{,}000} = 322$$

MEAN FORWARD ABSOLUTE PRESSURES CORRESPONDING TO DIFFERENT CUTOFFS

Initial Pressure lb. gage	Cutoff in fractions of stroke					
	¼	⅓	⅜	½	⅝	¾
15	17.72	20.77	22.05	25.14	27.28	28.70
20	20.70	24.27	25.76	29.37	31.88	33.53
25	23.68	27.77	29.47	33.61	36.47	38.36
30	26.67	31.27	33.18	37.84	41.06	43.19
35	29.65	34.76	36.89	42.07	45.66	48.02
40	32.63	38.26	40.60	46.30	50.25	52.86
45	35.61	41.76	44.31	50.54	54.85	57.69
50	38.60	45.26	48.03	54.77	59.44	62.52
55	41.58	48.76	51.74	59.00	64.03	67.35
60	44.56	52.25	55.45	63.23	68.63	72.18
65	47.55	55.75	59.16	67.47	73.22	77.01
70	50.53	59.25	62.87	71.70	77.81	81.85
75	53.51	62.74	66.58	75.93	82.41	86.68
80	56.50	66.24	70.30	80.16	87.00	91.51
85	59.48	69.74	74.01	84.40	91.59	96.34
90	62.46	73.24	77.72	88.63	96.19	101.17
95	65.44	76.73	81.43	92.86	100.78	106.00
100	68.43	80.23	85.14	97.09	105.37	110.83
105	71.41	83.73	88.85	101.33	109.97	115.67
110	74.39	87.23	92.56	105.56	114.56	120.50
115	77.37	90.72	96.28	109.79	119.15	125.33
120	80.36	94.22	99.99	114.02	123.75	130.16
125	83.34	97.72	103.70	118.26	128.34	134.99
130	86.32	101.22	107.41	122.49	132.94	139.82
135	89.31	104.71	111.12	126.72	137.53	144.66
140	92.29	108.21	114.83	130.95	142.12	149.49
145	95.27	111.71	118.55	135.19	146.72	154.32
150	98.25	115.21	122.26	139.42	151.31	159.15

All speed records for large ships have been lowered by the new British battle-cruiser "Princess Royal," which recently returned to Devonport on completing her experimental trials. She obtained a speed of 34 knots, or 39 miles an hour, which constitutes a world's record.

Steam Power Plant Pipe Fittings

By William F. Fischer

Information is given regarding the proper use of various weights and material of fittings for power plant piping.

Cast-iron pipe should never be used for high-pressure steam mains because of its treacherous nature. It is extensively used for underground water and gas mains, sanitation piping, intake water mains, condenser work, etc. It is also used, to some extent, for high-pressure, boiler-feed mains, as it resists corrosion better than wrought-iron or steel pipe and is cheaper.

It should be of full, extra-heavy weight with extra-heavy flanges, and designed for a bursting pressure of not less than six times the greatest working pressure employed. If the working pressure is to be 200 lb. per sq.in., the pipe should not burst at less than 1200 lb. per sq.in., and a much larger factor of safety is desirable. A cast-iron feed pipe should be well supported throughout its length, because of its excessive weight, placing the pipe hangers or brackets about 12 ft. apart.

Cast-iron pipe of various weights and thicknesses may be had with flanged ends or bell and spigot ends. The latter are generally used for underground work, making a tight job when properly put together, calked and leaded. For exposed piping flanged ends are used and the joints made up with gaskets. When ordering cast-iron flanged pipe the working pressure should be stated, also the flange dimensions and drilling.

Table 1 gives the thickness and weights of cast-iron pipe of various sizes for working pressures of from 25 to 150 lb. per sq.in., or the equivalent hydraulic head as stated.

Table 2 gives the thickness and weights per foot of cast-iron pipe for the heavier pressures, up to 347 lb. per sq.in. The working pressure should always be specified, as the manufacturers' standard dimensions vary slightly.

The weights given in these tables do not include the weight of the flanges, but for estimating the weight of the two flanges on each piece of pipe may be taken as equivalent to the weight of a running foot of the pipe.

The following simple formulas may be used in calculating the pressures and thickness of cast-iron pipe:

Let

W = Safe working pressure in lb. per sq.in.;

P = Average bursting pressure in lb. per sq.in.;

D = Inside diameter of pipe in in.;

t = Thickness of metal in in.;

F = Factor of safety ($P \div W$);

f = Unit fiber stress in pipe material in lb. per sq.in.

Then, based on an ultimate tensile strength of 20,000 lb. for cast iron,

$$P = \frac{40{,}000\,t}{D} \qquad (1)$$

$$W = \frac{40{,}000\,t}{DF} \qquad (2)$$

$$t = \frac{WDF}{40{,}000} \qquad (3)$$

$$f = \frac{WD}{2\,t} \qquad (4)$$

The following examples illustrate the use of the formulas:

At what pressure will a piece of 7-in. cast-iron pipe 0.6 in. thick, burst?

Using formula 1,

$$P = \frac{40{,}000\,t}{D} = \frac{40{,}000 \times 0.6}{7} =$$

3429 lb. per sq.in. nearly.

In Table 1 this pipe is recommended for a working pressure of 150 lb. per sq.in.; then the factor of safety will be,

$$F = \frac{3429}{150} = 22.8 +$$

With a factor of safety ($F = 10$), what would be the safe-working pressure (W) of a 7-in. pipe 0.6 in. thick?

From Formula 2,

$$W = \frac{40{,}000\,t}{DF} = \frac{40{,}000 \times 0.6}{7 \times 10} =$$

343 lb. per sq.in.

For a working pressure $W = 343$ lb. and a factor of safety of 10, what thickness of metal would be required for a 7-in. pipe?

Using Formula 3,

$$t = \frac{WDF}{40{,}000} = \frac{343 \times 7 \times 10}{40{,}000} = 0.6\ in.$$

Under a working pressure of 343 lb. per sq.in., what would be the unit fiber stress in the metal of a 7-in. cast-iron pipe of 0.6 in. thickness?

From Formula 4,

$$f = \frac{WD}{2\,t} = \frac{343 \times 7}{2 \times 0.6} = 2000\ lb.\ per\ sq.\ in.$$

Cast-iron pipe is regularly made in 12-ft. lengths, and always thicker than necessary to withstand the intended pressures, so as to safely resist all stresses and shocks likely to occur in transit, in handling and in erecting.

*TABLE I. THICKNESS AND WEIGHTS OF CAST IRON PIPE

Inside Diameter in Inches	Thickness in Inches				Approximate Weight per Foot, in Pounds. (Weight of Flanges not included.)			
	60 Ft. Head 25 Lbs. Pressure	115 Ft. Head 50 Lbs. Pressure	230 Ft. Head 100 Lbs. Pressure	350 Ft. Head 150 Lbs. Pressure	60 Ft. Head 25 Lbs. Pressure	115 Ft. Head 50 Lbs. Pressure	230 Ft. Head 100 Lbs. Pressure	350 Ft. Head 150 Lbs. Pressure
3	0.39	0.41	0.43	0.45	13.0	13.7	14.5	15.2
4	0.40	0.43	0.46	0.49	17.3	18.7	20.1	21.6
*5	0.42	0.46	0.49	0.53	22.3	24.7	26.4	28.7
6	0.44	0.48	0.52	0.56	27.8	30.5	33.3	36.0
*7	0.46	0.52	0.56	0.60	33.7	38.3	41.5	44.7
8	0.47	0.53	0.59	0.64	39.0	44.3	49.7	54.2
*9	0.49	0.55	0.62	0.68	45.6	51.5	58.5	64.5
10	0.51	0.58	0.65	0.71	52.5	60.2	67.9	74.5
12	0.54	0.63	0.71	0.79	66.4	78.0	88.5	99.0
14	0.58	0.66	0.77	0.86	82.9	94.9	111.5	125.3
*15	0.60	0.70	0.80	0.90	91.8	107.7	123.9	140.3
16	0.61	0.72	0.84	0.95	99.3	118.0	138.7	157.9
18	0.65	0.77	0.90	1.02	118.8	141.7	166.7	190.2
20	0.67	0.82	0.96	1.09	135.8	167.4	197.3	225.0
22	0.72	0.87	1.02	1.18	160.4	195.0	230.2	268.1
24	0.75	0.92	1.08	1.25	182.0	224.7	265.5	309.4
30	0.86	1.07	1.28	1.48	260.2	325.9	392.5	456.7
36	0.96	1.21	1.46	1.71	347.8	442.3	536.1	632.0
40	1.03	1.31	1.59	1.87	414.2	530.4	648.2	767.5
42	1.07	1.36	1.65	1.93	451.7	578.0	705.9	835.5
48	1.18	1.51	1.84	2.17	568.8	732.8	898.9	1067.1

Note 5, 7, 9 and 15 inch pipe are odd sizes, made to order only.
4" inch to 24 inch regularly made in 12 ft. lengths faced $\frac{1}{16}$ inch short for gasket.
*From R. D. Wood & Co. Catalogue.

Riveted Steel Pipe

Riveted steel pipe is made up of straight or spiral riveted plates; the first is known as straight-riveted; the latter as spiral-riveted pipe. They are furnished with either pressed- or forged-steel flanges riveted to the ends. The latter are preferable, being heavier and stronger. Such pipe is usually galvanized to prevent corrosion and to make the riveted joints tighter after calking. It is used extensively in noncondensing power-plant work for exhaust-steam mains, suction mains, large drains, etc., but should never be used for live steam, feed water or similar high-pressure work. It may be had in lengths up to 20 ft.

Spiral-riveted pipe, because of its construction, is stronger than straight-riveted pipe of equal size, thickness and quality. The single seam in spiral-riveted pipe being continuous, and helical from end to end of the pipe, stiffens it throughout its length.

In Table 3 is given the bursting pressure in pounds per square inch for spiral-riveted pipe of various sizes and thicknesses. The safe working pressure employed should never exceed 25 per cent. of the bursting pressure as listed in the table; in other words, a factor of safety

$$P = \frac{120{,}000\ t\ e}{D} \quad (5)$$

$$W = \frac{120{,}000\ t\ e}{D F} \quad (6)$$

$$t = \frac{W D F}{120{,}000\ e} \quad (7)$$

TABLE II. THICKNESS AND WEIGHTS OF CAST IRON PIPE FOR HIGH PRESSURE SERVICE

Inside Diameter, Inches	Thickness of Metal in Inches					Approximate Weight per Foot, in Pounds				
	400 Ft. Head 173 Lb. Pressure	500 Ft. Head 217 Lb. Pressure	600 Ft. Head 260 Lb. Pressure	700 Ft. Head 304 Lb. Pressure	800 Ft. Head 347 Lb. Pressure	400 Ft. Head 173 Lb. Pressure	500 Ft. Head 217 Lb. Pressure	600 Ft. Head 260 Lb. Pressure	700 Ft. Head 304 Lb. Pressure	800 Ft. Head 347 Lb. Pressure
6	0.55	0.58	0.61	0.65	0.69	38.3	41.7	43.3	47.1	49.6
8	0.60	0.66	0.71	0.75	0.80	55.8	61.7	65.7	70.8	75.0
10	0.68	0.74	0.80	0.86	0.92	76.7	86.3	92.1	100.9	106.7
12	0.75	0.82	0.89	0.97	1.04	100.0	113.8	122.1	135.4	143.8
14	0.82	0.90	0.99	1.07	1.16	129.2	145.0	157.5	174.2	186.7
16	0.89	0.98	1.08	1.18	1.27	158.3	179.6	195.4	219.2	232.5
18	0.96	1.07	1.17	1.28	1.39	191.7	220.4	238.4	267.1	286.7
20	1.03	1.15	1.27	1.39	1.51	229.2	263.0	286.3	320.8	344.6
24	1.16	1.31	1.45			306.7	359.6	392.9		
30	1.37	1.55	1.73			450.0	521.7	585.4		
36	1.58	1.80	2.02			625.0	725.0	820.0		

*From U. S. Cast Iron Pipe & Foundry Co.

of not less than four should be used, and then only when the pressure is practically uniform.

For pumping mains and similar piping where the pressure is apt to increase suddenly, or where shocks from water-hammer are probable, a larger factor of safety should be employed, not less than 6 or 8, depending upon the conditions. To find the safe-working pressure in pounds per square inch from the table, divide the given bursting pressure by the factor of safety. When ordering riveted pipe the gage or thickness of metal should always be specified.

The following formulas may be used in calculating the pressure and thickness of riveted steel pipe:

Where

P = Approximate bursting pressure in lb. per sq.in.;
W = Safe working pressure in lb. per sq.in.;
D = Inside diameter of pipe in in.;
t = Thickness of metal in in.;
F = Factor of safety = $P \div W$;
e = Efficiency of riveted joint in per cent.

*TABLE III. THICKNESS AND BURSTING PRESSURES OF SPIRAL RIVETED STEEL PIPE

Inside Diameter, Inches	Thickness, U. S. Standard Guage								
	20	18	16	14	12	10	8	6	3
	Approximate Bursting Pressure, Pounds per Sq. Inch								
3	1500	2000							
4	1125	1500	1875						
5	900	1200	1500						
6		1000	1250	1560	2170				
7		860	1070	1340	1860				
8		750	935	1170	1640				
9			835	1045	1460				
10			750	935	1310				
11			680	850	1200				
12			625	780	1080	1410			
13			575	720	1010	1295			
14			535	670	940	1210			
15				625	875	1125			
16				585	820	1050	1290	1520	1880
18				520	730	940	1140	1360	1660
20				470	660	840	1030	1220	1500
22				425	595	765	940	1108	1364
24				390	540	705	820	1015	1250
26					505	650	795	935	1154
28					470	605	735	870	1071
30					435	560	685	810	1000
32					410	525	645	760	940
34					380	490	600	715	880
36					365	470	570	680	830
40					330	420	515	610	750
	Approx. Thickness in Decimal Parts of an Inch								
	0.0375	0.05	0.0625	0.078	0.109	0.141	0.172	0.203	0.25

*(American Spiral Pipe Works.) (Condensed from Catalogue.)

The tensile strength of the steel plate varies from about 50,000 to 70,000 lb. per sq.in. The foregoing formulas are based on an average ultimate tensile strength of 60,000 lb. per sq.in. for steel plates. The efficiency of riveted joints in straight-riveted pipe varies from 50 to 85 per cent; average 70 per cent.

Brass and Copper Pipes

Brass pipe for power-plant work should be made to the same dimensions as wrought-iron and steel pipe, and threaded to the wrought-iron standard. When ordering brass pipe, iron pipe size should always be specified; otherwise the manufacturer might supply the lighter weight, thin-brass tubing, commonly used for ornamental work, brass hand railings, etc.; it is not of the proper size to take the standard pipe thread and is too thin for pressure work.

Brass pipe is more expensive than wrought-iron or steel pipe and is seldom used in large sizes, but in the small sizes is used for connecting boiler gages, feed-water connections between the feed main and boilers and also for hot-water service in general. It is not subject to corrosion. The average ultimate tensile strength is about 18,000 lb. per sq.in., or about one-third that of mild steel pipe of equal dimensions.

Copper pipe is seldom, if ever, used for power-plant work, except in the very small sizes, such as oil and lubricator piping, etc. It is very unreliable for high steam pressures, and deteriorates very rapidly under high temperatures and repeated stresses. At a temperature of 360 deg. F. its strength is reduced 15 per cent. and on this account it should never be used in steam power-plant work for high pressures or temperatures.

Effect of Boiler Scale

Tests and experiments made by the engineering department of the University of Illinois show the possibilities of heat loss from mineral incrustations inside of boilers varying in thickness from $\frac{1}{32}$ to $\frac{1}{8}$ in. The tests brought out the following results:

Character of Scale	Thickness	Composition	Loss
Hard	$\frac{1}{32}$ in.	Mostly carbonates	9.4 per cent.
Soft	$\frac{1}{32}$ in.	Mostly carbonates	7.2 per cent.
Hard	$\frac{1}{16}$ in.	Mostly carbonates	8.5 per cent.
Soft	$\frac{1}{16}$ in.	Mostly carbonates	8.0 per cent.
Hard	$\frac{3}{32}$ in.	Mostly sulphate	9.3 per cent.
Hard	$\frac{1}{8}$ in.	Mostly sulphate	11.1 per cent.
Soft	$\frac{1}{8}$ in.	Mostly sulphate	10.8 per cent.
Soft	$\frac{1}{8}$ in.	Mostly carbonates	11.0 per cent.
Soft	$\frac{1}{8}$ in.	Mostly carbonates	12.4 per cent.

This table illustrates how good evaporative results may be handicapped not only 12 per cent., but even more by allowing incrustations to increase beyond the $\frac{1}{16}$-in. thickness.

Farmers near Fort Wayne, Ind., are supplied with electricity by the central station at that point at the rate of 10c. per kilowatt-hour. The customer is obliged to furnish his own transformer, meter and all other equipment and pay a minimum of $2 per month meter charge.—*Gas Review.*

Spontaneous Heating of Stored Coal

The loss by escape of volatile matter has been greatly overestimated and reduction of heat value from weathering is small.

Storage under fresh or salt water to save calorific value is not justified but may prevent spontaneous combustion.

Causes of spontaneous combustion and how to prevent it in storing coal.

Owing to the many requests for exact information on the deterioration and spontaneous heating of coal, the Bureau of Mines has published a preliminary paper by Horace C. Porter and F. K. Ovitz, giving a summary of the results thus far obtained. An abstract of this paper follows:

A study was made of the loss of volatile matter from crushed coal during storage. The coal tested was in 20-lb. samples and represented a variety of kinds from widely separated fields. Each sample was broken to about half-inch size at the mine, placed as quickly as possible in a large glass bottle and shipped to the laboratory. At the laboratory the accumulated gas was withdrawn and the volatile products were permitted to escape freely and continuously at atmospheric pressure and temperature. The results of these experiments have been published in a technical paper of the Bureau of Mines, and will not be given here in detail. Suffice it to say that, although several coals evolve methane in large volumes, especially shortly after mining, the coals tested lost in one year from this cause only 0.16 per cent. at most of their calorific value. It seems, therefore, that the loss caused by escape of volatile matter from coal has been greatly overestimated.

At the instance of the Navy Department more elaborate tests were undertaken to determine the total loss possible in high-grade coal by weathering. The extent of the saving to be accomplished by water submergence as compared with open-air storage was to be settled, as was also the question whether salt water possessed any peculiar advantage or disadvantage over fresh water for this purpose.

Outline of Tests

In brief, the tests by the bureau were carried out as follows: Four kinds of coal were chosen—New River, W. Va., on account of its large use by the Navy; Pocahontas, Va., from its wide use as a steaming and coking coal in the Eastern states, and the principal fuel used in the Panama Canal work; Pittsburgh coal, as a rich coking and gas coal, and Sheridan, Wyo., sub-bituminous coal or "black lignite," as a type much used in the West.

In the tests with the New River coal, 50-lb. portions were made up out of one large lot, which had been crushed to quarter-inch size and well mixed, and portions, confined in perforated wooden boxes, were submerged under sea water at three navy yards that differed widely from one another in climatic conditions. Portions of 300 lb. each from the same original lot were exposed to the open air, both outdoors and indoors, at the same navy yards.

The Pocahontas coal was tested at only one point, the Isthmus of Panama, where run-of-mine coal was placed in a 120-ton pile exposed to the weather. Pittsburgh coal was stored as run-of-mine at Ann Arbor, Mich., both in open 5-ton bins outdoors and in 300-lb. barrels submerged in fresh water. The Wyoming sub-bituminous coal was stored at Sheridan, Wyo., both as run-of-mine and slack, in outdoor bins that held 3 to 6 tons each.

Duplicate samples were taken at each sampling, and the sampling of every lot, except the outdoor pile at Panama and the 300-lb. open-air piles at the navy yards, was done by rehandling all the coal. In the excepted cases it was not thought fair to disturb the entire lot. Therefore at Panama a vertical section of the pile, weighing 10 tons, was removed each time; eight gross samples were taken from this 10-ton portion, and each of the samples was quartered down. From the outdoor piles at the navy yards a number of small portions were taken from well distributed points in each pile, mixed, and quartered down. Small lots, finely divided, were purposely chosen with the New River coal, to make the tests most severe.

A fact the authors emphasize is that physical deterioration of the coal was not tested, such as increase of friability, or slacking of lumps. Consequently the results do not show how the relative availability of heat units may have been affected by physical changes; but they do show how much the calorific value, as determined in the laboratory, was affected by storage of the coal under the several conditions.

CALORIFIC VALUE IN B.T.U. PER POUND OF THE COAL AS FIRED, MOISTURE AND ASH FREE COAL BEFORE AND AFTER STORAGE.

Kind of Coal	Storage Conditions	Place	Calorific Value: As Stored	After 6 Months	After 1 Year	After 2 Years	Loss in 2 Years
New River, W. Va.	1-inch coal, under sea water	Portsmouth, N.H.	8703		8779	8781	0.09
		Norfolk, Va.	8774		8768	8737	0.46
		Key West, Fla.	8787		8772	8777	0
	1-inch coal, under fresh water.	Pittsburgh, Penn.	8752	8762	8769	8764	0
	1-inch coal, exposed to air indoors	Portsmouth, N.H.	8779	8738	8762	8739	0.46
		Norfolk, Va.	8758	8762	8736	8717	0.39
		Key West, Fla.	8774	8728	8770		
		Pittsburgh, Penn.	8799	8766	8719	8726	0.46
	R. o. m. coal, exposed to air indoors.	Portsmouth, N.H.	8774	8758	8748	8734	0.23
		Norfolk, Va.	8749	8764	8726	8726	0.23
		Key West, Fla.	8745	8728	8710		
		Pittsburgh, Penn.	8753	8758	8740	8743	0.11
	½-inch coal, exposed to air outdoors, moistened.	Portsmouth, N.H.	8741	8730	8700	8674	0.77
		Norfolk, Va.	8720	8709	8664	8625	1.08
		Key West, Fla.	8729	8721	8686	8692	1.85
		Pittsburgh, Penn.	8748	8726	8700	8665	1.12
	R. o. m. coal, exposed to air outdoors, moistened.	Portsmouth, N.H.	8772	8760	8730	8722	0.58
		Norfolk, Va.	8746	8745	8720	8706	0.46
		Key West, Fla.	8742	8748	8722	8682	1.29
		Pittsburgh, Penn.	8772	8740	8710	8696	1.04
Pocahontas, Va.	120-ton pile of run-of-mine coal, outdoors.	Panama	8794	8787	8762		
Pittsburgh bed.	4 barrels submerged in fresh water, outdoors.	Ann Arbor, Mich.	8741	8709	8728		
			8741	8754	8528		
Sheridan, Wyo., sub-bituminous.	Run-of-mine, in 5-ton bin	Sheridan, Wyo.	7370	7174		7135	*f* 3.19
	Run-of-mine, in 5-ft. bin	Sheridan, Wyo.	7370	7202		7094	*f* 3.75
	R. o. m., closed bin 15 ft. deep.	Sheridan, Wyo.	7370	7208		7082	*f* 3.28
	Slack, closed bin 15 ft. deep.	Sheridan, Wyo.	7355	7166		7080	*f* 3.66

f Heating value of representative coals of this kind, moisture and mineral free, 8758 calories. a On basis of original heating value of same coal portions. c After six months' storage, equal to 2 yrs. d After six months' storage, equal 3 inches at surface of bin. e Bin contained 5 ton (20 cwt.). *f* After two years and three-quarters.

Results of Tests

The accompanying table is a condensed summary of results from a large number of tests and determinations. The moisture, ash, and sulphur contents and

the calorific value of each sample were determined. To eliminate the effect of incidental impurities that were not concerned in the deterioration of the actual coal substance, the calorific values are calculated to moisture, ash, and sulphur free bases, and the ash as determined is corrected for errors due to oxidation of pyrites and dehydration of shale or clay.

The New River coal showed less than 1 per cent. loss of calorific value in one year by weathering in the open. In two years the greatest loss was at Key West, 1.85 per cent. Practically nothing was lost by the submerged samples in one year, fresh or salt water serving equally well to "preserve the virtues" of the coal. There was almost no slacking of lumps in the run-of-mine samples. In all tests the crushed coal deteriorated more rapidly than the run-of-mine.

The Pocahontas run-of-mine coal in a 120-ton pile on the Isthmus of Panama lost during one year's outdoor weathering less than 0.4 per cent. of its heating value, and showed little slacking of lumps.

Gas coal during one year's outdoor exposure suffered practically no loss of calorific value, measurable by the calorimetric method used, not even in the coal forming the top 6-in. layer in the bins.

The Wyoming coal in one of the bins deteriorated in heat value 5.3 per cent. during storage for two and three-fourths years, and more than 2.5 per cent. in the first three months. There was bad slacking and crumbling of the lumps on the surface of the piles, but even where this surface was fully exposed to the weather the slacking did not penetrate more than 12 to 18 in. in the 2¾-year period.

No outdoor weathering tests of coal of the Illinois kind have been made by the Bureau of Mines, but tests of such coal have been reported by Prof. S. W. Parr, of the University of Illinois, and by A. Bement, of Chicago, both of whom find a calorific loss of 1 to 3 per cent. in a year by weathering. Bement reports a slacking of lumps over 80 per cent. in one test and of about 12 per cent. in another test of small samples. Possibly with the Illinois as with the Wyoming coal, the slacking in a large pile would not penetrate far from the surface.

Storage under water unquestionably preserves the heating value and the physical strength of coal, but such storage practically makes necessary the firing of wet coal, and consequently the evaporation in the furnace of added moisture varying in amount from 1 to 15 per cent. according to the kind of coal. This is an important drawback to underwater storage of coals like the Illinois and Wyoming, which mechanically retain 5 to 15 per cent. of water after draining. High-grade Eastern coals, however, are ordinarily wet down before firing; hence the 2 or 3 per cent. of moisture these coals absorb during storage becomes of little consequence. Submerged storage absolutely prevents spontaneous combustion, and that alone may justify it when the coal is particularly dangerous to store and when large quantities are to be stored; but unless the coal is to be stored longer than one year, storing under water merely to save calorific value by avoiding weathering seems unjustified.

Spontaneous Combustion

Losses of value from spontaneous heating are much more serious than the deterioration of coal at ordinary temperatures. Oxidation proceeds more rapidly as the temperature rises. Beginning at ordinary temperatures, attacking surfaces of particles, and developing heat, it is probably, in some degree, an absorption of oxygen by the unsaturated chemical compounds in the coal substance. In a small coal pile this slowly developed heat can be readily dissipated by convection and radiation and very little rise in temperature results. If the dissipation of the heat is restricted, however, as in a large pile densely packed, the temperature within the pile rises continuously. The rate of oxidation plotted against the temperature makes a curve that rises very rapidly. When the storage conditions allow warming of the coal to about 212 deg. F., the rate of oxidation becomes so great that the heat developed in a given time ordinarily exceeds the heat dissipated, and the temperature rises until, if the air supply is adequate, the coal takes fire. Evidently, therefore, it is important to guard against even moderate heating of the coal, either spontaneous or from an external source. Increased loss of volatile matter and of heating value occurs with a moderate rise of temperature, even though the ignition point is not reached.

Spontaneous combustion is brought about by slow oxidation in an air supply sufficient to support oxidation, but insufficient to carry away all the heat formed. The area of surface exposed to oxidation by a given mass of any one coal determines largely the amount of oxidation that takes place in the mass; it depends on the size of the particles and increases rapidly as the fineness approaches that of dust. Dust is therefore dangerous, particularly if mixed with lump coal of such size that the interstices permit the flow of a moderate amount of air to the interior. Coals differ widely in friability, that is, in the proportion of dust produced under like conditions. In comparative tests samples of Pocahontas, New River and Cambria County, Penn., coals produced nearly twice as much dust (coal through a ⅛-in. screen) as coal mined from the Pittsburgh bed in Allegheny County, Penn. This variation in friability affects the tendency to spontaneous heating.

Ideal conditions for such heating are offered by a mixture of lump and fine coal, such as run-of-mine with a large percentage of dust, in piles where a small supply of air reaches the interior.

Effect of Volatile Matter

High volatile matter does not of itself increase the tendency to spontaneous heating. A letter of inquiry sent by the bureau to more than 2000 large coal consumers in the United States brought 1200 replies. Of these, 260 reported instances of spontaneous combustion and 220 of the 260 gave the name of the coal. The 220 instances were distributed as follows: Ninety-five were in semibituminous low-volatile coals of the Appalachian region, 70 in higher-volatile coals of the same region, and 55 in Western and Middle Western coals. This shows at least no falling behind of the "smokeless" type of coal in furnishing instances of spontaneous combustion, and no cause for placing special confidence in this class of coal for safety in storage.

The high-volatile coals of the West are usually very subject to spontaneous heating, but owe it to the chemical nature of their constituents rather than to the amount of volatile matter they contain. Strange as it may seem, the oxygen content of coal appears to bear a direct relation to the avidity with which coal absorbs oxygen; high-oxygen coals absorb readily, and therefore have a marked tendency to spontaneous combustion.

Effect of Moisture and of Sulphur

The effects of moisture and sulphur on spontaneous heating have been much discussed with wide difference of opinion. Little experimental evidence has been brought to bear on either question, and neither is yet settled. Richters has shown that in the laboratory dry coal oxidizes more rapidly than moist; but most practical users of coal believe that moisture promotes spontaneous heating. In none of the many cases of spontaneous combustion observed by the authors could it be proved that moisture had been a factor. Still the physical effects of moisture on fine coal, such as closer packing together of dust or small pieces may often aid spontaneous heating.

In certain laboratory experiments, where air was passed over samples of coal heated to 248 deg. F., enough heat developed to almost ignite the coal; the coal was then allowed to cool. Subsequent analysis showed practically no increase in the sulphate, or oxidized form of sulphur, and practically no reduction of the total sulphur in the coal. Though the results of these experiments are not entirely conclusive, they indicate that sulphur content contributes very little to the spontaneous heating of coal.

Importance of Freshly Exposed Surfaces

Freshly mined coal and the fresh surfaces exposed by crushing lumps exhibit marked avidity for oxygen, but after a time the surfaces become oxidized, "seasoned," as it were, retarding further action of the air. In practice, coal that has been stored for six or eight weeks and has even become already somewhat heated, if rehandled and thoroughly cooled by the air, seldom heats spontaneously again.

Suggestions in Storing Coal

With full appreciation that any or all of the following suggested precautions may prove impracticable or unreasonably expensive under certain conditions, they are offered as advisable for safety in storing bituminous coal.

1. Do not pile over 12 ft. deep, nor so that any point in the interior of a pile will be over 10 ft. from an air-cooled surface.
2. If possible, store only screened lump coal.
3. Keep out dust as much as possible; to this end reduce handling to a minimum.
4. Pile so that the lump and fine are distributed as evenly as possible; not, as is often done, allowing lumps to roll down from the peak and form air passages at the bottom of the pile.
5. Rehandle and screen after two months, if practicable.
6. Do not store near external sources of heat, even though the heat transmitted be moderate.
7. Allow six weeks' "seasoning" after mining and before storing.
8. Avoid alternate wetting and drying.
9. Avoid admitting air to the interior of a pile through interstices around foreign objects, such as timbers or irregular brickwork, or through porous bottoms, such as coarse cinders.
10. Do not try to ventilate by pipes, or more harm may often be done than good.

Artificial Coal Experiments in Germany

At the annual meeting of the Association of German Chemists, Dr. Bergius, of Hanover, read a paper on his experiments on the manufacture of artificial coal. He heated cellulose, or peat, with water, to a temperature of 340 deg. C., at a pressure of over 100 atmospheres, in a specially constructed apparatus. Both these substances were transformed into a product physically and chemically identical to coal.

At a temperature of 310 deg. C., the process took 60 hours, and at 340 deg. C., the transformation took place in eight hours. The speed of the reaction is doubled with every additional 10 deg. C. of temperature.

Dr. Bergius estimates on this basis that the time necessary for the formation of natural coal, at the temperature of the earth's crust, is 8,000,000 years, a figure approximating that given by geology.

The Wells Variable Speed Power Transmission Device

Speed-reducing gears of the various kinds, belts and cone pulleys, and other speed-changing devices, are open to the objections that fine adjustment of speed is not possible and time is wasted in shutting down to change from one speed ratio to another, not to mention the wear and tear and shock caused by the process.

Frank T. Wells, 1156 First National Bank Building, Chicago, Ill., has designed and applied for patent on a variable-speed power-transmission device which is intended to overcome these objections.

In Fig. 1, showing longitudinal and cross-sections, the transmission disk *A* is loosely mounted on shaft *B* and rigidly keyed to the driven shaft *C*. This disk contains eight cylinders *D* divided into sets of four. The cylinders of each set are connected together by the passages *E*. Working from the cranks *F*, which are integral with the driving shaft but disconnected from the driven, are the pistons *G*. The variable space in the cylinders *D* and passages *E* is filled with oil, glycerin, or some other low-volatile fluid of low viscosity and high lubricating quality. The valves *H* control the flow of the fluid from one cylinder to another. With the valves wide open there is only slight resistance to the flow of the fluid and practically no torsional force is transmitted from the driving shaft to the driven. With the valves partly or fully closed there is resistance to the free action of the pistons and hence some force is transmitted from one shaft to the other. Thus by regulating the valves any ratio of speed may be established between the two shafts, positively, gradually and without shock or jar.

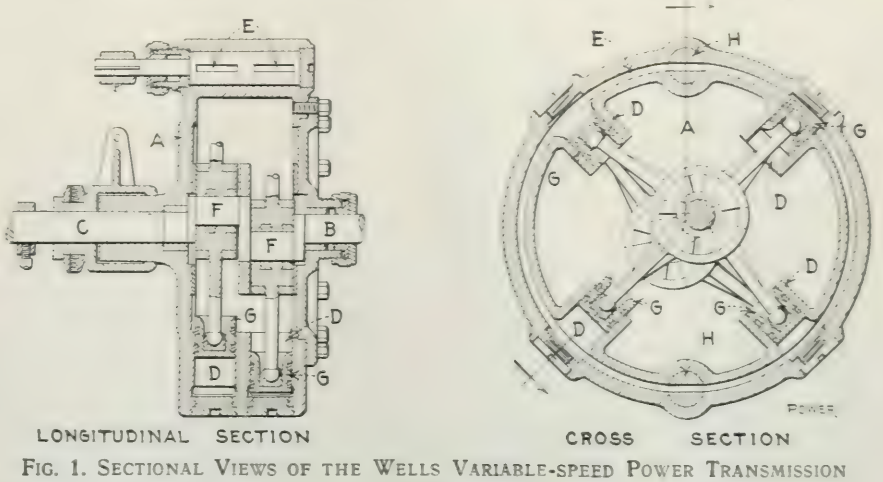


Fig. 1. Sectional Views of the Wells Variable-speed Power Transmission Device

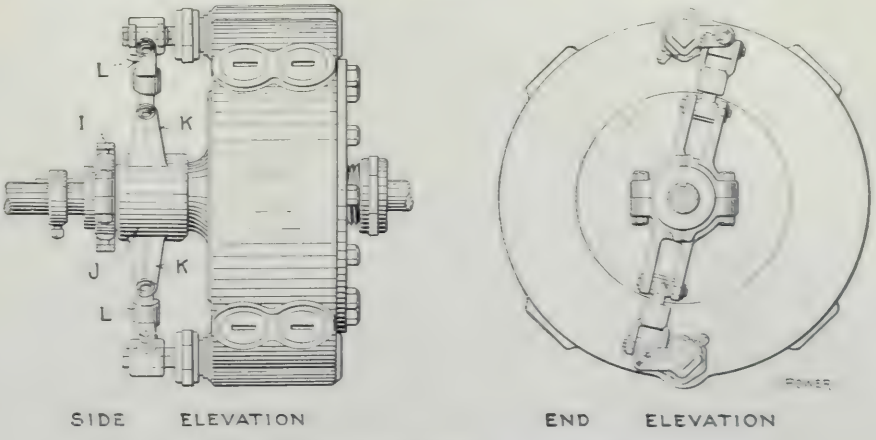


Fig. 2. Exterior Views of the Device

The valves are controlled by a lever, not shown, and the loose collar *I*, Fig. 2, which is attached to the sleeve *J*. By shifting the sleeve *J* along the shaft to the left, the arms *K* and links *L* cause the valves to close. Thus the speed of the driven shaft is regulated simply by shifting the hand lever much or little, according to the speed desired.

Thermometers for the Steam Plant

By J. C. Peebles

Some special thermometers and their receptacles for water and steam lines, condenser, economizer and flue gas temperatures.

In its simplest form the thermometer consists of a glass stem containing a fine capillary tube with a comparatively large bulb at the lower end. The bulb and part of the stem are filled with mercury, the expansion and contraction of which in the bulb, due to changes in temperature, cause its level in the tube to rise or fall. A temperature scale engraved on the glass stem makes it possible to read the temperature for any particular height of the mercury in the tube.

Many modifications of this comparatively simple instrument have been devised, so that the mercury thermometer may be applied to a variety of uses for which the simple glass stem is not well adapted. Some of the special forms for use in the steam plant, and with which the operating engineer should be familiar, will be given attention.

Thermometers are used on the boiler feed-water line, condenser and economizer; on steam lines, particularly for superheated steam; in breeching, uptake or stack for flue-gas temperatures, and for various other purposes where an accurate knowledge of temperature is desired.

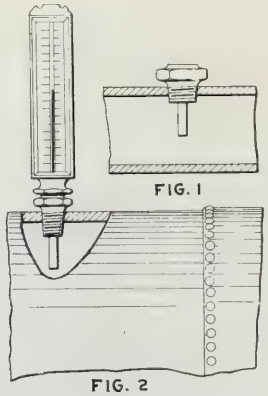


FIG. 1

FIG. 2

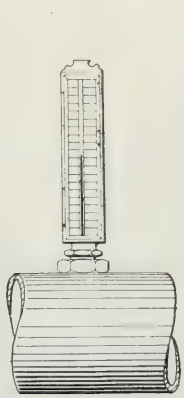


FIG. 3

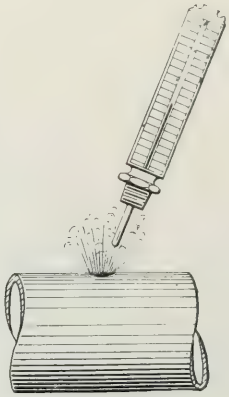


FIG. 4

The first step in the evolution of these special thermometers was the provision of a receptacle into which the simple glass-stem thermometer could be put, when it was desired to find the temperature in pipe lines, etc. Fig. 1 shows a form of special receptacle which is merely a plug with a wrench head at the top and a standard pipe thread below, usually 1 in. in diameter, drilled to receive the thermometer. The hole is partly filled with oil or mercury, except for high temperatures, 600 deg. F. and above, when fine steel filings or precipitated copper is used; the material in the well serves to make contact between the instrument and the wall of the receptacle.

Such an arrangement works well in pipe lines carrying water, brine or low-pressure saturated steam. For lines carrying air or high-pressure superheated steam, the results are not so good, owing to the imperfect nature of the contact between the thermometer and the medium whose temperature is sought. Furthermore, the thermometer is easily broken in such a position, the well becomes filled with dirt, and the mercury or oil distills or evaporates after a time so that the arrangement is unsatisfactory for permanent use.

Fig. 2 shows the next step, the mounting of the thermometer in a metal case, with a fitting at the bottom to screw into the oil well, the well being threaded on the inside to receive it. Instead of the fixed thread shown here, however, a union is better to make the connection, as shown by Fig. 3, so that the thermometer may be screwed home with the scale facing in any direction. This gives a good permanent connection; chances for breakage are minimized, and dirt cannot get into the well. The contact is no better than before; hence the application of the thermometer has not been materially widened. To improve the contact, an attempt was made to do away with the oil well, and connect the thermometer directly to the pipe, by means of its screw connection. This answers fairly well for permanent connections, but the pipe line must be cut out of service if the thermometer is taken out. From Fig. 4 it is evident that this is such a serious objection to this arrangement that it has never had a wide application.

The problem, then, is to produce a thermometer for use in pipe lines, tanks, etc., which shall come quickly and accurately to the temperature of the medium into which it is placed, shall be removable without putting the pipe or tank out of service, shall be free from corrosive action by hot gases, and reasonably proof against mechanical injury. Such a thermometer is shown in Fig. 5, and Fig. 6 shows the bulb end in enlarged section. In Fig. 6 *A* is the thermometer, *B* a thin-walled metal chamber containing a bath of mercury or fine steel filings *C* inclosing the thermometer bulb, *D* a screw to compress the packing in the stuffing-box *E* to hold the thermometer tube in alignment, *F* insulating material to prevent the conduction of heat along the stem, *G* a threaded fitting for attaching the thermometer to its socket and *H* a hexagon wrench head.

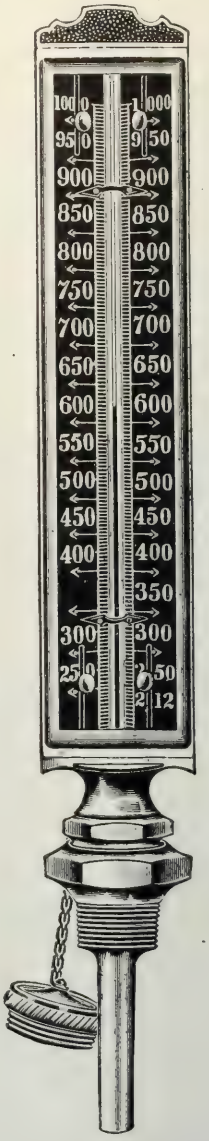


FIG. 5. SPECIAL THERMOMETER FOR PIPE LINES, TANKS, ETC.

Standard sockets of various lengths are made to receive this thermometer, and

are attached permanently to the pipe line. Some of these are shown in Fig. 7. When the thermometer is screwed home, its metal-protecting bulb fits snugly against the inner wall of the socket, giving a direct metallic contact from the outside of the socket to the bulb of the thermometer. This arrangement is not an oil well, for the socket contains no oil, but provides rather for a direct contact. Furthermore, the contact surfaces are made tapering to insure the best possible fit.

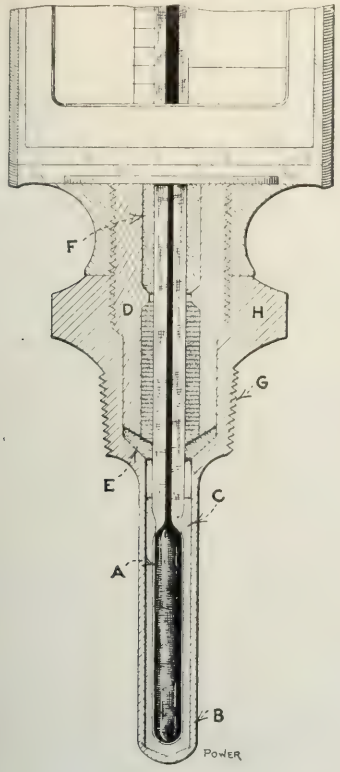


FIG. 6. BULB END OF FIG. 5 IN SECTION

The sockets shown in Fig. 7 are of bronze or steel, and are sometimes coated with lead or other substances best suited to the conditions of use to prevent corrosion. Each has a screw cap attached with a chain; thus the socket is protected from dirt when the thermometer is removed.

One interesting application of this thermometer is as a thermo-steam gage. It can be made with two scales, one for temperature and one for pressure, in which form it constitutes one of the most reliable steam-pressure gages known. In fact, the thermometer has many points of superiority over the Bourdon gage as an indicator of saturated-steam pressures, one of the most important being that it has no moving parts and therefore cannot get out of adjustment.

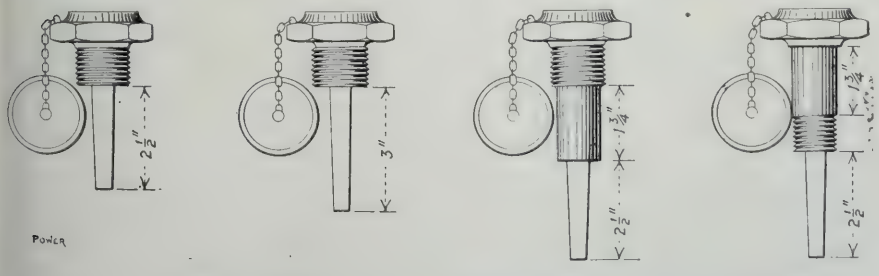


FIG. 7. STANDARD SOCKETS FOR ATTACHING PERMANENTLY TO PIPE LINE

Determining flue-gas temperatures, which is now recognized as important, requires a slightly different thermometer. Fig. 8 shows such an instrument and Fig. 9 the bulb end in section. The construction is similar to that of the pipe-line thermometer, except that no socket is used. The lower end of the instrument is surrounded with a heavy steel wire guard, which allows the hot gases to come directly in contact with the outer metal bulb *H*, but protects it from mechanical injury.

Flue gases at a comparatively high temperature are rather difficult to accurately measure. This is true also of air, superheated steam and many other gases, and is due to the poor contact which the gas offers to the thermometer bulb, because gases are poor heat conductors. To meet this difficulty the thermometer shown in Fig. 9 is sometimes made without the metal-protecting bulb *H* and mercury bath *C*. To protect the bulb, it is electroplated with copper, which is a good heat conductor. This produces a highly sensitive instrument which comes quickly to the temperature of the surrounding medium and is less subject to injury than one with an unprotected bulb.

One interesting instrument which has a variety of applications, is the maximum and minimum thermometer shown in Fig. 10. It is made in several different forms, one of the most serviceable being mounted in a cylindrical brass case less than 2-in. in diameter. In this shape it may be used in dry kilns, ship holds or coal bunkers to give temperature readings in inaccessible places. Stored coal generates heat through the decomposition of sulphur and iron which lowers the heat value and may cause spontaneous combustion. Through a gas pipe driven into the coal pile, a cylindrical maximum-minimum thermometer may be lowered, and the temperature accurately recorded.

This instrument has a double or U-shaped mercury tube, the left limb of which has the low temperatures at the top, while on the right side the high temperatures are at the top. Above the mercury, in each limb of the tube, is a small indicator or float, which is simply a very small piece of glass tubing sealed at both ends, and having a piece of fine steel wire run through the bore from end to end. A tiny spring made from a fiber of blown glass is attached to the indicator with its free end bearing against the inner wall of the mercury tube. When the mercury rises, the little indicator is carried up in the tube, and when it falls

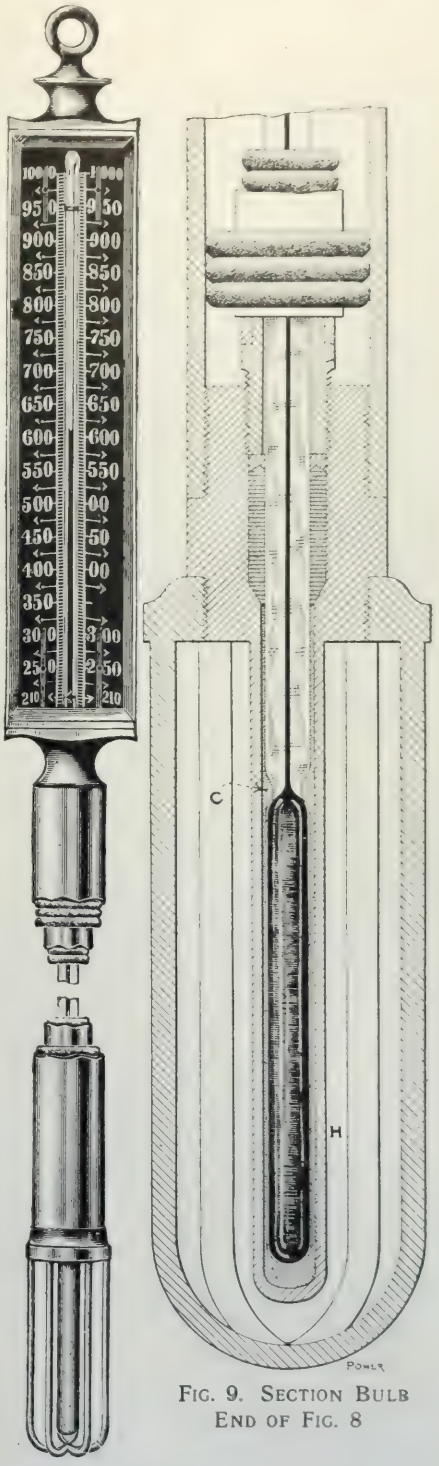


FIG. 9. SECTION BULB END OF FIG. 8

FIG. 8. FOR FLUE GAS TEMPERATURES

the indicator remains at the highest position reached, being held there by the little glass spring. The indicator may be drawn down again by a horseshoe magnet. The part of the tube between the two limbs of the capillary is enlarged into a long bulb and filled with creosote. When the creosote expands the mercury is forced down in the left-hand tube and up in the other. When it contracts the reverse takes place. The bulb at the top of the right capillary is filled with an oil having a low coefficient of expansion as compared with creosote.

Practically all of the simple glass thermometers on the market are pointed and calibrated under a condition of full

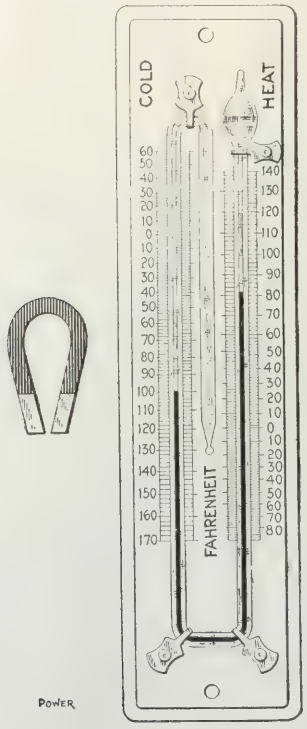


FIG. 10. MAXIMUM AND MINIMUM THERMOMETER

stem immersion. Often, as for flash and fire test of an oil, little more than the bulb is immersed. A thermometer calibrated for full stem immersion will give erroneous results when used with bulb immersion only, the readings being as much as 20 to 30 deg. low at a temperature of 600 deg. F. This has led to the production of a special thermometer having two scales on the stem, one to be used when the bulb only is immersed, and the other when the full stem is immersed. Thus the instrument becomes applicable to either condition of use. Perhaps the only objection which can be made to such a thermometer is the chance for confusion which arises when two scales are used.

When a thermometer is to be used under conditions which give bulb immersion only, it is probably better to use an instrument calibrated for that condition. Thermometers can now be obtained on the market which are correct for bulb immersion or for 1-in., 2-in. or 3-in. immersion as may be desired. A mark is etched on the stem at the point to which it must be immersed to give correct results. This does away with the confusion arising from two scales, although such a thermometer does not have as wide a range of usefulness as the one with the double scale.

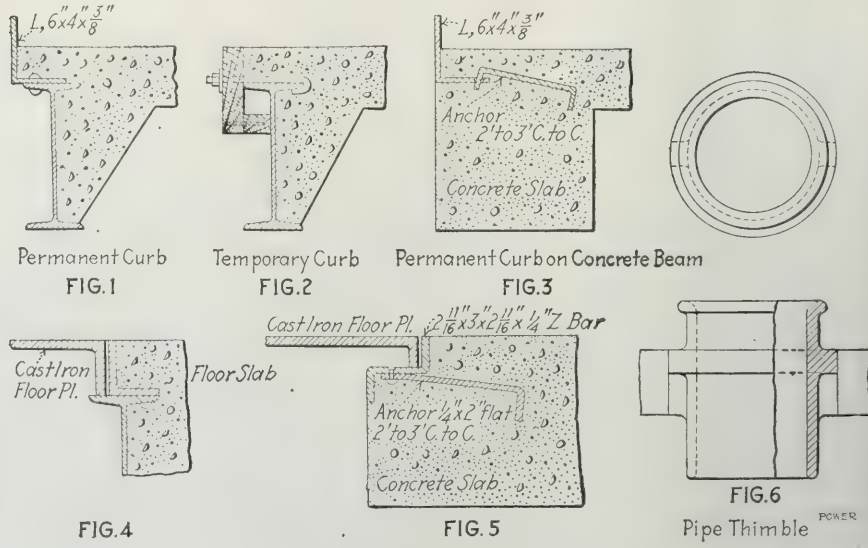
The special thermometers discussed here, except the maximum-minimum type, should all be calibrated for bulb immersion, as that is the condition under which they are used.

Curbs around Floor Openings

BY A. E. DIXON

A power-plant building must have floors or platforms pierced more or less for stair hatches, pipe openings, etc. Occasionally these openings are temporary, to be closed or occupied by later installations of apparatus. Other openings have seats for cast-iron floor slabs. In these days of concrete and steel a few kinks in regard to finishing off around these openings may not be generally known.

Fig. 1 is a curb designed for railed openings. The projection of the angle



above the floor level is to prevent wash water or tools from going over the edge. A curb of this kind is suitable around stairway hatches. When desired the upstanding leg of the angle can be made equal to the thickness of the floor.

Fig. 2 shows a temporary wooden curb which is level with the floor but can be made higher if necessary.

Fig. 3 shows a curb for use with concrete beam or slab floors. Concrete is particularly subject to damage at corners; hence some sort of metal-protecting piece is needed. A further advantage of the metal-curb angle is that it affords a point of attachment for the railing.

Figs. 4 and 5 show curbs for openings which are to be closed up with removable floor plates, Fig. 4 being intended for steel beams with concrete floor slabs, while Fig. 5 shows a construction for use where concrete framing is used. If desired an angle can be substituted for the Z-bar, but the latter is preferable as it gives thorough protection to the concrete.

Fig. 6 shows a cast-iron pipe thimble for setting in a concrete slab. Thimbles of this kind should be large enough to permit the pipe flange to pass through, with some allowance for pipe covering. The projection above the floor prevents sweepings or wash water from going through the opening. Two lugs at opposite sides prevent the thimble from turning and working loose while the collar at the floor level serves to anchor it. Such thimbles may be set in the forms, if they are carefully located to suit the pipes and well watched to prevent their being forced out of place when the concrete is put in.

Goulds New Vertical Centrifugal Pump

A new pump has been added to the line of centrifugal pumps developed by the Goulds Manufacturing Co., Seneca Falls, N. Y. It is of the vertical type, for service where it is necessary to pump from a pit, and can be operated submerged when desirable.

The impeller is of the inclosed type, and the end thrust is practically negligible. A ball-thrust bearing carries the weight of the impeller and shaft. A water-sealing ring is provided, making it unnecessary to draw the shaft gland up tight and insuring low friction losses.

For special liquids which attack iron this pump can be brass fitted or made entirely of brass. It can be belt driven or directly connected to an electric motor.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Old and Modern Arc Switchboards

The progress made during the past 20 years in electrical practice as regards switchboard construction is strikingly shown in the accompanying photographs. Fig. 1 is a view of the high-tension arc switchboard used in the Rockwell Place station of the Brooklyn Edison Co. in 1895, and Figs. 2 and 3 show one of the latest arc switchboards located in the Sumpter St. substation of that company.

The old switchboard was constructed mainly of wood and carried direct current at 2500 volts, supplied by a number of old-style Thomson & Houston arc machines. As will be seen from the illustration, the circuits were closed by inserting plugs to which were attached the insulated leads. The switchboard was handled without rubber gloves, and as the circuits were changed while alive, a considerable arc usually resulted when withdrawing a plug. It is easily conceivable that the constant handling would tend to deteriorate the insulation, especially where the lead joined the plug, thus making it dangerous for the operator and greatly increasing the fire hazard. In addition to this the network of wires at the front of the switchboard tended to complicate its handling and confuse the operator in an emergency.

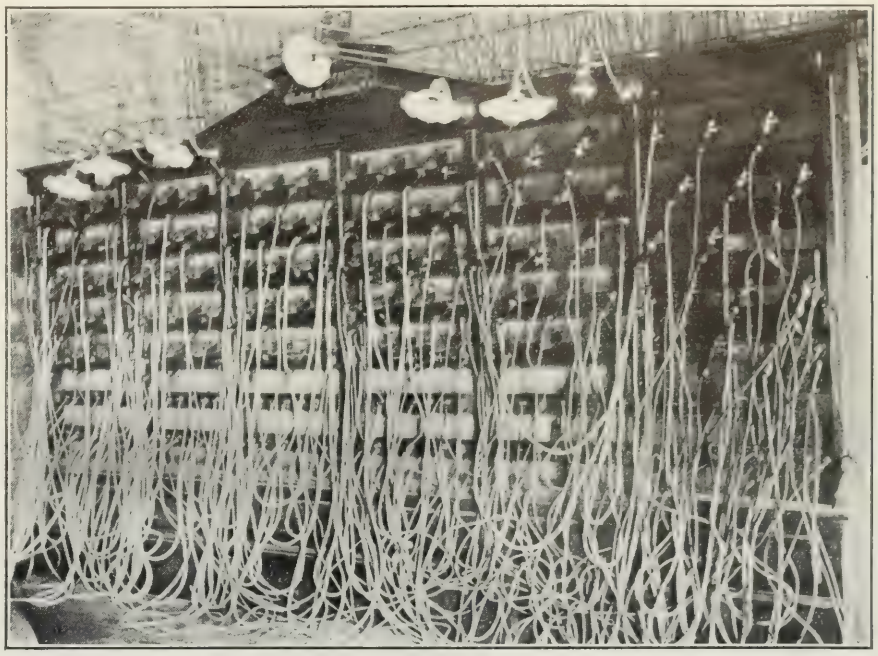


FIG. 1. ARC SWITCHBOARD AT ROCKWELL PLACE STATION, 1895

FIG. 2. ALTERNATING-CURRENT PANELS AND ARC-CIRCUIT PANELS

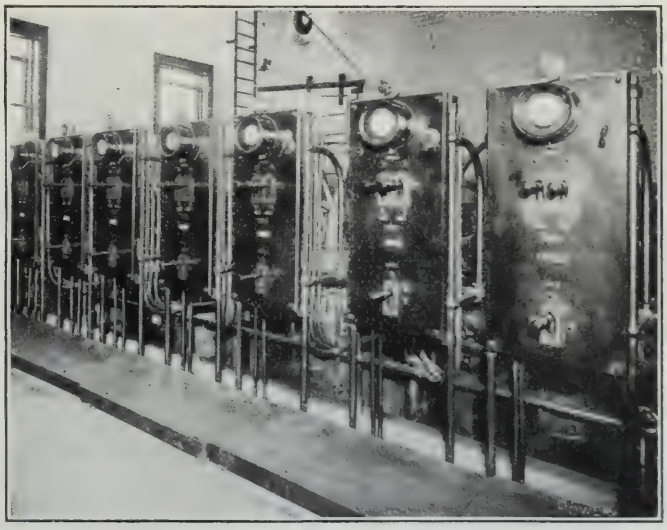


FIG. 3. RECTIFIER PANELS

In strong contrast to this is the modern arc panels. Here the 6600-volt, single-phase alternating current comes in to the oil switches mounted on the panels in the foreground of Fig. 2, passing from here to the rectifiers. The 7500-volt direct current coming from the rectifiers is led to the panels shown in Fig. 3; each rectifier has its own panel containing an ammeter, oil switches, bulb-tilting handle and indicating lamp.

From these panels the current is led to the horizontal busbars at the back of the two panels shown in the background of Fig. 2—one panel for the positive and one for the negative side of the circuit. To the vertical busbars (also not shown) are connected the arc-lighting circuits. Thus by inserting plugs in the receptacles any combination can be had; that is, any circuit can be thrown on any rectifier, or *vice versa*. Also, there is a row of receptacles for detecting grounds and two for instruments. The board has a capacity of 10 circuits and 10 rectifiers, each rectifier having a capacity of 75 lamps.

Why Dynamos Fail to Generate

By J. T. Williams

Among the many causes which make a dynamo fail to generate, loss of residual magnetism is one of the most troublesome. The magnetism that remains in the iron cover after the field current is cut off is called residual magnetism, and every dynamo must have a certain amount of this, before it can build up its field.

As a rule dynamos leaving the factory retain enough residual magnetism to start on, but there are several ways in which they can lose it. They sometimes lose it after having given no previous trouble of that nature in years. Some dynamos never lose their residual magnetism, while others seem to have a weakness for doing so. In many cases this peculiar action cannot be explained, but in some cases it may be due to excessive vibration and for this reason the machine should always rest on a firm foundation. Also the earth's magnetism may exercise a slight influence, and it may be well to set the machine so that its north pole is toward the north, in which case the earth's magnetism will pass through the machine in the same direction as its own. The earth's magnetism is very weak, but when assisted by vibration it is sometimes sufficient to affect the residual magnetism. Only two-pole machines can be set to fill this condition, as the poles on multipolar machines radiate in all directions.

Again, lines of force from a loaded dynamo may tread their way through the magnetic circuit of an idle machine in the reverse direction and neutralize its residual magnetism. The fields may accidentally get a slight flow of current in the wrong direction. This can very easily occur on a compound-wound or series dynamo, as current from the outside will always pass through the series coils in the opposite direction to that in which the current from the machine itself would pass. In shunt-wound machines, this would not happen, for as a motor the field winding shunts both the armature and the line.

If a dynamo were started with the field or armature connections reversed, whatever residual magnetism there may have been would be destroyed. When a field circuit is broken too suddenly the residual magnetism is sometimes brought down to zero or even reversed. When a dynamo loses its residual magnetism, or charge as it is called, the pole pieces will have little or no attraction for a piece of soft iron.

There are several ways in which the charge may be restored. A series dynamo seldom loses its charge so entirely that it will fail to pick up a field on short-circuit. When a compound-wound dynamo refuses to pick up with its shunt field, it can often be made to pick up by disconnecting the shunt coils and short-circuiting the machine through a small fuse that will prevent damage from the short-circuit. The same idea may be used on a plain shunt-wound dynamo by temporarily connecting all the shunt coils in multiple instead of series. A machine can usually be made to pick up a field by simply short-circuiting the armature by holding a piece of copper wire across the brushes or by rocking the brushes back from their neutral position. This has the effect to make the magnetism of the armature help the field to build up.

If none of these remedies produce the desired results, the fields must be recharged from an outside source. If the dynamo runs in multiple with other dynamos, this is an easy matter. It is only necessary to lift the brushes, or disconnect one of the brush-holder cables on the dead machine and throw in the main line switch the same as if the machine were going into service with the others. The fields will then take a charge from the line and the polarity will be correct. After the fields have been charged, the brushes or cable must not be moved until the main line switch is open, or a short-circuit will be made. If the dynamo does not run in multiple with another and there is a dynamo within wiring distance, disconnect the shunt field of the dead dynamo and connect it to the live circuit. Several cells of an ordinary battery may be used for charging the fields. In this case connect the field coils in series with the cells in series, and give the poles pieces of the dynamo repeated knocks with a hammer, while the charging is going on. If this fails, reverse the terminals of the battery and repeat the operation; it may be that the first time the battery is applied its magnetizing force opposes what little residual magnetism there may be in the iron. When all other available sources fail, connect the fields so as to obtain the least possible resistance, put them in series with the armature through a small fuse, and speed the armature considerably above the usual rate.

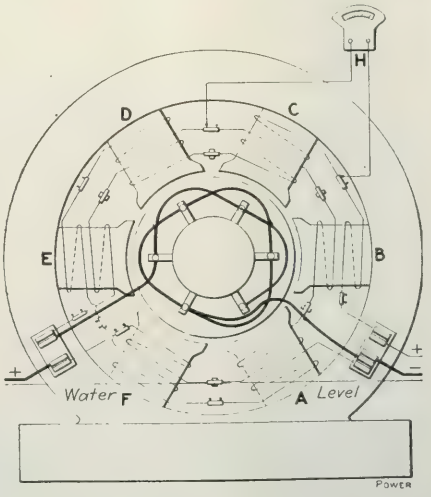
CORRESPONDENCE

Replacing Coils in Motor

A 100-hp. compound-wound, 550-volt direct-current motor was located in a well and direct-connected to a centrifugal pump. During unusually wet weather the water came up to the level indicated in the illustration, causing coil *A* to short-circuit and burn out. This brought on a series of troubles which required almost three days' work to overcome.

Coil *A* was taken out and sent to the shop to be rewound. As all the other coils appeared to be in good condition they were not taken out or tested. We had an extra motor which was a duplicate of the one in trouble, from which a coil was taken and put in place of coil *A*. The motor was started and the duplicate coil began to heat. As the motor from which this coil was taken had not been in use for some time, we assumed that there was trouble in the coil. It was taken out and replaced by coil *A*, which by this time had been repaired; but upon starting the motor, coil *A* began to heat again.

It finally occurred to us that it might be well to do some testing to locate this trouble. Accordingly a voltmeter was connected across the terminals of each field, as shown at *H;* then having disconnected the leads of the series winding, the current was put through the



Motor with Coils Indicated

shunt winding and the drop through each coil was measured; the result was:

Coil *A* = 130 volts.
Coil *B* = 100 volts.
Coil *C* = 97 volts.
Coil *D* = 98 volts.
Coil *E* = 100 volts.
Coil *F* = 15 volts.

From this we judged that a drop of about 100 volts was right and that coils *A* and *F* were wrong. Coil *F* was taken out and found to be water-soaked. Another coil was put in its place with coil *A* still in, and the motor started, but coil *A* continued to heat. The drop was measured again and all the coils were found to give close to 100 volts except coil *A*, which still showed 130 volts.

Coil *F*, having been repaired, was inserted in place of coil *A*, after which the motor was started and ran all right.

It was then discovered that a mistake had been made in rewinding coil *A* by giving it too much resistance. Our trouble might have been lessened if we had taken the trouble at first to make the voltage test. Furthermore, when a field coil is changed it is always best to test the polarity of the poles. This polarity test

is made by sending current through each coil and noting the polarity with a small pocket compass. If correct, each alternate pole will be of the same polarity. After work of this kind has been finished, the motor should be given a final test for grounds.

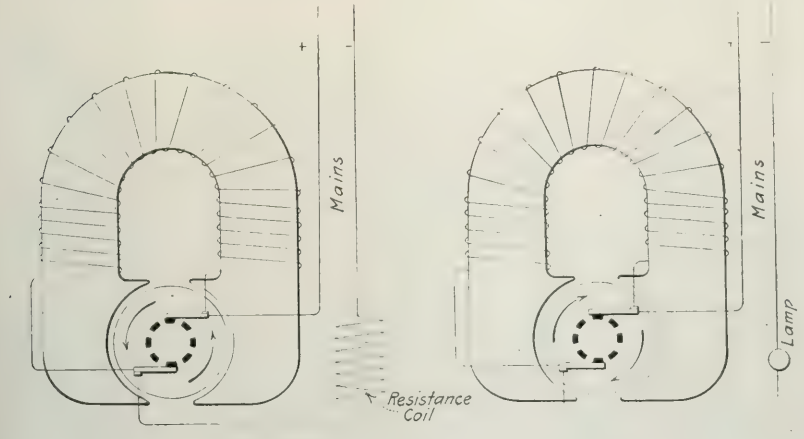
FREDERICK L. RAY.

Louisville, Ky.

What Reversed the Motor?

A ¼-hp., 110-volt direct-current motor taking current from central-station mains and directly connected to a 24-in. exhaust fan, was originally designed a series machine, but was changed into a shunt machine by placing the series field in shunt and inserting a resistance coil in series with the field windings.

The motor ran this way for a number of years, until it suddenly refused to run at all and upon investigating, the windings of both field and armature were found to be completely burned out. The machine was taken down and completely overhauled and after testing and finding everything in good condition it was connected to the fan again, all connections being made as before. It ran all right except that the fields showed a disposition to heat.



FIGS. 1 AND 2. MOTOR WITH COIL IN CIRCUIT AND WITH LAMP IN CIRCUIT

Thinking to lessen the field current, a 50-watt lamp was inserted in place of the coil, as in Fig. 2. Upon applying the current, however, the motor was found to rotate in the opposite direction and at a higher speed than before. No other connections were changed whatever; except those necessary to put the lamp in place of the resistance coil. The coil was replaced, and the machine was reversed again, assuming its original direction of rotation.

Later we took out the coil and connected it up as a regular series motor. It has now been running about a week and shows no signs of trouble; but what changed the direction of rotation?

If any engineers have had a similar experience, or can advance some logical reason for these peculiar actions, I should be glad to hear from them.

F. P. BOONE.

Des Moines, Iowa.

Running Three Phase Motors on Single Phase

Some time ago I was employed in remodeling the motive-power department of a large manufacturing plant which had been closed down for several months. It was desired to run several small motors about the plant for doing repairs and construction work, such as running lathes, drills, pipe machines, pumps and hoists for building material, but as the motors were all three-phase machines and the only current obtainable was single-phase of the same frequency it appeared to be a case for single-phase motors.

The three-phase machines would give about one-half of their rated capacity when running on single-phase current, but would not start themselves even under no load. The problem was then to find the easiest way of bringing them up to full speed, after which the load could be applied. A spare motor of the variable-speed type was set up and connected to its controllers in the regular way for three-phase current.

All the other motors to be used were hooked up for three-phase connections and three wires were run to all the motor switches and controllers. The two wires of the single-phase system were then connected to the two outside wires of the three-phase connections and a fuse was placed in the middle wire of the spare motor having twice the capacity of those in the two outside wires.

The spare motor was a low-speed type (600 r.p.m.), and had a large pulley on its shaft. When power was required for the other motors one man would revolve the shaft of the motor as fast as possible by pulling on the rim of the pulley, while another man would gradually cut out resistance in the controller as the motor came up to full speed.

This motor would then be left running idle and would generate three-phase current in its stator winding that would bring the other motors up to speed even when they were running with all the belts in place and partly loaded. This was found to be very convenient and this practice was continued for several months, until the demand for power became great enough to warrant the running of a three-phase alternator.

There were also a number of small motors in this plant that had absorbed considerable moisture. To dry them out, a small alternating-current motor was belted to a direct-current generator of low voltage and the direct-current mains were connected to one phase of the three-phase motors. The direct-current voltage was then adjusted to a point where full-load current was flowing through the windings of the motors. This was maintained until the insulation showed sufficient resistance; then the windings were given a heavy coating of insulating varnish.

A. J. DESAND.

St. Thomas, Ont., Canada.

Commutator Lubrication

Much has been written by POWER readers about commutator lubrication. However, I have had the care of dynamos and motors for the past seven years, and have never used a lubricant of any kind. I keep the brushes and commutators clean, free from copper and carbon dust, and the brushes well fitted to the commutators, and have found the lubricant that is put in the brushes during manufacture to be sufficient.

J. P. LAUTHERS.

Fort Miller, P. I.

Oil Switch Refuse to Release

The 3000-kw., 11,000-volt generator was protected by an automatic remote-controlled oil switch having a time-limit relay set for 2 sec. This refused to trip out one day, on a short-circuit, and the efforts of the switchboard operator to trip it from the board proved futile. He, therefore, rushed downstairs to the switch compartment and threw the switch out by hand. By this time the sealing water in the turbine gave out, and admitted air, which broke the vacuum. A spare switch was put into service, the vacuum restored and power was on the line in 5 min.

When the defective switch was taken apart it was found that the copper from the upper contacts had melted, and adhered to the bottom contacts, so that the tripping magnet was not strong enough to pull the contacts apart.

THOMAS SHEEHAN.

North Adams, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

The Junkers Engine—II

By F. E. Junge

A 1000-hp. Junkers Oil Engine

It is well known that the design of a small engine, however good, cannot always be applied to large units. The enlargement of dimensions, while retaining the constructive arrangements of the smaller type, may not only lead to difficulties in production and intolerable conditions in service, but may change economic conditions. Hence, a thorough investigation of the various relationships was necessary, and on these the preliminary experimental studies of Professor Junkers were concentrated.

But even the deductions from these researches could not alone determine the value and utility of the prospective engine. A single factor, such as reliability of lubrication or ignition, cannot be balanced by other features, however excellent. Hence, the value of an engine can be judged only upon the machine as a whole. Especially is this true with large engines operating on the Diesel principle, due mainly to the very high temperatures and pressures employed.

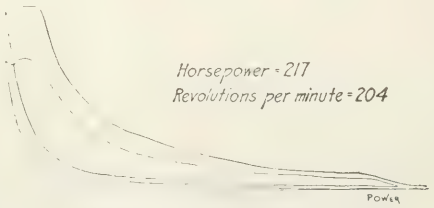


Fig. 7. Indicator Diagram

The first experimental engine built by Professor Junkers was designed for high pressures (100 atmospheres) and permitted an increased output of 50 per cent. above the normal. (200 hp.) The tests on this engine embraced, among other things, the efficiency of scavenging, the consumption of scavenging air, the completeness of combustion with various

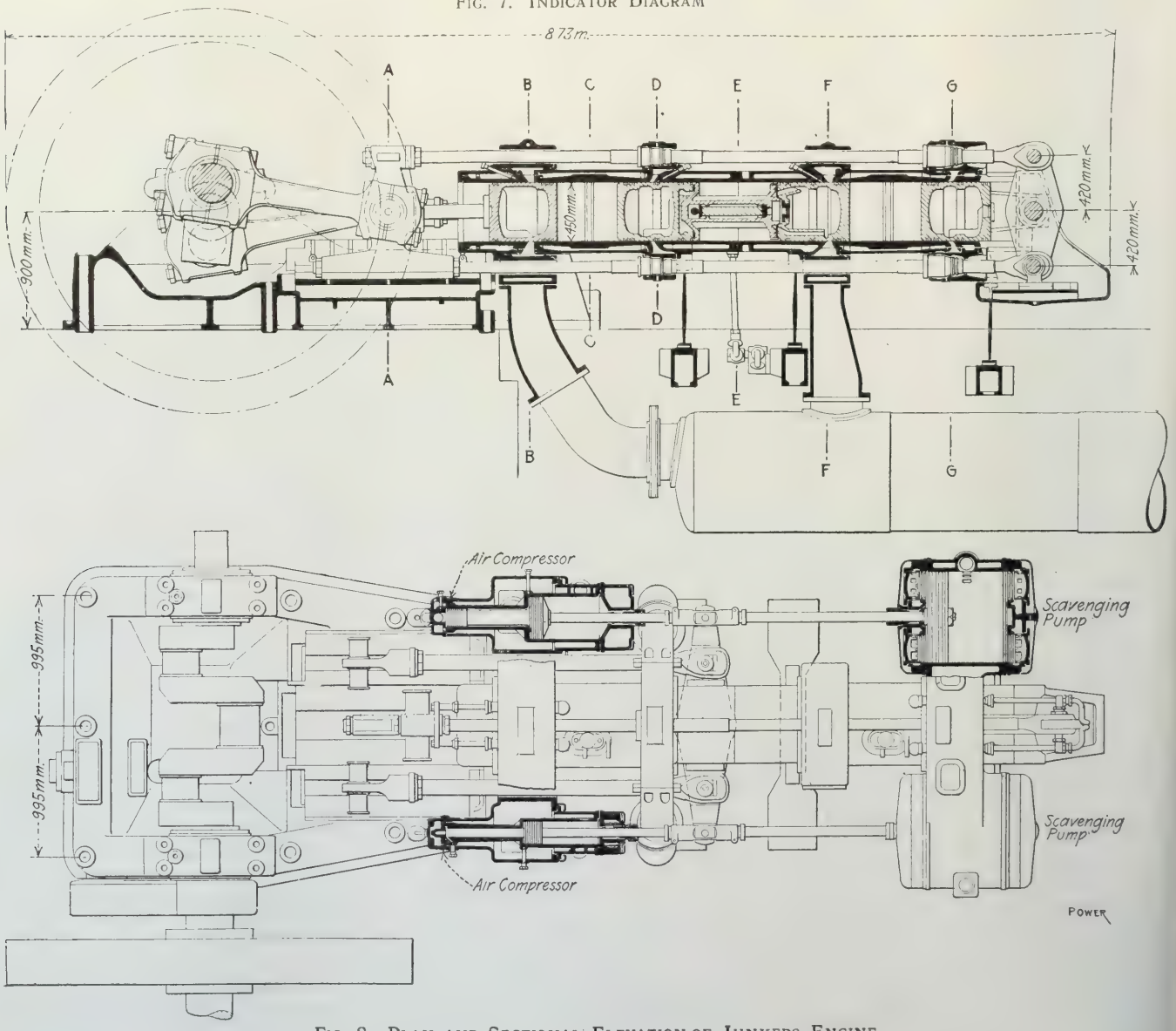


Fig. 8. Plan and Sectional Elevation of Junkers Engine

forms of injecting nozzles and different modes of fuel influx. Different kinds of fuels, such as various residues, mazout, paccura, and coal-tar oils, being used.

inder, and the four-stage air compressor is arranged on both sides of the front cylinder, two stages on each side. The scavenging pumps and the air compres-

inder has two fuel valves and one starting valve. The driving shaft is underneath the engine and drives two short camshafts carried by rocking frames.

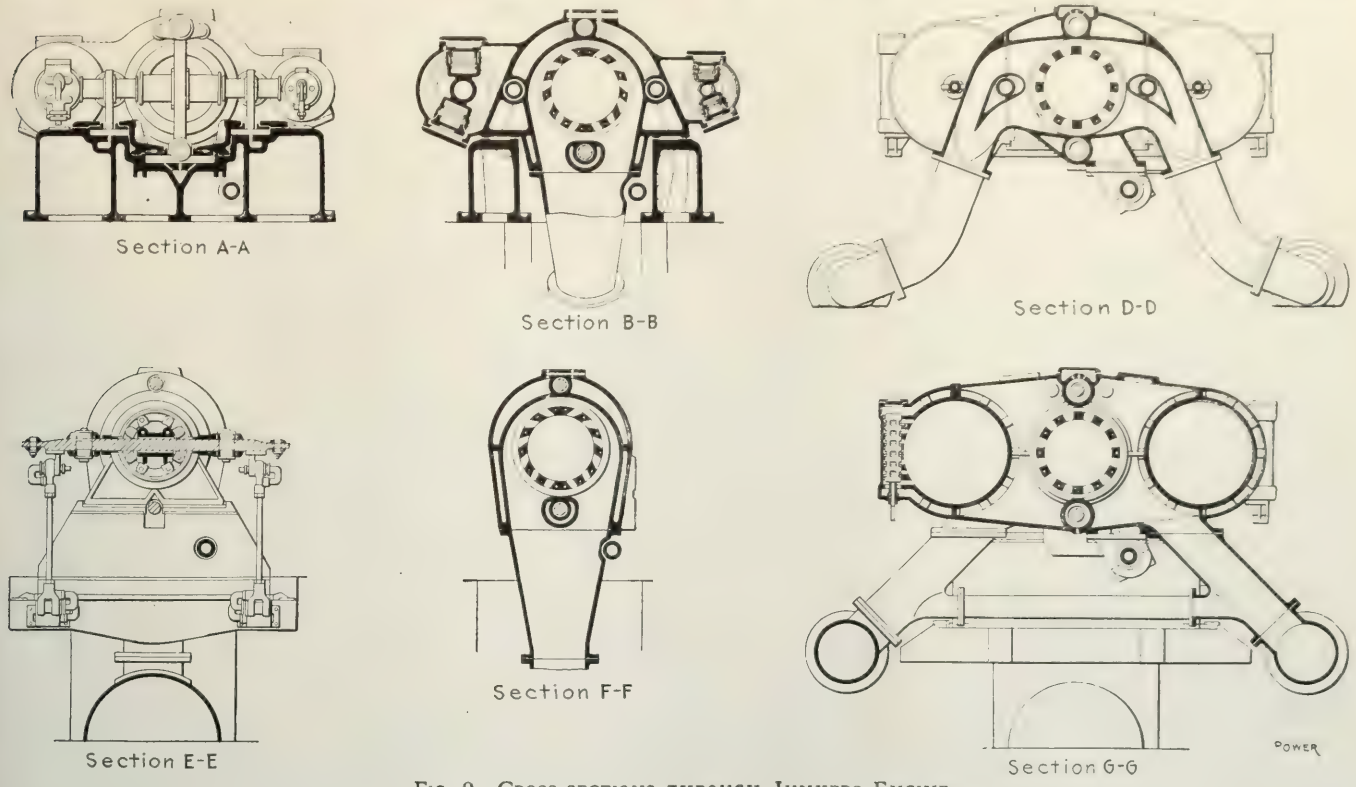


FIG. 9. CROSS-SECTIONS THROUGH JUNKERS ENGINE

Under normal running conditions at approximately 200 r.p.m. a fuel consumption of less than 0.4 lb. (200 grams) per effective horsepower was attained. The high mean effective pressure obtained from the indicator diagram was about 140 lb. per sq.in. under normal conditions, and 200 to 214 lb. per sq.in. with an overload; moreover the invisible exhaust showed efficient scavenging and complete combustion. Fig. 7 shows an average indicator diagram obtained on a 3-hr. test with 500 per cent. overload.

An extension and continuation of the tests undertaken on the 200-hp engine was carried out on an engine of 17.7 in. (450 mm.) bore and a 17.7-in. stroke of each piston yielding 1000 hp. The engine is built in tandem like the smaller one and is capable of 50 per cent. overload. It is also provided with a reversing gear. Moreover, to reduce the total length of engine the distance piece formerly employed was dispensed with, the bridge piece as well as the rear traverse piece being guided in slots at the extremities of the cylinder, a ratio of total stroke to diameter of cylinder of 2:1 was determined upon. Figs. 8, 9 and 10 show the plan, longitudinal, and cross-sections of the 1000-hp engine. The connection of the piston rods and the side rods with the cross-pieces is accomplished by means of moveable joints in order to avoid unequal stresses. The scavenging pumps are at both sides of the rear cylsors are actuated by the center traverse piece. This mode of operation is simple, reliable, and is accomplished with small frictional work.

The scavenging air is lead to the cylinders from both sides with as little ob-

for each cylinder, one being used for reversing. By rotating the rocking supports about the center line of the driving shaft from the controlling platform the ahead and reversing cams may be brought into contract with the valve-lever roller. Fig.

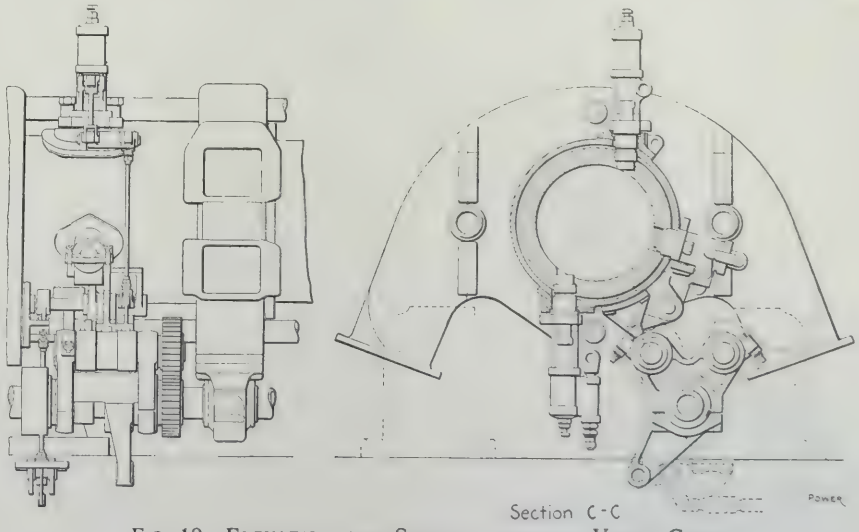


FIG. 10. ELEVATION ANE SECTION THROUGH VALVE GEAR

struction as possible, see sections *DD* and *GG*. The exhaust ports are so arranged as to allow a uniform exit of the escaping gases along the whole of the circumference of the cylinder as shown in sections *BB* and *FF*. The valve gear of the engine is actuated by cams as shown in Figs. 10, 11 and 12. Each cyl-

11 shows the valves closed in the ahead position and open in the reversing position. On each of the camshafts a starting cam is situated between two injection cams. The lower fuel valve of each cylinder is actuated directly by a lever and the upper valve by a system of links. Fig. 12 shows the action of the starting

valve, both when the engine is running ahead and when reversing. The three rollers for each cylinder are placed in axial alinement with each other, the interposed lever for actuating the starting valve being mounted on an eccentric journal, so that it can be put out of action by the levers leading to the controlling platform. For reversing it is only necessary to swing the rocking frame over and keep the starting valves in action till the engine rotates in the reverse direction.

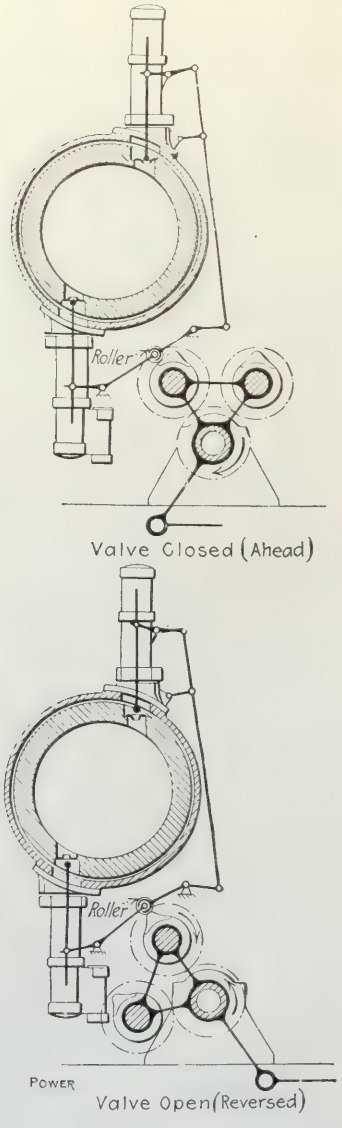


FIG. 11. FUEL VALVE MECHANISM

The advantages claimed for a large oil engine of this design are as follows:

1. The engine realizes the two-stroke-cycle principle with its admitted economic advantages and solves the important and difficult problem of providing good scavenging by the aid of the pistons.

2. The use of two pistons and the illustrated solution of the mechanism connecting them with the crankshaft permits balancing the reciprocating parts, taking up the forces exerted by the pistons in the mechanism itself, instead of the frame absorbing the unbalanced forces. Thus the main engine parts experience only such stresses as are created by useful external forces, and are free from uncontrollable heat and internal-strains.

3. The subdivision of the aggregate stroke by using two pistons permits a high ratio of stroke to diameter of cylinder, simultaneous with a high rotative speed, thus insuring favorable economic conditions, which are increased by the form of combustion chamber.

4. Dispensing with the cylinder cover and the scavenging valves reduces large-

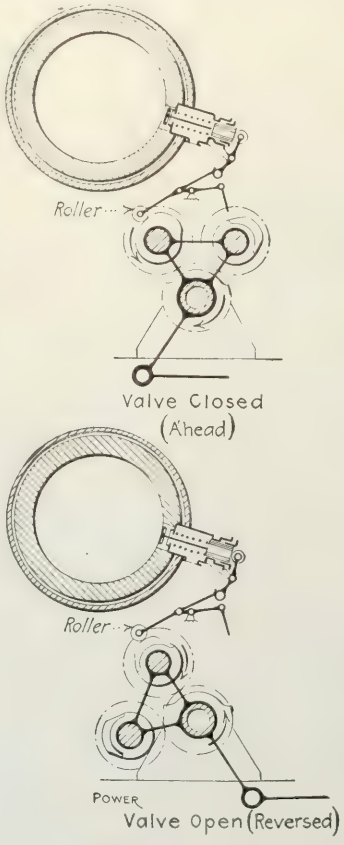


FIG. 12. STARTING VALVE MECHANISM

ly those difficulties which are caused by excessive stresses in the material, owing to wide temperature ranges in the cylinder wall.

Tar Oils in Diesel Engines

A recent test on a 480-hp. Diesel engine in Maiderich, Germany, showed that crude heavy tar oil can be successfully used as a substitute for the lighter oils in Diesel engines. It is known that when hard coal tar is distilled there are obtained, besides benzol and naphthaline, both of which are now used in internal-combustion engines, a number of light, medium and heavy tar oils, the last remaining products of the series being anthracen oil and pitch. Likewise, if brown-coal tar is distilled a series of oils is obtained beginning with benzene and followed by solar oil, light paraffin oil, dark paraffin oil, gas oil and creosote oil.

Formerly only the lighter grades of the oil had been successfully used in Diesel engines, although attempts were made to use raw tar directly. This, however, is a rather crude method because all the valuable byproducts of tar are lost by combustion. Now manufacturers have succeeded in using also the heavier grades of tar oil which are cheaper, costing 45 marks ($10.70) per ton, while the lighter grades cost 50 marks ($12), and gas oil, which is used for starting, 110 marks ($26) per ton, f.o.b.

The Diesel engine on which these trials were made is of the standard four-stroke-cycle type, having 4 cylinders. The compression during ignition is 33 atmospheres and the compression of the air for fuel injection 60 atmospheres (485 and 882 lb. respectively). The shaft which carries the eccentrics for operating the fuel valves, rotates at half the speed of the main shaft. Either compressed air, if available, or carbon dioxide is used for starting. Before the engine is warmed up gas oil must be used as auxiliary fuel, afterward tar oil can be used exclusively, even when the engine has been shut down for 2 or 3 hr. After a continuous run of several weeks it was found that the presence of soot in the cylinder was due not to the fuel itself but to a more than liberal application of lubricating oil. The only trouble experienced was with the fuel pumps, which showed signs of wear due to the tarry constituents.

A remarkable feature ascertained by this test was that, although the heat value of the heavy tar oil is 400 cal. less than that of the light tar oil, not only was the average consumption at all loads lower, but the speed variations were considerably reduced, and the engine ran quieter.

Another interesting observation was that the fuel consumption at full load was about 3 per cent. higher than at ¾ load; this slight increase probably being due to the fact that for the comparatively low heat value of the heavy tar oil full load is synonymous with overload.

When light tar oil was used the fuel consumption at ¾ load was 320 gm. plus 27 gm. gas oil for starting, which, with prices as quoted, corresponds to a fuel cost of 1.9 pfennig or 0.45c. per kw.-hr. When crude heavy tar oil was used the fuel consumption at ¾ load was the same and the cost 1.72 pfennig or 0.40c. per kw.-hr. When no gas oil is added the fuel cost is equal to 1.04 pfennig or 0.25c. per b.hp.-hr. As the most favorable consumption is reached at 80 per cent. load, it is commendable to use light tar oil above this limit, if highest economic efficiency is to be secured. Computing the total operating cost for 300 days in the year, at 20 hr. service per day, and 80 per cent. average load, which include, beside fuel, charges for lubrication, water, attendance, upkeep and repair, the service of one effective horsepower-hour, at prices obtaining in Germany, costs 1.75 and 1.52 pfennig, or 0.42 and 0.36c., for light and heavy tar oil, respectively.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heat Transmission through Corrugated Iron

By A. H. Blackburn*

The modern tendency to construct shops and manufacturing establishments with large window areas, the concrete or steel frame often forming little more than a grid for the windows, has directed much attention to the calculation of the heating requirements of such buildings. The Green Fuel Economizer Co., of Matteawan, N. Y., recently had an opportunity to ascertain the transmission through a building of this kind, the structure being a shop for the manufacture of its fans and blowers, and the results will, no doubt, be of general interest.

The shop measures 225x48 ft., with an average height of 32 ft. There are continuous windows on one side 15 ft. high and on the opposite side 19 ft. high. Otherwise, the building is covered with corrugated iron of single thickness without lining. The crevices at the eaves are filled with asbestos and the corrugated iron is cemented in at the bottom, and other measures have been taken to make the building as air tight as possible. The exposure of the windows is east and west, the smaller windows being on the east side.

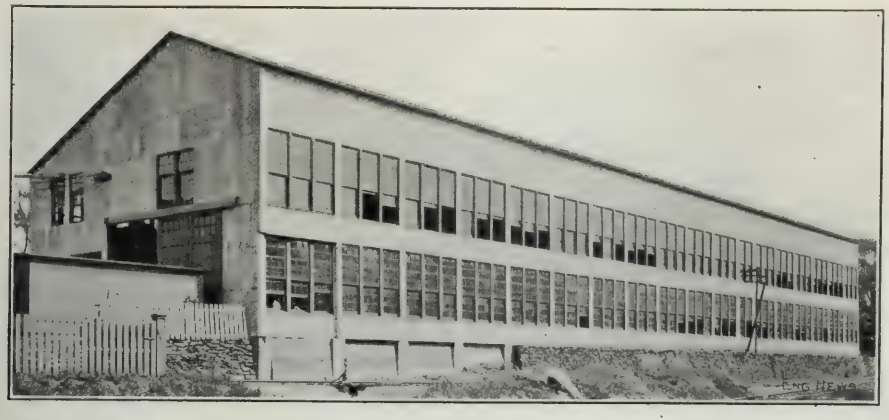


Fig. 1. Exterior of Fan Shop Covered with Corrugated Steel Sheathing

Coefficients given by different authors for the rate of heat transmission through single windows vary somewhat. Prof. Homer Woodbridge gives for single windows with a southern exposure, a coefficient of 1 B.t.u. per sq.ft. per hr. per deg. difference in temperature between inside and outside. For northern exposure he suggests adding 35 per cent., for eastern exposure 15 per cent., and for western exposure 25 per cent. Ludwig Dietz, quoting Dr. Rietschel, of Berlin, and also the specifications of the Prussian Government, give for single windows 1.026, and Dr. Rietschel suggests increasing this 10 per cent. for northern exposure or northeasterly or northwesterly exposures, and 10 per cent. for especially strong winds. The only figure for corrugated iron without sheathing that has been found is one by Rietschel of 2.132, but without any statement as to whether this refers to superficial wall area or to the actual surface of the iron.

Various attempts have been made to analyze the coefficient of heat transmission from one fluid to a solid substance and from the solid to a second fluid. The formula commonly employed is of the form,

$$k = \frac{1}{\frac{1}{A_i} + \frac{1}{A_o} + \frac{x}{e}}$$

where

A_i = Transmission from the inside fluid to the wall surface;

A_o = Coefficient of heat transmission from the outside wall surface to the second fluid;

e = Conductivity of the material in heat units per square foot per hour per degree difference per inch of thickness;

x = Thickness in inches.

For the value of A_i and A_o Grashof gives the following equation:

$$A = c + d + \frac{(40c + 30d)}{10{,}000} T$$

in which c depends upon the velocity of the fluid. Rietschel gives for air at rest $c = 0.82$; for slow motion, as in contact with cold windows, $c = 1.03$; for rapid motion, as outside of buildings, $c = 1.23$. For glass d is given as 0.60 and for sheet iron as 0.57. For single windows T is given as 36, no value being given for iron.

Calculating from the above values, it is found that for a single-glass window $A_i = 1.84$ and $A_o = 2.07$. For sheet iron $A_i = 1.81$ and $A_o = 2.04$. For e Rietschel gives 190 for iron and 6.6 for glass. It is evident that the term $\frac{x}{e}$ in the denominator of the expression for k can be neglected for iron and is practically negligible for glass.

Using the values of A_i and A_o just found, k for a single window works out as 0.958, and for a single thickness of sheet metal as 0.955, or practically the same. This is not taking account of the additional surface of the sheet metal wall due to the corrugations, which in the building under discussion number one to every 2½ in., the depth of the corrugation being 1 in. The actual surface of the metal is approximately 1.35 times the superficial area. This would make the rate of heat transmission for corrugated sheet iron, figured on the basis of superficial area, approximately

$$1.35 \times 0.955 = 1.29.$$

That the values obtained by Grashof's method for both glass and sheet iron are lower than the values used in ordinary practice may possibly be explained on the hypothesis that the constants used in ordinary practice have been increased to account for losses due to leakage, the opening and shutting of doors, etc. Weight is lent to this view by the results of the tests which follow, bearing in mind that the construction of the present building makes it practically air-tight. At the same time the values obtained were slightly greater than Grashof's formula would call for. Any close agreement is not to be expected, as the sheet-metal-working machinery and shafting housed in the building were in constant operation during the test, producing heat which was not measured, while on the other hand, doors were being opened for the passage of men and materials on an average of once every 10 min.

Of the total surface of the building approximately 7538 sq.ft. is windows, in-

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cluding the sash, 8247 sq.ft. is wall surface and 11,925 sq.ft. is roof; the walls and the roof are given in superficial area. The total of wall and roof areas is 20,172, which, increased by the factor 1.35 to account for the corrugations, gives 27,130 sq.ft. total metal surface. The total superficial expanse of the building is 27,710 sq.ft., and the total actual surface, counting the metal in the corrugations, is 34,658 sq.ft.

To heat the building an engine-driven centrifugal fan draws air through a hot-blast heater having six 7x8-ft. sections of four rows of pipe each. There are altogether 6816 lin.ft. of 1-in. pipe in the heater, or 2272 sq.ft. of heating surface. The heater is ordinarily drained of condensate and air by a Dexter vacuum system. Circular sheet-iron conduits with

methods were employed. First, it was calculated from the amount and rise in the temperature of the air passed through the heater, and second, from the condensation in the heater. For the latter purpose, the steam pipe from which the steam was introduced to the heater was carefully drained and the condensate from the heater was weighed. Figured by the air method, the consumption of heat was 2,084,000 B.t.u. per hr., and by the condensate method, 2,029,730 B.t.u. per hr. Using the air figures, the average rate of heat transmission through the superficial area of the building, that is not counting the corrugations, was 1.42 B.t.u. per sq.ft. per hr. per deg. difference, and using the steam value, 1.38 B.t.u.

Taking Professor Woodbridge's figures

such difference between the iron and the glass. Allowing for the many undeterminable conditions, it would probably be safe in calculating the heat supply for buildings of this kind to provide for a coefficient of transmission of 2 B.t.u. for the whole wall and roof area.

The coefficient of transmission through the surface of the hot-blast heater in this test was 9.41 B.t.u. per sq.ft. per hr. per deg. average difference in temperature between the air and the steam, figured by the ordinary arithmetical method, in which the average difference in temperature between the steam and the air is assumed to be the difference between the steam temperature and the mean of the air temperature on entering and leaving. According to the logarithmic formula, which takes account of the fact that the

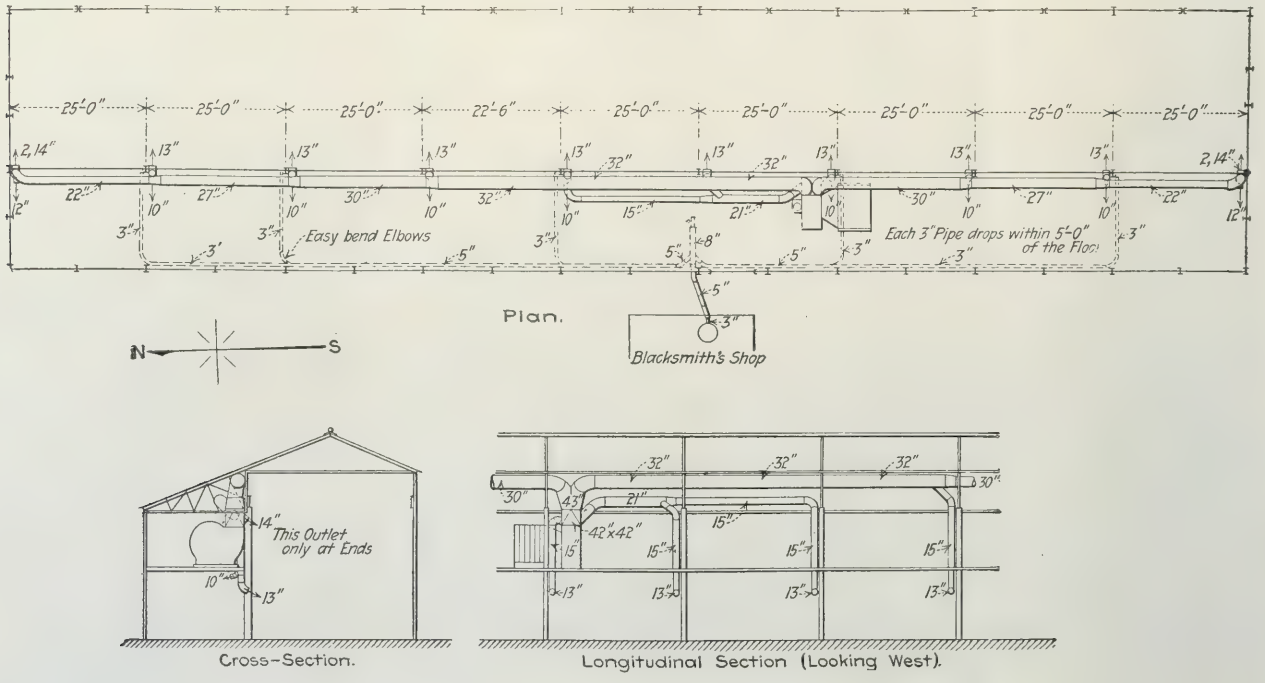


FIG. 2. PLAN AND SECTIONS OF BUILDING ON WHICH TESTS WERE MADE

outlets directed down into the zone occupied by the workmen distribute the air throughout the shop.

The fan and heater had been running for about three hours when the first readings were taken at 8:30 a.m., and the temperature of the shop was maintained constant until readings were discontinued at 11:30 a.m. During the test the fan was run at 258 r.p.m., receiving 22,416 cu.ft. of air per minute, figured to 50 deg. F., the air actually being received by the heater at 73 deg. F. and delivered from the fan at 156 deg. F. The temperature of the steam in the heater was 212 deg. F., and of the air delivered from the farthest outlet 141½ deg. F. Under these conditions, the temperature of the building, measured 3 ft. from the floor, was 66 deg. and measured in the gallery, 70 deg., the temperature out of doors being 15 deg. F.

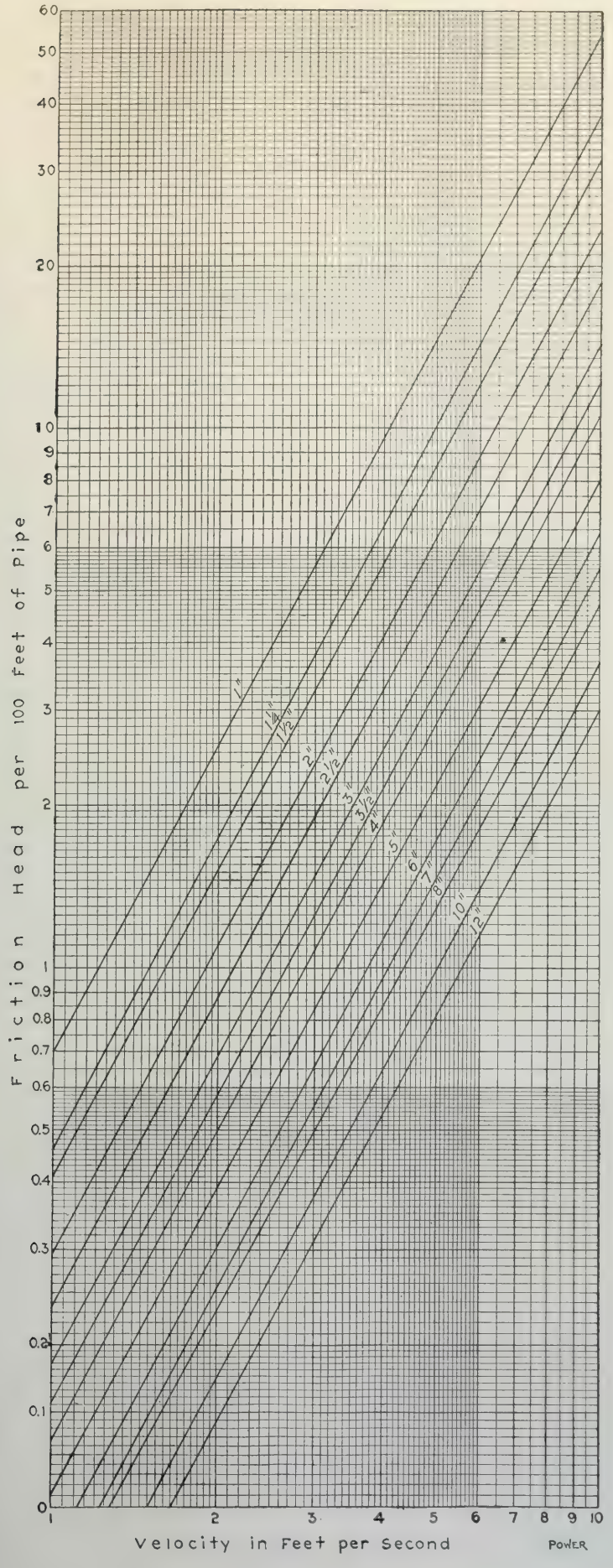
For determining the total heat, two

for the rate of heat transmission through the windows gives for a window on southern exposure, 1 B.t.u. per sq.ft. per hr. per deg. difference, which, however, is to be increased 15 per cent. for eastern exposure and 25 per cent for western exposure, or in the present case, an average of approximately 20 per cent. Figuring the window surface as 1.2, the value for the corrugated iron, figuring superficial area only, is 1.5, or figuring the whole surface of the iron, 1.13. This is less than an equivalent amount of glass surface and seems hardly likely, although the rate of heat transmission per square foot of actual surface through corrugated iron may be less than through a flat iron surface.

By taking 0.958, obtained by Grashof's method for glass windows, the value for the actual iron surface comes out at 1.19, but on the basis of this formula, there would be no reason for expecting any

air rises more rapidly in temperature in passing over the first rows of tubes than in passing over the later rows of tubes, the value is 10. The mean physical velocity of the air through the heater was 1210 ft. per min.

In a subsequent test, with air entering at 72 deg. F. and leaving at 166 deg. F., and steam at 213 deg. F., in which the velocity of the air was 760 ft. per min., a coefficient of transmission of 7.5 (logarithmic) was obtained; 10 and 7.5 are in the ratio of the 0.64 powers of the respective velocities; that is, the coefficient did not vary directly as the velocity, as frequently asserted. If, instead of mean velocity, flow in pounds of air per minute through the heater is used, which amounts to the same thing as velocities reduced to a standard temperature, the coefficients are found to be in the ratio of the 0.62 powers of the rates of flow.



FRICTION HEAD IN PIPES 1 TO 12 IN. IN DIAMETER

Friction Loss in Wrought-Iron Pipe

BY W. L. DURAND

In the July 9 issue appeared an article by Ira N. Evans entitled "Friction Formulas for Commercial Pipe." In it he has taken six of the best formulas for the friction of water in commercial wrought-iron pipe and the results of several tests and compared them in tabular form and also in a series of curves. The values are given in friction head per 100 ft. of pipe and include velocities up to 15 ft. per sec. and commercial sizes of pipe from 1 to 12 in.

Due to the widely varying tables that appear in different handbooks and the necessity of elaborate interpolation to make them usable, a great deal of time is generally wasted in solving problems in which friction of water in pipes is a factor. From a close examination of the table given by Mr. Evans, evidently using the average value of all the formulas will give results as close as could be desired in hydraulic work.

These values seemed to be so much superior to any previously published that I plotted them on logarithmic paper and have used the chart with a great saving of time.

In plotting on logarithmic cross-section paper, in which the rulings are made at distances proportional to the logarithm of the number, all equations of the general form $y = B\,x^n$ will plot as straight lines. Several of the formulas are of this form, and those that are not vary from a straight line to only a slight extent. The average values then may safely be represented by a straight line, as the greatest variation will be much less than the difference between any two values that make up the average.

The method of using these curves is self-evident, there being three variables and in any problem containing two of them the third can be found immediately from the chart. If only one variable is known several values of one of the other two may be assumed and the corresponding values of the third found from the chart. The values that seem to best fit the problem may then be chosen.

The chart is plotted with velocities from 1 to 10 ft. per sec. as abscissas and friction head per 100 ft. from 0.1 ft. to 100 ft. as ordinates. Each curve is for a definite pipe size as indicated, and with a little variation all the way through for velocities of 1 ft. per sec., the chart shows values practically identical with the averages in the table.

For example, take a 3-in. pipe with a velocity of flow of 6 ft. per sec., the friction head per 100 ft. of pipe would be 5.45 ft. To produce a drop in head of 2 ft. in 100 ft. of 5-in. pipe would require a velocity of 4.7 ft. per sec.

New York Chapter Opens Season

On Oct. 14, in the Engineering Societies Building, the New York Chapter of the American Society of Heating and Ventilating Engineers held its first meeting of the season.

The topic up for discussion was the relative efficiency of wrought-iron pipe coils and cast-iron surface used in connection with the hot-blast system. There appears to be very little reliable data on this subject and a great difference of opinion among various authorities. On minor details these differences were reflected at the meeting, but all agreed th the difference in efficiency between wrought-iron coils and cast-iron heaters, all prime surface, was very slight. The efficiency of pipe coils depends largely on the number of pipes deep, the velocity of the air through the surface and the extraction of air from the interior of the pipes. The resistance to air flow was thought to be slightly less in the cast-iron heater, and it was suggested that the wrought-iron showed a tendency to scale up and become less efficient. The cost of connecting up favored the coils.

Cast iron with extended surface showed very poor results at velocities around 1000 ft. per min., but with velocities as low as 400 ft. per min., the radiation obtained per square foot of measured surface more nearly approached that from all prime surface.

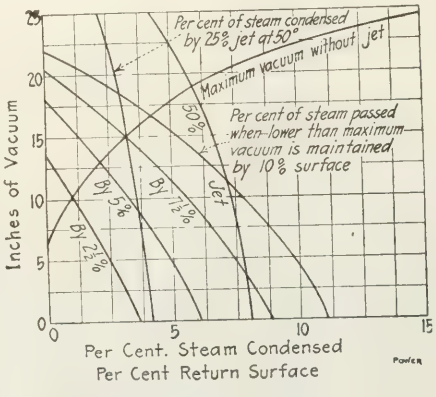
With an air flow through the heater of approximately 1000 ft. per min. it was common to obtain 800 or 900 B.t.u. per square foot per hour from either kind of surface, but a case was cited where the above figures for radiation had been doubled by running up the velocity to 1800 ft. per min.

A report of progress from the committee on moving-picture shows and other routine business was followed by the report of the tellers of election. The officers for the ensuing year are: President, F. G. McCann; vice-president, D. D. Kimball; secretary, Joseph Graham; board of governors, W. M. Mackay, Conway Kiewitz and W. W. Macon.

Vacuum Heating Chart

In the course of developing a standard method of testing radiator steam traps for return-line vacuum-heating systems, James A. Donnelly, a member of the committee on tests of the American Society of Heating and Ventilating Engineers, devised an interesting chart showing the vacuum which may be maintained with varying percentages of return surface (radiating effect), the percentage of steam passed and condensed in the return when a pressure higher than the maximum vacuum is maintained, and the percentage of steam condensed when jet water is used. The chart is based upon the assumption that the water of condensation is about 10 deg. cooler than the steam from which it is condensed in cast-iron radiators.

A glance at the chart will show that with return surface amounting to 10 per cent. of the radiating surface, it is possible to carry a vacuum of 22 in. With a vacuum of 10 in., 7½ per cent. of the steam would be passed. With 5 per cent. return surface a maximum vacuum of 18 in. can be carried and with 10 in. of vacuum about 3 per cent. of the steam would be passed.



VACUUM HEATING CHART

Using 25 per cent. of jet water at 50 deg. F., the percentage of steam condensed varies from 2 to 4 per cent., with the vacuum carried. With 50 per cent. jet water the percentage of condensation lies between 4 and 8 per cent.

The curve running from the left of the diagram to the upper right-hand corner shows the maximum vacuum which can be carried without jet water, with various percentages of return surface.

CORRESPONDENCE

Returning Condensation to Boiler

Recently I was called upon to lay out the heating work for a building which, in addition to a low-pressure heating system, also contained a clothes dryer and a hot-water tank. The clothes dryer is in the basement, which is 8 ft. high and the boiler and hot-water tank in the boiler room, the floor of which is 2 ft. below the basement floor.

The clothes dryer is heated by a coil, containing about 800 ft. of 1-in. pipe, located 12 in. above the floor. As the water line of the boiler is 6 ft. 9 in. above the boiler-room floor, the coil is 3 ft. 9 in. below the water line, so that the water of condensation cannot return directly to the boiler. The coil in the hot-water tank is 12 in. above the boiler water line, but this is not enough to return the condensed steam in the coil. To operate the dryer and hot-water tank requires mechanical means to return the condensed steam to the boiler.

The condensed steam from the dryer and hot-water tank is carried to a receiving tank of about 50 gal. capacity. This tank contains a 9-in. copper float, connected to a tank switch arranged so as to operate on about 6 in. difference of water level. The tank is placed so that the high-water line is 18 in. below the dryer coil. The receiving tank connects to a small electrically driven centrifugal pump controlled by the tank switch so that when the water reaches the high-water line the pump is thrown on and continues running until the water line has dropped 6 in. This pump discharges into the return line from the low-pressure heating system near the boiler, and a check valve is placed on the house side to prevent the pump discharging into the heating system.

The same result could be obtained by using a steam pump in place of the electric pump. To operate the pump, however, would require carrying 15 to 20 lb. pressure on the boiler and using a reducing valve on the line running to the heating system. The exhaust steam from the pump could be used in the heating system by first passing it through an oil separator and installing a back-pressure valve and exhaust line to the roof. This arrangement would cost practically the same to install and would probably be less expensive to operate.

Another arrangement which would be more expensive to install, but would not cost much to operate, would be to carry about 10 lb. on the boiler and use a reducing valve as above. The water of condensation could be brought to a lifting trap placed on the boiler-room floor which would lift into a return trap placed above the boiler. For successful operation, however, the return trap must be at least 3 ft. above the boiler water line. This arrangement could not be used in low boiler rooms unless some place on the floor above could be found for the trap.

With the latter two methods, due to the pressure in the boiler being higher than the heating system, all the water of condensation would have to be returned by mechanical means, instead of only that from the hot-water tank and clothes dryer.

W. L. DURAND.

Brooklyn, N. Y.

Occasionally, pulpit and press give gratifying recognition to the men who keep the industrial wheels moving without making any noise about it. Over 3000 men of nearly every trade and vocation heard a clergyman in St. Patrick's Cathedral, New York, a recent Sunday evening lecture on the rights and duties of labor.

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FRED R. LOW, Editor.

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POWER

Load Dispatching

In some form or other the load-dispatching function exists in practically every successful electrical distribution system, and as the values of the connected load and the investment increase, the need of centralized control of operation becomes more imperative. Only within the past few years has the work of the load dispatcher been organized on large systems upon a thoroughly scientific basis, and still many central-station and power-service men little realize its scope and value. On the great Western hydro-electric transmission systems and to some extent in the lighting, power and railway properties of the East one finds this specialized work most highly developed. It reaches its greatest value on systems supplied with electrical energy from interconnected plants and whether these installations are steam or hydraulic, a first-class load-dispatching organization becomes one of the most important factors in economical operation.

The dispatching office must give twenty-four-hour service and be fully equipped with telephones, recording and indicating instruments, schematic diagrams or working dummies of all high-tension lines and switches, together with log sheets. The practice of receiving readings of load by telephone or telegraph at short intervals from all important electrical centers on the system, facilitates the handling of generating units, tie lines and transmission lines. While entailing considerable work, it is essential in the economical use of water and fuel, the disposition of station and substation labor and the transfer of loads to permit inspection and repairs.

In the actual work of dispatching loads the frequency meter is important. A drop of even two or three cycles is a hint that excessive loads, governor difficulties or other troubles are being encountered. Prompt work with the telephone, the throwing in of tie lines and possibly the starting up of auxiliary steam plants may be required, and the dispatching force must have good judgment to eliminate needless expense. If the load can be carried in any other way than by bringing cold boilers up to full steam pressure, economy will be served. Where auxiliary steam-plant capacity is likely to be needed it must be anticipated considerably in advance of the load unless oil fuel is employed.

The safeguarding of lines cut out for repairs, the transmission of time signals and general oversight of operation afforded by a good dispatching organization are worth far more than their cost, and we look to see this phase of system administration greatly extended in the future handling of associated generating plants and distributing networks.

Flyball Automatic Cutoff Governors

Generally there is no good excuse for a governor belt becoming oil soaked, even though it runs direct on the crankshaft. A little time spent in making wipers or a baffle in the form of a disk to be fastened to the crankshaft between the belt and the bearing from which the oil comes would prevent this evil and save the engineer much trouble.

A governor belt should be as free from faults as possible. Not only does the good regulation of the engine depend upon it, but only too often the safety of the engine, its operator and others in the immediate vicinity as well as the buildings depend upon the belt performing its functions properly. An oil-soaked belt that is never cleaned will soon become nearly as great a menace as a weak and overloaded boiler. Of course this is not so if the safety device of the governor is in working order, but in plants where the governor belt through neglect, is rotten, the safety device is usually inoperative through the same cause.

When it is considered that through the exigences of the service, an engineer will speed up an engine beyond the safe rim velocity of the flywheel and at the same time neglect the belt and safety device, the seriousness of the situation

becomes alarming. If the load on the engine is steady and the belt not too oily, trouble is experienced only on starting the engine. In such cases the speed of the governor lags behind that of the engine, due to slipping of the belt, and it is very probable that the safe rim velocity of the flywheel will have been exceeded before the governor speeds up sufficiently to produce an earlier cutoff and reduce the speed of the engine.

Conditions of this kind exist because of the ease with which they are overcome at the moment trouble is likely to occur. By pushing back on the governor reach-rod the engineer secures an earlier cut-off until the governor has come up to speed, when it will take care of itself. A condition so easily surmounted is soon forgotten, until eventually the time comes when the reach-rod is not attended to soon enough and the engine is wrecked and perhaps human life sacrificed to criminal negligence.

Another highly dangerous practice among some engineers is to leave the pins in the governor column after the engine is started. Some do it because they "forget," others, deliberately because "it saves time."

In plants subjected to heavy overloads which come on suddenly, the speed of the engine, and consequently the speed of the governor, is almost instantly reduced unless a very heavy flywheel is used. The governor, therefore, will drop its spindle so far that the safety cam of the releasing mechanism will be forced over against the hook, preventing the catch from engaging the block. This action closes the steam valves just when a large amount of steam is needed to carry the momentary overload. To avoid the closing of the valves at such times some engineers leave the pin in the governor column or block the over-balanced "dog" that serves the same purpose as the pin.

Blocking the spindle will insure steam when the governor speed is suddenly reduced, but it also prevents the governor automatically checking excessive engine speeds if the governor belt should break. The risks taken therefore where such measures are resorted to to avoid interruptions of service are very great, but it is dangerous practice and should obtain only in extreme cases. Human life and property are worth more than the loss entailed by service interruptions.

Safety First

Greater safety in power-plant operation being a reform we are continually advocating, we would like to add our indorsement to that of other publications, notably the *Railway Age Gazette*, of the "Safety First" movement. Important as are speed and comfort to the traveling public, safety is of much greater moment, and this is the basic principle upon which improvement is urged.

We are not prepared to offer any suggestions as to how safety may be increased in the railway field, but there are several ways in which it ought to be done in stationary power-plant practice, and we intend to keep our lungs and tongues tuned up on this score as long as we feel we can do any good.

Our idea is to hook up a little more than the railroads only to this "Safety First" movement, and we trust our readers will ally themselves with it at once and each do his part to try to have wrong conditions righted in his locality. From time to time, we shall call attention to these wrong conditions, and whenever our friends find them obtaining in their plants they should make it their business to call the attention of the management to them. Such an action is a duty each owes to himself for his own welfare as well as that of the general public.

Overloaded Boilers

Few things look more harmless than a boiler under pressure; yet few things are so capable of destruction.

With no reflection upon the design or workmanship of any boiler, it is as dangerous a contrivance as humans have to deal with. A designer or maker has meager influence regarding the treatment boilers shall receive after they have been turned over to the purchaser. The engineer also often has little to say about this. His business is to operate them under the most favorable conditions that he can produce—conditions which often conduce to danger rather than safety.

Inadequate boiler capacity is unquestionably common to over fifty per cent. of all plants, chiefly because the owner insists that the engineer "get along for the present." Under such conditions and with service that must not be interrupted, proper care of the boilers is well-nigh impossible. If the men succeed in carrying the peak load, regardless of how hard they force the boilers to do it so long as they are able to hold the water, they feel they are doing well, and they are.

The designers and the makers of boilers can incorporate in their product the best form of vessel, details of structure and metals that all the years of boiler practice have proved superior. To the maker is available the finest facilities and workmen to produce a boiler consistent with the best design. But these boilers, the best, cannot long perform their function if overloaded, forced to the limit and left to care for themselves until so hurriedly examined that serious defects are still undiscovered, when they are again put in service.

With water-tube boilers the usual effect of such operation is the splitting or "letting go" of a tube. If the fireman is unfortunate enough to be charging or cleaning a fire when a tube "lets go," the circumstances of which he is a victim are most vividly portrayed in last week's foreword picture. It does not end with the fireman, however, the whole plant suffers the shock; worst of all, life is often sacrificed to the demands of the service.

The remarkable part of it all is that the men so stoically, so fatalistically accept the horror of such accidents.

"I" vs. "We"

A man in charge of a plant, or any man having men under him, will secure better results if he does not try to be self-important, but uses a little tact. Men under the direction of another are quick to appreciate consideration at his hands even in small ways. For example, it is just as easy to say, "We must get this work done today," as to say "I want that work done today," and the chances are much more favorable that the work will be done. The use of "we" encourages co-operation, keeps the men interested by suggesting that they have a share in the responsibility and is gratifying to them because it does not remind them of their inferior position.

It's a pleasure to work under a boss who makes you feel that you are working *with* him rather than *under* him. He is the kind that encourages the men to make suggestions in regard to the work, realizing that every man under him knows something that he does not know.

When the boss shows interest in his men he will find they will show interest in him.

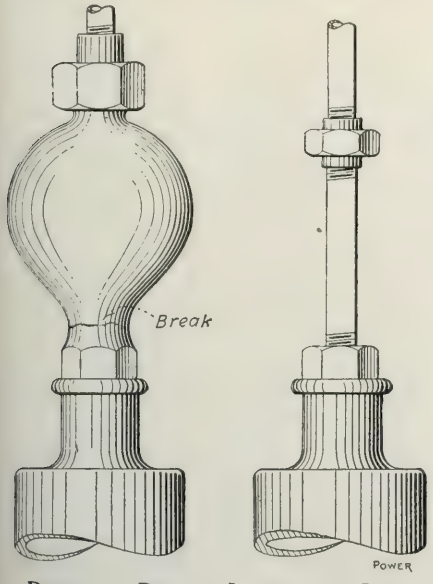
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Repairs to Lubricator Condenser

The following account of a repair job that I did while in charge of a small steam plant may be of interest to other engineers.

A governor spring on one of the engines broke, the weight hitting the dome or condenser of the lubricator, breaking



REPAIR TO BROKEN LUBRICATOR CONDENSER

it nearly off at the neck. It was too badly damaged to be soldered and we had no spare one, so I cut the dome off at the break, tapped it out with a ¼-in. pipe tap, reamed the lower end of the pipe out, so it would not interfere with the pipe that runs down into the body of the lubricator and put the pipe into the neck of the dome. I also put a union in the pipe, to permit removing the lubricator, as it was cold weather. When I put it on I filled the pipe with cold water, and the lubricator gave us no trouble. The illustration shows the break and the way it was repaired.

ALFORD H. SMITH.

Rogers, Ark.

Oil Deposit Causes Explosion in Air Line

We have two 22 and 14 by 14-in. air compressors, discharging into a 6-in. main under a pressure of 100 lb. The receiver is small and located about 200 ft. from the compressors. We get considerable moisture and oil from the receiver, and moisture is carried all through the lines to wherever air is used.

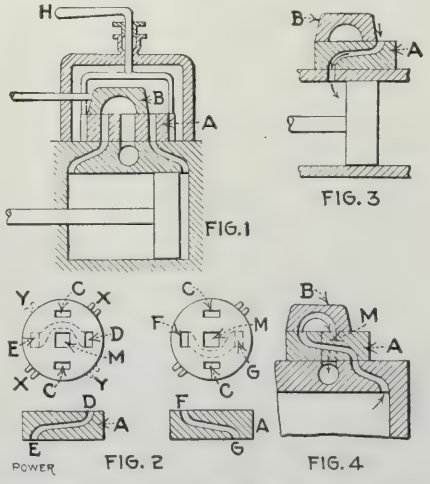
Recently an explosion occurred in the main line, breaking a 6-in. elbow near the receiver. We found the oil had formed a carbon deposit in the pipe which in some way created a gas that later exploded. Can anyone tell me what caused this, and what to do to prevent its reoccurrence?

H. H. DELBERT.

Titusville, Penn.

Peculiar Reversing Mechanism for Launch Engine

The reversing mechanism described below was found on an old government launch engine. The engine had only one eccentric and that was keyed and setscrewed to the shaft. The steam chest was round and very deep. Inside it was the casting *A* which had two sets of ports, the arrangement of which is shown in Fig. 2. To avoid confusion of too many lines, two drawings in Fig. 2 were made to show how the ports ran. The ports *C* went straight through as did also the



UNUSUAL ENGINE REVERSING MECHANISM

exhaust port in the center. The port *D*, Fig. 2, led through at the other side of the casting at *E*, and the port *F* led out at *G* in the same manner.

At *X* were two lugs which engaged the forked end of the handle *H*. This forked end was inside the steam chest and the stem had a stuffing-box and was packed where it passed through the steam-chest cover. With the lugs *X* in the position shown in Fig. 2 the ports *C* were parallel with the valve in its movement. With the valve *B* on the bottom, as in Fig. 1, steam would be admitted at the head end of the cylinder.

If the yoke lever *H* be thrown 90 deg. in the direction of the arrow, so that the lugs *X* occupy the positions shown by *Y*, the port *D* will be at the bottom and the port *F* at the top. With the valve in the same position, as shown in Fig. 1, steam will be admitted to the top of *A* and enter the open port, and the exhaust from the cylinder will follow a similar course and out through the square exhaust passage *M*. Fig. 3 shows the course of the steam and Fig. 4 the course of the exhaust.

L. A. SUVERKROP.

New York.

Engine Room Signal and Telephone System

Our plant furnishes power to old mills scattered over a wide area. All the mills contain old shafting, couplings, gear drives and aged belts which give much trouble and cause many shutdowns in the course of a year.

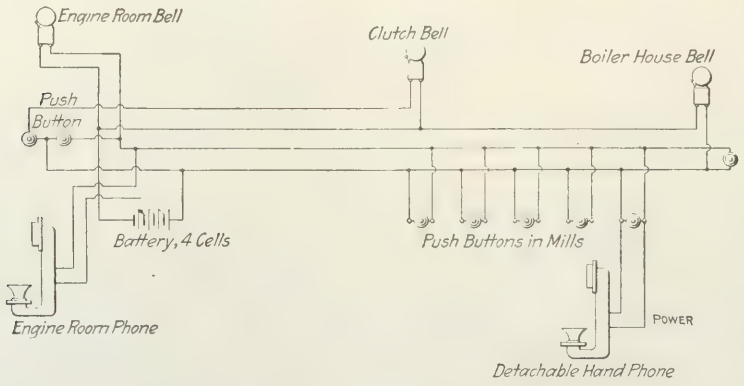
The management could not be induced to install an automatic engine-stop system, so I put in a signal-bell system as illustrated. We have a code of signals, known to the repair-shop help, overseers and foremen in the mills and to the engine-room force. This system worked fairly well, though sometimes wrong signals were given and the engine moved contrary to intentions, but it saved time and was better than useless running back and forth between the engine room and the seat of trouble.

We gradually extended the system to all the mills, using a damp-proof office wire for the circuits. Some time ago we had trouble with a jackshaft at the head of a rope drive which kept breaking repeatedly and soon found that the signal code was not sufficient to cover the amount of engine movements required when making repairs. Finally I was permitted to install hand telephone sets on the signal circuits. I permanently attached one 'phone in the engine room to the signal circuit with a long cord so that I could use it and work the engine at the same time. One 'phone was placed permanently at the signal-button station next to the main drive-clutch couplings, another was given the repair gang. Each push-button station had two brass terminal screws connected to the signal circuit so that a 'phone could be attached. A telephone call was added to the sig-

nal code. The latter was not abandoned, but the 'phones were used when the code was found insufficient to convey the messages.

The 'phones have cut in two the time formerly wasted in making shutdown repairs. The shafting oiler also has a hand 'phone which he carries with him when at work on shafting. We also use them frequently when one of the engine-room force is at work on lighting circuits in parts of the mill remote from our interphone system.

A call bell operated from the engine room was placed in the main drive-clutch station and a signal bell in the boiler house is used to notify the firemen to stop or start the waterwheel which is near the boiler house and quite a distance from the engine room.



WIRING DIAGRAM OF ENGINE-ROOM SIGNAL AND TELEPHONE SYSTEM

BENJAMIN S. HANSON.

Broad Brook, Conn.

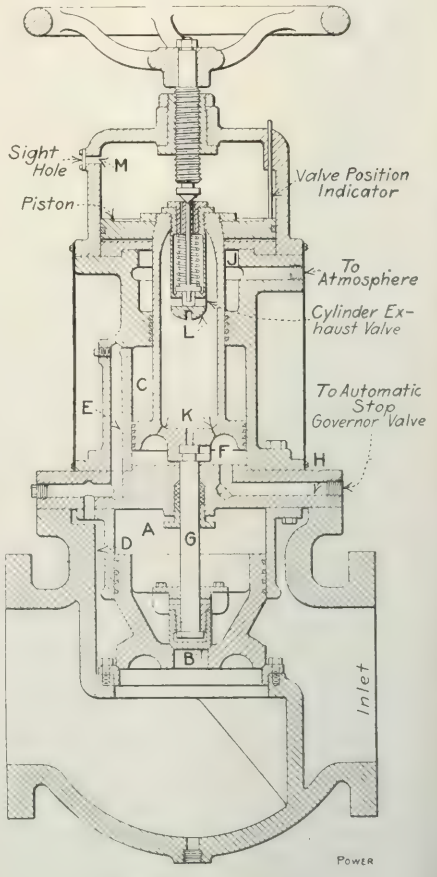
Experience with Automatic Throttle Valve

The illustration shows the automatic throttle valve of a 3000-kw. turbine. Steam is admitted at the upper side of the lower valve disk filling the space above the valve. By reason of leakage past the balance piston, immediately above the valve disk, full boiler pressure is established at *A*, in the balance cylinder and, so long as the pressure is on its seat. The pilot valve *B* being opened, the pressure in *A* is relieved, and the valve disk becomes balanced. The high-pressure steam entering the valve freely passes into the space *C* above the operating piston through the ports *D* and *E*. It also passes in limited quantities to the space *F* below the piston both by leakage past the piston and around the stem *G*, so, should there be no outlet for the steam at *H*, full pressure will be established at *F*, causing the valve to open as the handwheel is raised, as the upper part of the operating piston, where it enters the dashpot, is exposed to atmospheric pressure only.

Should the pressure be relieved from *F*, steam will be exhausted from below the piston and the pressure in chamber *C*, will force the valve to the closed position. On the upper part of the valve are the dashpot and piston, for preventing the valve from slamming violently maintained, it tends to hold the valve disk on its seat. There is also a valve-position indicator to show the position of the valve with reference to its seat. At *J* is a metallic packing with a connection to the atmosphere to carry away any leakage. There is also a device to prevent the sudden opening of the valve after it has automatically closed. Should the operator reset the automatic stop valve without first closing down the throttle-valve handwheel, it is possible to readmit steam to the turbine slowly instead of instantaneously turning on the full head of steam. This is accomplished by holes *K* through the operating piston, and a spring-loaded valve called the cylinder-exhaust valve, located in the upper part of the operating piston. The lower side of this valve is exposed to the full pressure, which, however, is insufficient to compress the spring. The valve being opened, the valve proper comes up against the body head. The upper stem, however, has a little farther to travel before the head on the lower end of it is stopped against the bushing on the upper part of the dashpot. This permits the cylinder-exhaust valve spring to expand a trifle, but still not sufficient for the steam pressure to force open the cylinder-exhaust valve *L*.

Should the valve close automatically, the operating piston with the valve *L* will descend from the upper throttle-valve stem, when the spring will be entirely released, and the valve *L* will be free to open. Then, if the automatic stop governor valve is closed at this time, thus preventing egress of steam at *H*, it will be impossible for the steam pressure to increase at *F*, because the steam will be free to escape through the holes *K*, the valve *L* and the sight-hole *M*. Hence the valve will not open until the handwheel has been screwed down, forcing the cylinder-exhaust valve *L* to its seat.

One of our turbines is able to handle the load, so we keep it on the line for two weeks, and then change over to the other, which is the same size. Recently, when the engineer on the afternoon watch, changed over the turbines, he found that the operating piston stuck when within about an inch of full open, and that the handwheel had to be screwed down a little to keep the cylinder-exhaust valve to its seat. Everything went well until the night engineer came on duty. He was an experienced and very careful man, but the throttle valve was a mystery to him. Being observant, he was not long on duty, when he noticed the handwheel on the throttle valve was not in the wide-open position, and proceeded to open it. This relieved the cylinder-exhaust valve, and exhausted the steam from *F*, forcing the valve to the closed position, shutting off steam. As a big load was being carried at the time, it did not take long before the turbine came to a stop.



AUTOMATIC THROTTLE VAVLE

It all happened so quickly that the engineer did not know what occurred or how to correct the trouble, so started the other turbine. The power was off the line for 15 min. The chief had the valve taken apart the following day and the operating piston put in the lathe and lightly turned down, which made it a good working fit. Now the watch engineers know more about automatic throttle valves than they ever expected to.

THOMAS SHEEHAN.

North Adams, Mass.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Scale in Turbine Sealing Glands

The letter in the Sept. 10 issue, relating to the means devised by Mr. Pivens to obviate trouble experienced from scale forming in the sealing glands on the spindle of his steam turbine was interesting. He is not the first who has had trouble from that source. His apparatus will accomplish the object for which it was built, if the check valve *K* is reversed from the position shown in the illustration in the above issue. With the valve placed as shown, when valve *D* opens to the atmosphere, the air would enter the condenser faster than the tanks and break the vacuum on the system.

The device as shown has another serious disadvantage, in that every time a tank of water is extracted from the condenser practically 20 cu.ft. of air will be admitted to the condenser in which there is 28 in. vacuum. This adds work to the air pump, and unless they are large, will impair the vacuum. I note that pipe *G* leading from the upper tank to the condenser is restricted in diameter $\frac{3}{16}$ in. to prevent impairment to the vacuum. This will also lower the capacity of the apparatus.

I have used a device for this purpose, and accomplished satisfactory results with apparatus that can be bought in the open market and will not admit air to the condenser. All that is required is an automatic vacuum trap that uses a jet of steam to blow the water out into a tank placed at an elevation to give the required pressure on the glands of the turbine. The trap should be set about 6 ft. below the connection in the exhaust end of the turbine and, where a barometric condenser is used, the suction side of the trap should be connected to the lowest point in the exhaust pipe.

Mr. Pivens states that he finds 4 lb. sufficient pressure at the inlet of the glands. My experience is that this pressure is not sufficient for most installations, and the builders' recommendation of 15 lb. will be found to give more reliable action. All turbines do not act alike; while 4 lb. might do in one case, other turbines will require more and engineers who are going to try this scheme had better place the storage tank high enough to give 12 to 15 lb. pressure. There is ample condensation in the exhaust end of the turbine to furnish sufficient water to seal the glands.

I placed some bullseyes in a jet condenser attached to a 2000-kw. steam turbine, and put electric lights inside of the condenser; it was interesting to watch the action of the water as it entered the condenser and the way it was affected when the steam struck it. One of the features that could be clearly seen was the water or condensed steam running down the inside of the exhaust connection from the turbine.

R. D. TOMLINSON.

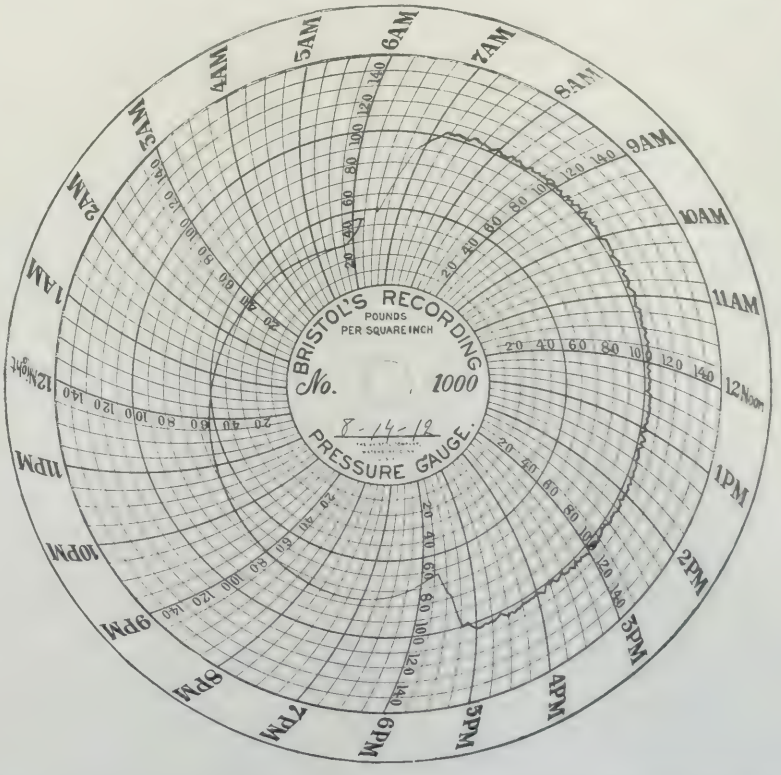
New York City.

Steam Chart from Hand-Fired Boiler

There appeared in a recent issue a steam chart showing the pressure maintained in a boiler for 24 hours' run. The letter only told the capacity of the boilers, and nothing about the load they carried. The accompanying chart is shown for comparison. We have two 54-in. by 16-ft. horizontal return-tubular boilers of about 75 hp. each. The draft is 0.19 and 0.21 in. respectively; no damper regulator is used. We burn about 7000 lb. of mine-run coal in 10 hr., and evaporate 92 cu.ft. of water per hour from 212 deg. This is equivalent to 8.2 lb. of water evaporated per pound of coal. We have a 400-amp., 220-volt direct-connected generator. Considerable steam is also furnished for rubber vulcanizers and other purposes.

H. M. WALKER.

Chicago. Ill.



PRESSURE CHART FROM HAND-FIRED BOILER NOT EQUIPPED WITH DAMPER REGULATOR

Air Compressor Crank Repair

On pages 432 and 433 of the Sept. 17 issue. T. H. Heath describes the method used to repair a disk crank which had a piece broken out of the eye. The crank disk was stated to have been 26 in. in diameter and of 3½-in. face. The repair consisted of a ring 1¾ in. thick, 3½ in. wide and bored out $\frac{1}{8}$ in. less than the diameter of the disk, heated and shrunk on the disk.

Under the above conditions, the inside diameter of the ring would be $25\frac{15}{16}$ in. bore. The mean diameter would be $27\frac{11}{16}$ in. Neglecting compression in the disk and ring, the extension of the metal in

the ring would be 0.1965 in. along the mean circumference, of which the length is 86.98 in., or a ratio between this extension and the mean circumference of 1 to 443. Assuming 29,000,000 as the modulus of elasticity,

$$\frac{29,000,000}{443} = 65,463 \textit{ lb. per sq.in.}$$

is the stress, which evidently is too high.

The above assumes that there is no compression in either the disk or ring. While there would be some compression which would tend to relieve this stress somewhat, it would not be sufficient to reduce it to a safe working value. Had the allowance for shrinkage been a scant $\frac{1}{32}$ in. (0.025 in.) the repair would have been more reliable.

J. K. McINTYRE.

Newburgh, N. Y.

Excess Air Indicated by Superheat

The above subject was discussed editorially in the Aug. 20 number of POWER. With the same fuel as was used in the case mentioned and all other conditions the same, the temperature of combustion depends on the amount of air in the mixture. In ordinary practice the gas per pound of fuel runs from 16 to 35 lb., these figures were found in actual practice, one being a large chain-grate installation and the other a hand-fired 500-hp. plant.

For convenience of figuring and to be within more common practice, take a case where the same coal is burned with 20 lb. of dry stack gas per pound of coal and in the second case 30 lb. of dry gas per pound of coal; the amount of gas per pound of coal can be determined from the coal and gas analyses. If the fuel contains 75 per cent. carbon, 4 per cent. hydrogen and 8 per cent. oxygen and the gas analysis shows 9 per cent. CO_2, no CO, for every 12 parts of carbon burned, it requires 32 parts of oxygen to attain complete combustion. The atomic weights of carbon and oxygen are 12 and 32, respectively. Then to burn 12 tons of carbon into CO_2 (carbon dioxide) will require 32 tons of oxygen. To burn 0.75 lb. of carbon, therefore, will take

$$(0.75 \div 12)\ 32 = 2 \textit{ lb. of oxygen}$$

To get this amount of oxygen from the air will require

$$2 \div 0.23 = 9 \textit{ lb., nearly}$$

As 0.23 is the amount of oxygen in a unit volume of air by weight, the other 0.77 is nitrogen. The volume of CO_2 is the same as the volume of oxygen used to burn the carbon and from analysis is 9 per cent. of the volume of the stack gases. The volume of nitrogen is to the volume of oxygen as 0.23: 0.77; therefore, the per cent. of nitrogen taken in with oxygen will be the same as their volume to each other, that is,

$$X : 9 :: 0.77 : 0.23$$

$$X = 9 \times 0.77 \div 0.23 = 30 \textit{ plus}$$

The total per cent. of CO_2 and nitrogen will be

$$9 + 30 = 39 \textit{ per cent.}$$

of the total dry stack gases per pound of coal. If 39 per cent. of the flue gases constitutes 9 lb. of air the total weight of gas per pound of coal will be

$$(9 \div 39)\ 100 = 23.07$$

As 0.75 lb. of carbon and the oxygen and hydrogen present in the coal and passed with the stack gases have not been taken into account, the weight of the total gas will be

$$23.07 + 0.75 + 0.08 + 0.04 = 23.94$$

With no loss of combustible the temperature of the fire will be the total heat per pound of fuel divided by the product of the pounds of gas and the specific heat. The specific heat of gases runs near 0.24 in very many cases in practice and can be taken as such. If the combustible consumed developed 12,500 B.t.u., the temperature must be about 2230 deg. F., moisture neglected. With 20 lb. of gas per pound of fuel the temperature would be 2600 deg. F. and with 30 lb. 1740 deg. F. It is then apparent that the temperature of the gases is directly proportional to the amount of air.

The heat absorption of the boiler depends on the temperature difference, velocity of the gases and the cross-section of the gases. With an increase of the gases the velocity must be greater to take care of the increased volume; therefore, better absorption. With a decrease of the gases, the velocity is less, but the higher temperature difference compensates for the lack of velocity and the gases will leave the absorbing surface of the boiler at about the same temperature.

This fact emphasizes strongly that the rating of boilers at 10 sq.ft. of heating surface to a horsepower is liberal and that seldom are all of the surfaces worked to full capacity. When a boiler is not overworked and the same rate of combustion is maintained, the final temperature of the gases is not affected by variation of the amount of air, unless this variation is stretched beyond the usual practice.

If the gases enter the superheater at 600 deg. F. and leave at 450 deg. F., there is a drop of 150 deg. in the superheater and with 20 lb. of gas per pound of combustible the superheater will absorb

$$20 \times 0.24 \times 150 = 720 \textit{ B.t.u.}$$

for every pound of combustible. With 30 lb. of gas per pound of combustible the superheater will absorb

$$30 \times 0.24 \times 150 = 1080 \textit{ B.t.u.}$$

The resulting superheat will be in the proportion of 1440 to 2160, if the specific heat of superheated steam is 0.5. This means that every pound of combustible will deliver into the superheater, when 20 lb. of air is used, enough heat to superheat 100 lb. of steam 14.4 deg. F. When 30 lb. of air is used per pound of coal, the steam will be superheated 21.6 deg. F. It is useless to call attention to the fact that this is distribution of the heat and that final economy will be in favor of burning the fuel with 20 lb. of air. In practice the temperature of the gases entering the superheater is much higher than 600 deg. F., and this temperature is taken simply as an illustration. With higher temperatures such as are used in practice the emphasis in favor of more air per pound of fuel in the superheater will be still more striking.

With the amount of steam, temperature of gases and degree of superheat known, the pounds of gas can be found. With the temperature, quantity of gas and steam known, the superheat can be found. The rate of combustion and flow of steam being the same, the superheat will vary with the amount of gases per pound of fuel.

H. MISOSTOW.

Department of Smoke Inspection,
Chicago, Ill.

Tilt Trap as a Boiler Feeder

Some interesting questions regarding that always important matter of the proper way to feed a boiler are brought up by Mr. Wickland's letter in the Sept. 24 issue.

Nowadays all uptodate plants have some arrangement for feeding hot water. Cold water not only causes injurious strains in the boiler tubes and shell, but usually contains a considerably larger amount of scale-forming matter and corrosive gases than hot water which has passed through an open heater.

The tilt trap, as Mr. Wickland says, is much like the injector in some ways. It will, however, operate with any temperature of the feed water, which cannot be said of the injector. The big disadvantage is that the feed is intermittent and the degree to which this is disadvantageous varies with the temperature of the water passing to the boiler. The temperature is never equal to that of the water in the boiler, with either an injector or tilt trap. With steam at 125 lb. gage the temperature is 353 deg. F. in the boiler. That the water coming from the trap would ever exceed 250 deg. F. is doubtful, and it is often much lower. Supposing the trap discharged 3 gal. of water at 250 deg., then

$$(353 - 250) \times (8 - \tfrac{1}{3} \times 3) = 2570 \textit{ B.t.u.}$$

is required instantly to heat this water to the boiling point in the boiler. This will cause the boiler to make less steam di-

rectly after the trap discharges than before. The evaporation will, therefore, be more or less intermittent, depending upon the temperature of the trap discharge, and the capacity of the trap as compared with the boiler rating.

What effect a large volume of comparatively cold water being suddenly admitted will have on the circulation will depend largely on conditions. In the modern return-tubular boilers the circulation is fairly rapid, particularly when they are developing their rated horsepower or more. Then it is doubtful if such intermittent injections would work appreciable harm. However, many boilers in use have a circulation that is anything but what it should be. A volume of water, much below the temperature of the boiling water would in such cases, particularly if injected through the blowoff connection or at a very low point in the drum, tend to "pocket," making the heat transmission at such points poor and consequently tending to overload the other parts of the boiler.

Another point must not be overlooked in considering this matter. The large volume of water, which as already pointed out is comparatively cold, even though above 212 deg. F., coming in contact with the shell and tubes, will cause local contraction, thereby setting up strains. There is thus the same fault as with cold-water feeding. This is a most dangerous evil, too, for it is something which we cannot see by trying the gage cocks or looking at the steam gage, and finally when the tube or shell gives way in the prime of life, as it were—even then we are apt to forget that cold-water strains have helped it along.

The question whether it is advisable to use live steam for heating water when exhaust steam can be had for the purpose has been answered so many times that the uptodate engineer says "No" even in his sleep. If exhaust steam is being wasted then the use of live steam, either by a tilt trap or injector, is certainly not advisable in any plant over 50 hp. in size having any pretensions to efficiency. The use of exhaust steam in this way is pure "velvet" and often goes far to make the purchase of a boiler-feed pump advisable to replace an injector or tilt trap in a plant where there is no other supply of exhaust steam.

A good open feed-water heater will receive the condensate and store it for boiler feeding, allowing the boiler-feed pump to take a continuous supply. Back pressure can be carried if desired, provided the heater has a float instead of a water seal, or the steam condensation can be returned to the heater through an ordinary steam trap. It seems as though the heater and feed-pump combination is worth the increased cost when we consider that the heater precipitates the temporary hardness, expels the air and other corrosive gases, and not only feeds a purer water, due to the condensation of the exhaust steam required to heat the water, but also employs a waste product instead of valuable live steam, thus adding to the steaming capacity of the boiler.

EDWARD H. ROBIE.

Philadelphia, Penn.

Mr. Wickland raises the question as to the efficiency of a return trap as against a steam pump and compares its operation with that of an injector. An injector is capable of not only feeding water into a boiler but also of lifting it from below the boiler water line and in its operation the steam is mixed with the water and passes again into the boiler, acting as Mr. Wickland states, like a feed-water heater.

The operation of a return trap is entirely different and hardly admits of comparison with that of an injector. In the first place a return trap will not lift water from below the boiler water line and is only successful when placed at least 3 ft. above the water line. Two traps must, therefore, be used if the water is to be lifted, one to raise the water and the other to return it to the boiler. Secondly, the principle on which a return trap works is to have the water in the trap under the same pressure as the boiler and depend on gravity for feeding it into the boiler. This is accomplished by admitting steam into the bucket of the trap and the steam required is about three-quarters of the volume of the bucket for each dump, or approximately that of the water discharged. For 100 lb. pressure the weight of a cubic foot of steam is approximately 0.25 lb. while that of 1 cu.ft. of water is about 60 lb. Therefore, each 100 lb. of water will require about 0.4 lb. of steam.

If the steam pump is used each 100 lb. of water will require from 1½ to 2 lb. of steam, so that the amount necessary for a return trap is from one-third to one-fourth that for a pump. Moreover, with a return trap the heat in the steam is at all times returned to the boiler, while with a pump the conditions must be such that all the steam is used in heating the feed water so that a pump can equal in economy a return trap. This is a condition that is more rare than common. If two traps must be used the economy will be the same but the steam consumption would be double.

Added to the greater economy of a return trap the fact that it requires practically no attention or repairs makes it a highly desirable and economical means of feeding water to a boiler.

The last three questions asked by Mr. Wickland could only be answered after a long and elaborate series of experiments had been made, and which, up to the present time, I do not think has been done. In any event the effect of intermittent admission of feed water would be slight, so slight that it can be neglected as a factor in determining the economy between a return trap and a feed pump.

W. L. DURAND.

Brooklyn, N. Y.

Systematic Note Keeping

Under this heading, Lloyd V. Beets wrote, in the Sept. 3 issue, of his method of recording those ideas and incidents which he later worked up into articles for the technical magazines, and asked for suggestions as to any improvements that could be made in it.

His method is certainly efficient and systematic, and the only criticism that could be made would be that it might be simpler and time and labor saved which is now used in copying the notes from the desk pad to the ledger. In my own work I always carry in my pocket a small loose-leaf book and, when anything worth recording happens, make notes in it. For a short article it is often only necessary to write a title at the head of the page with the assurance that, when it is desired to write on the subject, the title will be enough of a reminder around which to construct the article. For a longer article more notes and data may be placed on the same page under the heading. It may also be necessary to construct a skeleton for the article consisting of a number of subheadings arranged in the order that they would be written about. There is room for all of these things on the page and they can be altered or added to from time to time at any date between the beginning and completion of the article as the data is always at hand where it can be worked on during spare moments.

The heading and skeleton of the article gives an opportunity to write a rough draft of the entire article, during spare moments. Much of my work has been done while waiting for or traveling on cars, at lunch time, or during other short intervals that otherwise would have been wasted. When the article has been finished the notes in the loose-leaf book can be taken out and thrown away so that no dead matter is ever carried around, which would be the case with the ordinary type of notebook.

To keep a record of the articles between the time they are written and published, I find it best to typewrite them, making a carbon copy which is filed as a record until the article is published.

G. H. MCKELWAY.

Brooklyn, N. Y.

The demand for tank steamers for transporting petroleum in bulk to Oriental countries is leading to the conversion of ordinary steamers into tank vessels. It is interesting to note that Hongkong shipyards are undertaking such work.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Steam for Oil Burners

In using steam atomizers for burning fuel oil under boilers, what proportion of steam generated is required for operating the oil burners?

R. G. L.

From 2 to 10 per cent., depending on the type of burners employed and the skill in handling.

Pounding Check Valve

How can the pounding noise of a check valve on the supply line to a house tank be prevented?

C. P.

Pounding of a check valve on the discharge line from a pump can usually be stopped by placing a good-sized air chamber in the line near the inlet to the check valve, and running the pump at a higher speed.

Advantages of Synchronous Booster Converter

What are the advantages of a synchronous booster converter over a regulating pole or split-phase converter?

G. B.

The synchronous booster converter has a higher efficiency, especially at lower voltages, and better commutation, especially at maximum load and high voltage, besides being lighter in weight and occupying less floor space.

Utilizing Heat of Blowoff

Can the heat contained in scum blowoff water be employed for heating fresh feed water in an ordinary closed heater to as high temperature as by use of exhaust steam?

F. P. K.

As a large part of the heat in exhaust steam is in the form of latent heat, exhaust steam used in a closed heater of proper proportions is capable of imparting practically the same temperature to feed water as the temperature of the steam itself. Blowoff water could not conveniently be employed before being reduced to atmospheric pressure and liberation of heat in excess of latent heat at 212 deg. F. The transfer of heat would necessarily be to a lower temperature of the feed water and the efficiency of the heating surface would rapidly become impaired by impurities contained in the blowoff water.

Vacuum Breaker on Open Heater

Why is it necessary to use a vent pipe or vacuum breaker on an open feed-water heater?

C. G.

A vent pipe is employed to prevent accumulation of air or the formation of a vacuum within the heater. A vacuum is liable to occur when only a small amount of steam is discharged to the heater. A vacuum breaker is used between the engine cylinder and the heater to prevent water of the heater or condensation in the exhaust pipe from being carried into the engine cylinder should a partial vacuum be formed before the exhaust valves open, as result of early cutoff.

Loose Crankpin

Can a good, safe job be done by riveting fast a crankpin on a 16x42-in. Corliss engine which has been run loose in the crank for several days?

T. E.

If the pin is simply loose without any appreciable variation from the size of the crankpin hole, a good job of riveting may answer for temporary use, but the repair should be regarded as temporary until replaced by a pin inserted with a press fit. If the hole is larger than the old pin, the former is probably worn out of round and reboring the crankpin hole as well as a new pin will be required as a permanent and safe repair.

Resistance of Stay Bolts

What boiler pressure would be resisted with a factor of safety of 7 by standard V-threaded firebox stay-bolts ¾-in. outside diameter made of iron having a tensile strength of 42,000 lb. per sq.in. of cross-section, spaced 4¾x4¾-in. centers?

F. M.

The diameter of stay-bolts at the bottom of the thread would be 0.6057 in., and the area 0.288 sq.in., and the strength of each would be

$$42{,}000 \times 0.288 = 12{,}096 \text{ lb.}$$

hence the allowable strain per stay-bolt with a factor of safety of 7 would be

$$\frac{12{,}096}{7} = 1728 \text{ lb.}$$

Each stay-bolt would support

$$4\tfrac{3}{4} \times 4\tfrac{3}{4} = 22.5625 \text{ sq.in.}$$

therefore the allowable boiler pressure per square inch resisted by the stay-bolts would be

$$\frac{1728}{22.5625} = 76.58 \text{ lb. per sq.in.}$$

Drips of a Vacuum Heating System

In a vacuum-exhaust steam-heating system, how can the drips of mains be connected to the returns so as not to sustain loss of steam or impair vacuum in the returns?

F. M. B.

A main drip should be sealed by running it so as to form a deep trap before discharging to the returns. The highest point of the trap so formed should be 20 to 30 in. lower than the connection to the steam main, and the deeper the trap the higher the vacuum may be carried before siphoning will take place.

False Admission Line on Diagram

What defect in operation of an indicator would falsely produce resemblance to a perfect admission line, forming a sharp corner like a perfect cutoff where it intersects the expansion line of the diagram?

A. C.

If the pressure during admission is higher than the instrument can properly indicate, as might happen when using a spring which is too weak for indicating the initial pressure, or when the piston is prevented by dirt or other obstruction from carrying the pencil to the proper height for the scale of spring employed, then a false admission line will be drawn below the height of true admission pressure and it will be parallel with the atmospheric line, in that respect resembling a perfect admission line.

The diagram will present the appearance of a true diagram having its upper part removed by a straight line dropping by a sharp turn into the true expansion line after cutoff has actually occurred, and insinuate a sharpness of cutoff that would be impossible excepting by absolutely perfect cutoff mechanism.

Study Questions

This Week's Questions
Last Week's Answers

(121) A storage battery consists of 25 cells arranged in series-multiple with five groups in parallel and five cells in series in each group. The electromotive force of each cell is 2 volts, and its internal resistance is 0.04 ohms. With an external resistance of 0.5 ohms, what current will pass through the circuit?

(122) An engine takes steam the full stroke and operates condensing. The admission pressure is 60 lb. abs., and the back pressure 2 lb. abs. Neglecting condensation, how much water must be evaporated for this engine per effective indicated horsepower per hour if the specific volume of steam at 60 lb. is 7.037 cu.ft.?

(123) A tank holds 100 gal. and has two discharge outlets, one discharging 14 gal. per min., and the other 9 gal. per min. Using the outlets one at a time, how long must each be open to empty the tank in exactly 10 min.?

(124) What is the content of a cylindrical air tank having two bumped heads (one convex and one concave) if the body is made from a sheet 6 ft. square, and all seams have 3-in. lap (neglecting thickness of the material)?

(125) A beam 24 ft. long has two supports, one at the right end and the other 4 ft. from the left end. What is the load on each support if the beam has a uniform load of 800 lb. per ft. of length, a concentrated load of 9000 lb. 8 ft. from the right support and one of 5000 lb. at the extreme left end.

Answers to the above will appear in the next issue. Answers to last week's questions follow:

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(116) The part of the tank displaced by the wall corresponds to what is known in geometry as a cylindric ungula, and when the plane cutting off an ungula passes through the center of the base the volume of the ungula may be found by the formula

$$V = \tfrac{2}{3}\, r^2 h$$

where

r = Radius of the base;
h = Height of the ungula.

In the problem given $r = 6$ in. and $h = 9$ in. Then

$V = \tfrac{2}{3} \times 6^2 \times 9 = 216$ *cu.in.*

is the volume of the ungula. Had the cylinder been complete its volume would have been the area of the base times the height, or

$3.1416 \times 6^2 \times 24 = 2714.34$ *cu.in.*

The content of the tank, therefore, is

$2714.34 - 216 = 2498.34$ *cu.in.*

(117) Let x = Price of the packing per pound. Then $\frac{3x}{2}$ was the change offered. The cost of the packing first cut would have been

$$\frac{3x}{2} + 10$$

and the cost of half of it

$$\frac{\frac{3x}{2} + 10}{2}$$

This would be 5c. less than the money first offered; hence

$$\frac{\frac{3x}{2} + 10}{2} = \frac{3x}{2} - 5$$

$$\frac{3x}{2} + 10 = 3x - 10$$

$$3x + 20 = 6x - 20$$

$$3x = 40$$

$x = 13\frac{1}{3}$c. = Price of the packing per pound.

(118) One pound of water raised from 50 to 75 deg. F. absorbs

$75 - 50 = 25$ *B.t.u.*

Then 5½ lb. absorbs

$25 \times 5.5 = 137.5$ *B.t.u.*

1 *Heat unit* $= 778$ *ft.-lb.*

Then 137.5 B.t.u. corresponds to

$137.5 \times 778 = 106{,}975$ *ft.-lb.*

(119) At 60 deg. F. 1 cu.ft. of water weighs 62.37 lb. Then 2 gal. of water weighs

$$\frac{231 \times 2}{1728} \times 62.37 = 16.675 \text{ lb.}$$

When its temperature is raised from 60 to 69 deg. F., or 9 deg., the heat it absorbs is

$16.675 \times 9 = 150.075$ *B.t.u.*

This, then, is the quantity of heat lost by the iron in cooling from its unknown temperature to 69 deg. F. The specific heat of cast iron being 0.13, it loses 0.13 B.t.u. in cooling 1 deg. Then

$$\frac{150.075}{0.13} = 1154.423 \text{ deg. F.}$$

is the number of degrees the iron has decreased in temperature. Hence its original temperature was

$69 + 1154.423 = 1223.423$ *deg. F.*

(120) The friction F can be found from the formula

$$F = fN$$

where

f = Coefficient of friction = 0.06;
N = Normal pressure, lb. = $10 \times 2000 = 20{,}000$ lb.

Then

$F = 0.06 \times 20{,}000 = 1200$ *lb.*

The distance through which it is acted against for one revolution of the shaft is the circumference of a circle of the mean radius, or

$2\pi r = 2 \times 3.1416 \times 6 = 37.699$ *in.*

In feet, this is

$$\frac{37.699}{12} = 3.14 \text{ ft.}$$

The shaft making three revolutions every 2 sec. makes

$$\frac{3 \times 60}{2} = 90 \text{ r.p.m.}$$

Then in a minute the distance through which friction is acted against is

$90 \times 3.14 = 282.6$ *ft.*

The friction being 1200 lb. the energy consumed in friction is

$1200 \times 282.6 = 339{,}120$ *ft.-lb.*

and the horsepower consumed in friction

$$\frac{339{,}120}{33{,}000} = 10.276 \text{ hp}$$

NEW PUBLICATIONS

KNOTS, SPLICES AND ROPE WORK. By A. Hyatt Verrill. Published by the Norman W. Henley Publishing Co., 132 Nassau St., New York, N. Y. 1912. Size, 4½x6¾ in., 102 pages, illustrated. Price, 60c.

This book is written in a practical manner, giving complete and simple directions for making the most useful and ornamental knots in common use. In addition to the text matter there are 150 illustrations which show how each knot, tie and splice is formed and its appearance when finished. The chapters on splicing, pointing, seizing and serving make the volume very complete.

ENGINES AND BOILERS. By W. McQuade. Published by A. Wheaton & Co., Ltd., 143 Fore St., Exeter, England. 1909. Size, 5½x8½ in.; 87 pages; illustrated. Price, $1.50.

Although this volume is intended as a handbook for young engineers on the construction and working of steam, gas, oil and petrol engines and steam boilers, it deals with each subject in such a limited manner that no one subject is given the attention it deserves.

The illustrations do not typify modern construction as found in this country, many showing the design of engines now obsolete. Stationary and marine construction are treated irrespective of their application one with the other. The only subject treated at any length is that of slide-valve action. Nine pages have been devoted to boilers of various types, and very little has been said regarding any of them. In reality the volume is more in the nature of a very elementary introduction to the steam-engineering field.

AERIAL OR WIRE ROPE WAYS. By A. J. Wallis-Taylor. Published by Crosby, Lockwood & Son, 7 Stationers' Hall Court, Ludgate Hill, E. C. London, 1911. Size, 5½x9 in., 246 pages, illustrated. Price, $3, net.

This rewritten work contains parts of an original work now out of print, but revised and brought uptodate. More subject matter and illustrations have been added to the chapter devoted to the details of construction, electrically driven wire-rope ways and miscellaneous information.

Descriptions of a number of rope-way installations are described and illustrated which give a very comprehensive idea of the various uses to which aërial rope-

ways may be put, also installed and operated.

A particularly interesting chapter concerns coaling vessels at sea, in which a number of the plans that have been tried out are illustrated and explained. The volume treats on the method employed for hoisting and conveying both in this and foreign countries.

Much miscellaneous information is given for calculating the strain on the carrying rope; also hints as to the splic-securing, preserving and care of wire ropes.

HIGH EFFICIENCY ELECTRICAL ILLUMINANTS AND ILLUMINATION. By Rollin W. Hutchinson, Jr. Published by John Wiley & Sons, New York; Chapman & Hall, Ltd., London. 278 pages; 147 illustrations. Cloth, $2.50 net.

This work gives a complete and comprehensive description of the principal types of modern electric lamps manufactured in this country and Europe together with a discussion of their various characteristics and uses. The text is written in a manner making it intelligible to the layman as well as to the engineer and designer. The author has not attempted to introduce exhaustive details, but has aimed to make his book a complete work of general reference.

In the first chapter, the theory of light and color is presented according to the latest authentic observations. The structure of the eye is described, together with the causes of the sensations it experiences from lights and colors of varying intensities. Color values of various sources of light are given as are also the comparative calorimeter readings.

A complete explanation of the methods and procedures followed in photometry is presented in the next chapter. Various leading types of photometers are described by diagrammatic illustrations. Definitions are also given of the various units of light measurement.

Mr. Hutchinson devotes the next chapter to a treatise on metallic filament and electrolytic incandescent lamps. A short general history of the evolution of these types is given, together with a discussion of the chemical and physical properties of the metals used, including the sources from which they are obtained. The various processes of filament manufacture are explained and a well illustrated description is given of the principal types of lamps using the filaments. An especially noteworthy feature of this chapter is the presentation of statistics. Among these is given the total cost of producing light with the lamps, at different voltages and at various power rates, together with figures showing the comparative cost of producing given amounts of light with carbon, gem, tantalum and tungsten lamps. First costs are considered at length as are also the courses followed by various companies in supplying the lamps to customers. The difficulties encountered in the manufacture and use of metallic filament lamps are enumerated and explained, together with the means used for obviating them.

A chapter is devoted to arc lamps, the author setting forth the main points in the design of the principal types, giving especial consideration to the flaming arc variety. By diagrammatical illustrations, the mechanisms are clearly explained.

Vapor lamps are next taken up, the Cooper-Hewitt, the Kuch quartz mercury vapor lamp and the Moore tube light being described in detail. Several photographs of actual installations of some of these lights are shown to illustrate their practical efficiency.

The author devotes the remainder of the book to discussing the fundamental principles involved in the efficient and economical illumination of interiors and streets. Throughout the work careful attention has been given to the presentation of the statistics most needed by the estimator and engineer, together with the uninvolved descriptions sought by the layman. For a general reference upon electric lamps and illumination, the book is commendable.

ELECTRIC CRANE CONSTRUCTION. By Claude W. Hill. Charles Griffin & Co., Ltd., London; J. B. Lippincott Co., Philadelphia. Cloth; 306 pages, 6x9 in. Over 366 illustrations; 23 tables. Price, $8.

The author of this work has made it primarily a book of reference for engineers rather than an elementary treatise for the general reader. The descriptions follow English practice almost entirely, the illustrations being taken from the products of English manufacture rather than from cranes in general. However, the portions of the book devoted to the methods of procedure for computing the dimensions of the various parts and for estimating upon the design and construction can be applied to general use.

The opening chapters are devoted to descriptions of the different types, each chapter dealing with one particular class. The headings, in order of presentation, are overhead cranes; locomotive and portable jib cranes; derrick cranes; transporters; shear legs; revolving cantilever cranes and cableways. In the consideration of each class, machines of various makes and capacities are described in logical order, numerous photographic and diagrammatical illustrations being presented, together with large plates taken from working drawings.

Proceeding to the calculations of the component parts and elements, the author first discusses the procedure used in computing the driving power required. The figures are based upon observations taken from actual service, hence are of practical value as well as theoretical. Several tables and curves present data useful for this branch of design.

Starting torque and acceleration are next considered followed by a chapter devoted to the design of crane structures. Formulas for the computation of the stresses and loads involved are derived mathematically and discussed, the methods used by leading authorities being presented and explained in due order. Struts and columns are first considered followed by a section devoted to girders. Numerous curve sheets and diagrammatical plates illustrate the text.

In the next two chapters dealing with machine design and brakes, the calculation of dimensions of shafting, bearings, pedestals, wheels and the component parts of brakes is explained, the illustrations showing the construction of various details.

The author begins the chapter on toothed gearing by explaining the application of the lever principle to gearing, force diagrams being used for clearness. He then discusses the determination of the proper size and shape of the teeth. The forms of teeth best suited for various conditions are shown and the reasons for their use are clearly presented. This chapter, while giving the detailed information required by the engineer, is also intelligible for the most part to the layman.

Hooks, lifting magnets, ropes and chains are next considered, a noteworthy feature in this connection being the presentation of several tables giving in systematic order, the various properties of chains and steel wire ropes necessary to the designer. Following three chapters are devoted to the electrical equipment, including the design of magnets, motors, controllers, etc. This portion is involved for the most part with mathematical derivations; hence is of more value to the engineer than to the general reader.

Concluding the work with a treatise on installations, the author gives general directions for the selection and use of the equipment under various conditions. A number of prominent installations now in service are described.

The text of the book, considered from an engineering standpoint, has been well prepared, careful attention having been given to detailed information required by the designer, the data needed by the estimator, the explanations of practical value to the construction engineer and the theory sought by the student. However, the author's copious use of large folded plates for illustrating is to be condemned rather than approved, in that the cumbersome bulk thus added is a decided detriment to ease and convenience in the use of the book. The plates could have been reduced and printed upon regular pages without materially affecting the purpose for which they were intended. Aside from this objectionable feature, the book ranks well as a work of reference for students and professional crane men.

Smoke Abatement in Scotch Cities

The gas and electricity departments of Glasgow Corporation are competing in the exhibition of appliances for the reduction of smoke from furnaces and kitchen ranges. The electricity department has a complete electrically fitted restaurant, where cooking, kitchen work, cleaning, heating and lighting are effected by electric current.

The gas department has a "gas-equipped house," in which cooking and heating are done by gas fires, and all lighting is obtained from incandescent-gas burners of different types. The corporation is encouraging the reduction of smoke by holding these periodical exhibitions, and also lending gas cookers free, having loaned 37,000 since Mar. 1.

Gas Engine Association Convention

The National Gas Engine Association will hold its annual meeting at the Claypool Hotel, Indianapolis, from Dec. 3 to 5. Among the papers to be read are: "Gas Engine Design," Prof. C. D. Wagner; "Gas Producer for Power and Foundry Service"; "Gas Engine Economy," A. E. Potter; "Two-cycle, Two-cylinder Opposed Engines," Ben Heer. There will be the usual exhibits of accessories, engine parts, etc., and space for exhibits may be had by applying to R. A. Oglesby, Hercules Electric Co., Indianapolis, or to Albert Strittmatter, secretary, 224 East Seventh Ave., Cincinnati, Ohio.

New Plant in Service

The trial runs of the plant in Bamberger & Co.'s new department store at Newark, N. J., have been completed and the equipment is now carrying the daily load. This plant is one of the largest and most uptodate department-store plants in the East.

Transfer of Atlas Works Completed

The transfer of the buildings of the old Atlas engine works, Indianapolis, Ind., to its new owners was made in Superior Court No. 4 of that city on Oct. 15. The consideration was $1,500,000.

The new concern will be known as the Lyons-Atlas Co. and its officers are: President, James W. Lyons; vice-president, William P. Lyons; secretary and treasurer, George Lyons. All are from Chicago. James W. Lyons was formerly with the Allis-Chalmers Co.

Attorney for the holders of patents were also present to protect the interests of their clients.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Discarded hoisting rope is being converted into cigarette paper in Europe. First time we have been able to give that odor its proper designation. Thanks be, they can't utilize wire rope—yet!

Hall of Fame? Sure! E. D. Bangs sells cylinder oil for those bangs in your engine; H. L. Lapham, in setting his valves, will lap 'em; Smoke Inspector Oyster keeps mum, and writes what he has to say, and they can't distribute the current without Cable (J. A.).

Reuben, Reuben, I've bin thinkin'
What a glad world this will be
When them b'ilers cease their bustin'
And get safe for you an' me.

Laws don't seem to make us keerful,
Folks gits reckless jist ther same;
An' when we hev jined ther angels
Jury sez we was to blame!

HORAN—Th' ol' man sez that main damper nades renewin,' Terence.

DORAN—It's damper-tic'lar he's gettin' all to oncet.

Noted surgeon says that the heavier the service, the bigger the brain, and he has weighed up a three-pounder recently willed to him. Based on this theory, the twelve-hour, seven-day engineer ought to look like a dirigible balloon with feet under it; the service is heavy enough.

"Liquor in the Cooler"—Headline.

All right, R. M'Geddon, we have been battling some time to do just this thing, but the nearest we have yet come to "rich liquor" is that amber-hued stuff that's helped Milwaukee to enjoy more fame.

"A bed-cover and sheet wrecks a tandem-compound engine," says the *Travelers Standard*. In falling, they were drawn in the engine-room window by the revolving flywheel and lodged in the shaft governor. Safety engineers, take notice: there's a new hazard among you.

A millionaire, one Lemon (pronounced "Lay-mong" for a millionaire), is asked by the "Adriatic's" commander to "cease sending champagne and gold coins" to his stokers. Never mind, Cap, this proceeding will never become a habit. Most stokers (and editors) only know the lemon with the little "l."

SOCIETY NOTES

On the invitation of the Commonwealth Edison Co., it was unanimously voted by the executive committee of the National Electric Light Association to hold the 1913 convention in Chicago.

At the annual meeting of the American Society of Mechanical Engineers, to be held in New York, Dec. 3 to 6, sessions will be given to the following subjects: Textiles, administration, cement manufacture, machine-shop practice and railroads. The Gas Power Section will also hold its usual meeting. On Wednesday evening, Dec. 4, the members of the society will tender a reception to Prof. John E. Sweet, honorary member and past president, who was one of the founders of the society, and who celebrated his ninetieth birthday Aug. 21 last.

Dean John Price Jackson, of the School of Engineering of the Pennsylvania State College, reports that the enrollment of freshmen in engineering numbers 354, the total enrollment of engineering students being 964. Prof. J. A. Moyer, recently of the University of Michigan, professor of mechanical engineering, is head of the department, and Prof. George W. Eckhard, recently professor of civil engineering at Milliken University, is assistant professor of structures.

OBITUARY

William P. Abbey, former president of the Sunderland (Mass.) Electric Light & Power Co., died at his home in North Sunderland, Mass., on Oct. 5. Mr. Abbey was born in Sunderland on Aug. 1, 1876.

PERSONAL

J. F. Nagle, for several years erecting engineer for the Allis-Chalmers Co. and the William Tod Co., of Youngstown, Ohio, and lately master mechanic of the New York, New Haven & Hartford R.R. power house at Cos-Cob, Conn., has been appointed superintendent of construction for the Boston Sanitary & Development Co., Boston, Mass.

Karl E. Eriksson, of Stockholm, Sweden, is in this country studying American practice in high-tension engineering. He is a graduate of the Royal Technical Institute of Stockholm and was an instructor in its department of electrical engineering for four years. He has been retained by the Swedish patent office as expert on electrical matters. Mr. Eriksson has conducted research work on high-tension phenomena and has written articles on protection of high-tension lines and other matters of a theoretical character, all of which have been published by the Swedish Technological Society.

Moments with the Ad. Editor

One of the sights most interesting to the visitor to New York at present is the giant Woolworth building—the newest and highest of the sky-scrapers to lift its head above Manhattan's famous sky-line.

This building as everybody knows is the result of the success of the Woolworth 5 and 10 cent stores. Probably you have one of these stores in your own town.

The most remarkable thing about the whole thing is that back of it is the personality of one man—a man in no sense a genius, just an everyday, companionable, modest man, but one who was capable of seeing a big opportunity and of seizing it.

In a recent interview Mr. Woolworth talks of his early life:—

"There isn't much to tell. I stayed there until I was twenty-one, working in the fields in the summer, attending district school in the winter, and finishing off with a couple of terms in a business college in a neighboring town. I never liked farm life and always wanted to get into the dry goods line, so the first chance I got I took a job in Watertown, N. Y. I thought perhaps they'd give me a small salary to start with, but the head of the firm contended on the contrary that I ought to pay him tuition while I was 'learning the business,' so we split the difference and I worked three months for nothing. After that I received $3.00 a week with the promise of a semi-annual raise of 50 cents a week. At the age of 26 my income was $6.00 a week—hours 7 a. m. to 10 p. m.—and when my earning capacity reached $8.50 a week I married the little woman who has been my best friend for nearly forty years."

Nothing extraordinary about that, is there? But then Mr. Woolworth got the idea which was to make him rich and famous. He got the idea of establishing 5 and 10 cent stores. At first he was not successful but he stuck to it.

And to make a long story short, today the $13,500,000 Woolworth Building lifts its head to the clouds.

"Now, looking backward, Mr. Woolworth, what do you consider has been the secret of your success?" he was asked.

"First, because I gained a reputation of giving value for value," was one of the things he replied.

"Value for value" a simple phrase and yet consider all it means. No business can really grow big without it—no idea whatever its value can succeed without it. Bringing it right down to all of us, consider the advertisers who week after week tell about their products in Power to you, its readers. No force on earth could hold them in its columns issue after issue unless they gave "value for value."

Mr. Woolworth's career must be an inspiration to every red-blooded man living in this land of opportunity. It is a direct challenge to every such man to seize the opportunities around him.

The opportunity of the engineer is to equip his plant so that it attains the highest efficiency. And the products advertised in Power give him the means to do it.

POWER

Vol. 36 NEW YORK, NOVEMBER 5, 1912 No. 19

Finding and Leaving

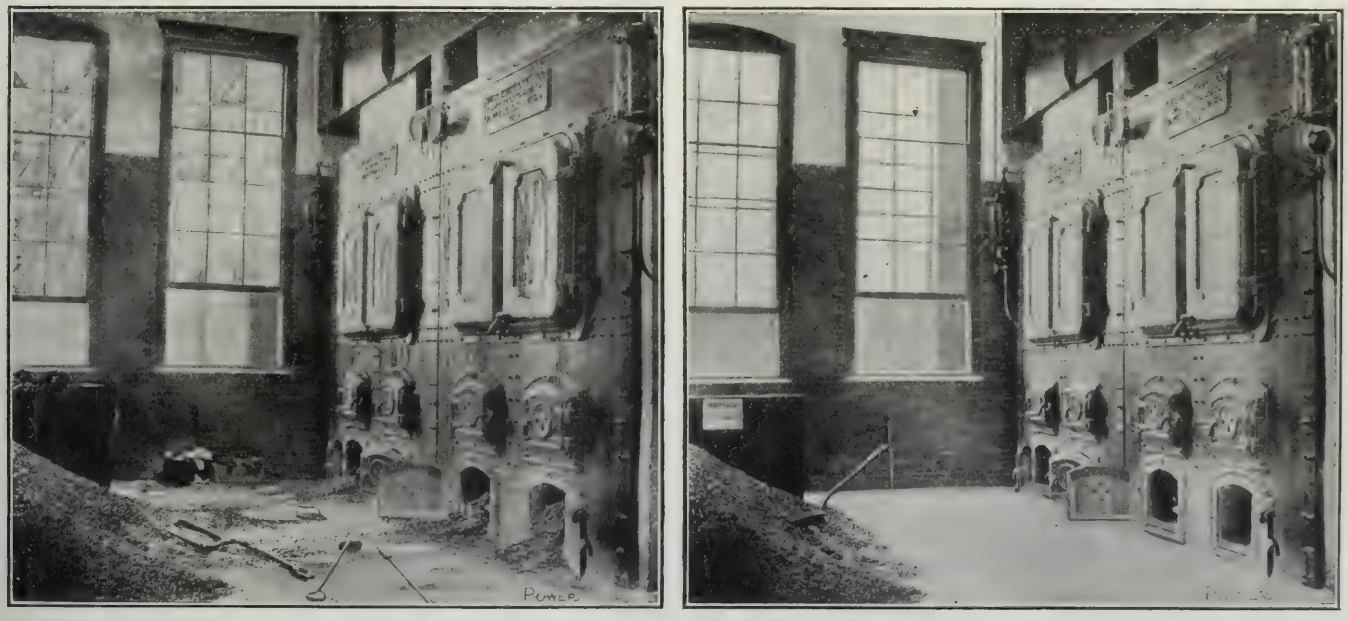
Did you ever say, "If the way I do things don't suit the other fellow, he can do them to suit himself?"

Hardly the expression of a progressive engineer, but one that has been voiced many times in small steam plants, running on a day and night shift. But if a man has done his work the best he knows how, there will not be a feeling that the other fellow will not be satisfied.

In small plants, in which a day and a night engineer are employed, frequent opportunities for friction arise, although it is generally understood that the day man is in charge of the plant. Sometimes he is the more capable of the two, again he is not. Both feel capable of operating the plant and generally each feels that he is the better.

If both pull together, satisfactory operation will result; if one pulls one way and the other another, discord and unsatisfactory results are sure to follow.

A common cause for discord is shirking on the part of one of the engineers. Neither should come on duty and find the fires dirty, the ashpit full of refuse, coal and ashes strewn all over the boiler room floor, boiler fronts dusty and the tubes full of soot. The engine room should also be found clean, the engine wiped, oil cups filled and the tools in their proper places.



The Wrong and the Right Way to Leave the Fire Room.

Many engineers, rather than work with the dirt of other men about, will clean it up, thus doing labor that does not belong to their shift.

But because he does this extra work, do not put him down as an "easy mark." He is the better man of the two and shows his true calibre. He will find his proper place in a larger plant at better pay.

The shirker will, unless he changes his ways, continue to chuckle at his success in forcing another to do his work, and will finish his career in a fourth-rate steam plant, at about $12 per week, disgruntled and disliked.

Losses in the Steam Cylinder

By Prof. R. C. H. Heck

A study of general relations and various steam quantities, calculation of indicated steam consumption, and the construction of the diagram of specific steam rates.

Perhaps the clearest and most comprehensive understanding of the losses occurring in a steam-engine cylinder can be got by considering first the ideal case in which these losses would be absent, then comparing actual with ideal performance.

The essential actions upon or the working medium in any heat-engine plant are, to receive heat from the source, convert part of it into work, and reject the remainder. In the steam plant these are quite distinctly separated, constituting the respective functions of the boiler, of the engine cylinder, and of the condenser (often the atmosphere). Obviously the cylinder will best serve its purpose if it does not encroach upon the purely thermal functions of imparting and abstracting heat.

What takes place in the ideal heat

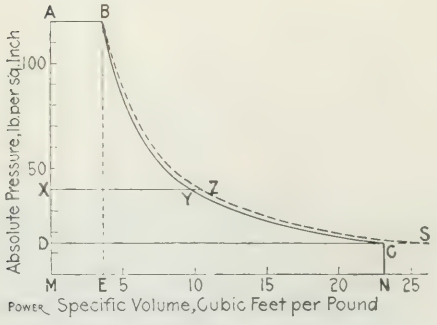


FIG. 1. THE RANKINE CYCLE

engine is represented by the diagram, Fig. 1, generally called the Rankine cycle. In a general discussion, it is convenient and proper to consider the performance of the unit weight of steam, rather than the particular quantity which a certain engine would use per revolution. Fig. 1 is therefore drawn in true proportions for 1 lb. of steam, with the limiting pressures 120 lb. and 15 lb. per sq.in., abs. The cylinder is assumed to have no clearance space and to be thermally neutral, as if made of some imaginary substance which can neither absorb nor conduct heat.

With theoretically perfect conditions, there would be no pipe and valve losses; then the height *MA* represents the pressure and the length *AB* or *ME* the volume of the pound of steam, both when leaving the boiler and when entering the cylinder. As the simplest case, the steam is taken to be dry and saturated at *B*, but the same kind of ideal operation can be reasoned out for steam that is initially either wet or superheated. The heat received from the fire is brought into the cylinder unchanged, neither diminished in amount by escape through the pipe walls, nor impaired in usefulness by pressure drop due to pipe friction or other throttling action. Then at the cutoff point *B* the pound of steam carries its full internal or intrinsic energy, and the area *ABEMA* shows the external work of vaporization, finally carried over to and performed upon the piston. The heat in the steam at *B*, plus the external work *ABEMA*, constitutes the "total heat" of the pound of steam.

Because the cylinder can neither absorb, give off, nor transmit heat, the expansion *BC* must be adiabatic, or the work performance beyond and after *B* must be wholly by virtue of the store of energy received from the outside source (the fire). The work upon the piston, area *BCNEB*, in relation to the heat con-

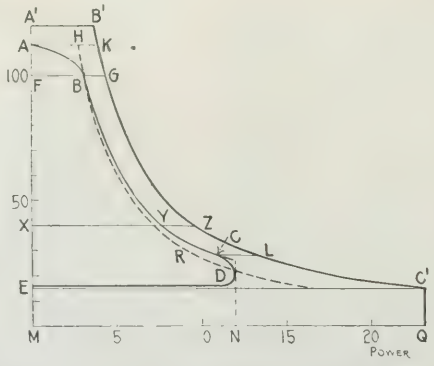


FIG. 2. DIAGRAM SHOWING EFFECT OF PIPE RESISTANCE, INITIAL CONDENSATION AND INCOMPLETE EXPANSION

tent at *B*, determines the shape of the curve *BC*. In giving up the heat equivalent of the external work of expansion, the steam partly condenses; the amount of condensation is indicated by drawing the saturation line *BS*, got by laying off from *MA*, at any pressure, the full volume of the pound of steam. It may be noticed that the expansion is complete, or is continued until the line *DC* of exhaust pressure is reached at *C*. This line *CD* shows complete exhaust, which is necessary under the assumption of no clearance space.

Still governed by this last assumption, the diagram of actual performance takes the form *ABCDEA* in Fig. 2. Pipe radiation is eliminated by considering that all water of condensation due to this cause is removed by a separator at the engine. Pipe resistances lower the pressure of the steam from *A'B'* to *AK*, the work represented by area *A'B'KAA'* being changed back to heat and retained as a part of the total heat of the steam that passes the separator. Initial condensation by the cylinder walls cuts down the volume at cutoff to *FB*, and makes the whole expansion curve *BC* lie well within the ideal curve *B'C'*; but because of return of heat from the cylinder walls to the expanding steam, curve *BC* rises considerably, above an adiabatic *BR*, drawn from *B*. -The total lost area *A'B'C'DC-BAA'* is chargeable partly to pipe resistances, partly to incomplete expansion in the drop from *C* to *D*, chiefly to the thermal action of the cylinder walls in diminishing the effective volume to that limited by the curve *BC*.

The actual indicator diagram and the performance which it represents are further complicated by the existence of clearance space and of compression. In Fig. 3 is shown such a diagram with the saturation curve *SS* drawn for the whole weight of steam present during expansion, including the clearance steam. On the average, the latter may be taken as dry saturated at the beginning of compression at *E;* in some cases it probably contains a little moisture, in others some superheating has been detected. Then the weight of steam which has the volume *KE* must be added to the amount of working steam which comes from the

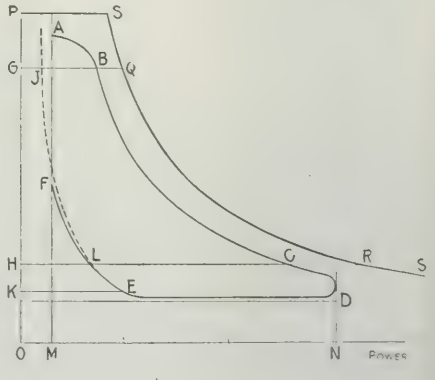


FIG. 3. DIAGRAM SHOWING ACTUAL PERFORMANCE

boiler, to get the total weight for which the curve *SS* is laid out. The question of compression and its effects will later be considered more at length, as one of the main divisions of the subject.

CALCULATION OF INDICATED STEAM CONSUMPTION

Fig. 3 has the pressure and volume data needed for the calculation of indicated steam consumption. The word "steam" is here (and commonly) used to cover the whole mixture of vapor and

liquid; it might be better to speak of the total water present in the cylinder, but that word is rather too closely associated with the liquid state. Consider the illustration to be an ordinary indicator diagram, showing the performance of the particular quantity of steam used by some engine in one cylinder end per revolution; the curve *SS*, possible of location only when a measurement of actual steam consumption has been made, may not be definitely known. The length *MN* represents not only the stroke of the piston but also the volume displaced by the piston in one stroke, or the nominal cylinder volume. If *A* is the piston area in square inches and *L* the stroke in feet, this volume is

$$V = \frac{LA}{144}\ cu.ft. \qquad (1)$$

To whatever scale *MN* represents the volume *V*, in the same unit *GB* will measure the space filled with steam at pressure *OG*, *HC* the space at pressure *OH*, and *KE* the space at pressure *OK*, the clearance *OM* being included in all these volumes of steam present. It is convenient to use the ratios

$$e_1 = \frac{GB}{MN} \quad e_2 = \frac{HC}{MN} \quad c = \frac{KE}{MN} \qquad (2)$$

the symbol *e* referring to the expansion curve, *c* to the compression curve. Considering the cutoff point *B*, and dropping the subscript from *e* for the present, gives the volume out to *B*,

$$V_e = eV \qquad (3)$$

and the volume out to *E*,

$$V_c = cV$$

Now saturated steam has the special property—differing from air or any dry gas—that only the pressure need be known to fix the volume per pound or the weight per cubic foot, both usually given as columns in the steam table. Letting *d* (for density) represent the latter quantity, d_e will be used for the pressure p_e or *OG*, d_c for the pressure p_c or *OK*; then the weight of total steam—that is, working steam plus clearance steam—contained in the cylinder with the piston at *B*, is

$$w_e = e d_c\ V\ lb. \qquad (4)$$

while the weight of clearance steam alone at *E* is

$$w_c = c d_c\ V\ lb. \qquad (5)$$

Evidently, the steam apparently consumed per revolution, or the indicated weight of the working steam by itself, is

$$w = w_e - w_c = (e d_e - c d_c) V = k V \qquad (6)$$

The factor in parentheses, for which the symbol *k* will be used, indicates the weight of net indicated steam per cubic foot of piston displacement. This is clearly apparent if both sides of equation (6) be divided by *V*; for then

$$\frac{w}{V} = k \qquad (7)$$

and w/V is certainly the steam weight per revolution per cubic foot of nominal cylinder volume. In analogy to specific volume (volume of the unit of weight) and specific weight (weight of the unit of volume), the steam rate *k* per unit of displacement will be called the specific steam consumption or steam rate. Upon this conception is based the graphical method of representing steam consumption to be taken up later. By the use of this method actual steam, indicated steam, missing steam, clearance steam, etc., can be equally well diagrammed.

Between the quantity *k* and the corresponding rate in pounds per horsepower-hour there is a very simple relation, which is perfectly well known but for completeness of presentation had better be deduced here. Since *w* in equation (6) is the weight per revolution, it must be multiplied by the revolutions per minute, *N*, to get steam per minute and by 60 to get steam per hour. Then the weight *W* per hour, giving *V* the value in equation (1), is

$$W = \frac{60\ LAN}{144} \times k \qquad (8)$$

Dividing this by the indicated horsepower,

$$H = \frac{PLAN}{33{,}000}$$

gives the steam rate W_H in pounds per horsepower-hour

$$W_H = \frac{13{,}750}{P} \times k \qquad (9)$$

where *P* is the mean effective pressure in pounds per square inch.

Various Steam Quantities

Returning to Fig. 3, the working assumption will be made that the clearance steam is dry saturated at the point *E*, the beginning of compression; then its actual weight is given by the volume *KE* as

$$w_c = c d_c V = k_c V \qquad (10)$$

according to equation (5). From *E* carry up a (dotted) curve of constant steam weight, cutting the line *HC* at *L* and *GB* at *J*; then *HL* and *GJ* represent the same weight w_c.

If the actual consumption of working steam has been determined, whether by the weighing of feed water or of the discharge from a surface condenser, the weight *w* will be laid out as *JQ* or *LR*; or, in general, it will locate the constant-weight curve *SS* for the total steam $(w + w_c)$. Between the indicated working steam w_i given by equation (6) and shown, in changing value, by *JB*, *LC*, etc.—and the actual steam *w* lies the missing quantity w_m, represented by *BQ* or *CR*. This is really present in the form of water, except as the deficiency w_m may be partly due to leakage, but is of such insignificant bulk that it fails to make any impression upon the volume which gives the indicated steam. The subscript *m* may be thought of as standing for "missing" or as related to the moisture fraction *m*. If the shrinkage from *GQ* to *GB* were all due to cylinder condensation,

$$BQ = \frac{w_m}{d} \qquad (11)$$

would measure the moisture in the cylinder at *B*, expressing it as a fraction of the weight of steam admitted.

In the ordinary practical workup of indicator diagrams, which goes no farther than the calculation of indicated horsepower and steam consumption, the missing steam is referred to or compared with the working steam, after the manner of equation (11). It is more logical, however, to let *m* represent the ratio of *BQ* to *GQ*, or of missing steam to total steam, and this is done when the closer thermal analysis of steam action is undertaken. The complement of the moisture fraction as thus defined, or $x = (1 - m)$, is the quality fraction for the whole steam in the cylinder, assuming that none has escaped by leakage.

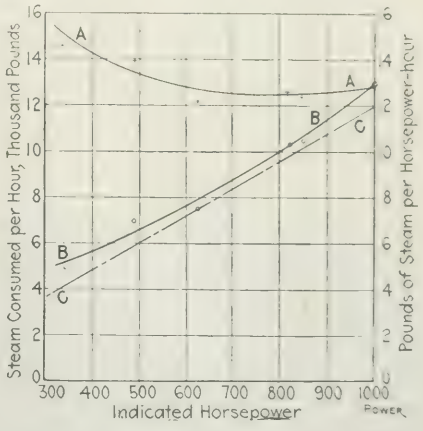


Fig. 4. Curves of Total Steam and Steam Per Horsepower

The missing quantity *BQ* at cutoff is of strong interest; it fixes the effective initial volume of the unit weight of steam, and is quite closely proportional to the amount of heat yielded to the cylinder walls by the steam, or can with little difficulty be adjusted into such proportionality. But as expansion progresses, the volume between curves *BC* and *QR* bears a less and less direct relation to heat interchanged. As a help in comparing realized with ideal performance, the missing quantity *CR* at release is of less value than would be the volume or weight between *C* and an adiabatic curve from *Q*, like *CL* in Fig. 2. The useful result got by calculating indicated steam at release is the difference between the weights corresponding to volumes *CL* and *BJ*, or the increases in steam weight due to reëvaporation.

The Diagram of Specific Steam Rates

In Fig. 4 is given an example of the two curves commonly used for showing

the steam consumption of an engine; these are, the curve *AA* of steam per horsepower-hour and the curve *BB* of total steam used per hour. The tests selected for illustrative purposes were made by Professor Jacobus upon a 20 and 40 by 42-in. cross-compound Corliss engine, and are published in the *Transactions* of the American Society of Mechanical Engineers, Vol. 24, page 1274. The engine had full jackets (on the heads and barrel of each cylinder) and a reheating receiver; the steam pressure was 150 lb. gage and the average pressure in the condenser about 1 lb. abs.; the speed varied from 122.7 to 120.6 r.p.m. in a range of loading from 340 to 1000 i.hp.

The first curve in Fig. 4 shows merely a ratio, the second an absolute quantity. For a representation of the secondary steam quantities and a study of their

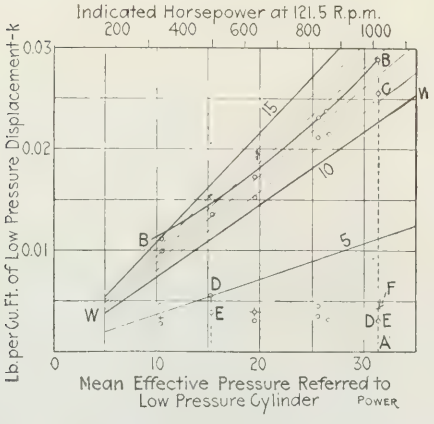


FIG. 5. DIAGRAM OF SPECIFIC STEAM RATES

manner of variation, the latter curve is preferable; and from it is developed the scheme now to be used, exemplified in Fig. 5. In the later diagram, the ordinate is the specific steam consumption k, or the steam per cubic foot of displacement. Division of steam per hour W by the revolutions per hour $60N$ gives the steam w per revolution; further division by nominal volume V gives the specific rate k. In an ordinary double-acting engine this V is the double volume, or the sum of the two forward-stroke displacements by the two piston faces. In analogy to the elimination of the speed and size factors from steam quantity, the base of the diagram is the mean effective pressure, as representing in simplest terms the power developed.

Referring to the lettering along the ordinate for the highest-power test in Fig. 5, the quantities plotted may be described as follows: *AB* is the total steam consumed by the engine, in cylinders and in jackets; *BC* is the steam used in the jackets and reheaters; then *AC* is the cylinder steam k, or the working steam received; *CD* is the indicated steam k_{ic} at cutoff and *CE* the indicated steam k_{ir} at release, by equation (6); normally these are different, the second usually the greater because of reëvaporation, as on the other ordinates. Measuring them downward from *C*, gives the two steam weights most useful in a study of cylinder losses, namely, *AD* or k_{mc}, the missing steam at cutoff, and *DE*, the change in steam weight during expansion. Expressed thus in an absolute measure, rather than by ratios or percentages based on other steam quantities, these values are in decidedly the clearer and more useful form.

Finally, *AF* or k_{mf} is the missing steam at cutoff, according to a formula devised by the writer; it is supposed to agree fairly well with *AD*, serving as a means for estimating the probable value of the latter. In this set of tests, data for the formula are given in but three out of the six tests made; and the degree of conformity of formula to fact leaves much to be desired.

It may be noticed that in Fig. 5 the mean effective pressure is reduced or referred to the low-pressure cylinder; and, of course, the steam weight k is determined by the volume of that same cylinder, even though the missing steam k_m (using this as a general symbol for *AD*, *AE* or *AF*) belongs to the high-pressure cylinder. Not so much the absolute numerical value of k_m as its relative value in the various tests of a series is the matter of interest.

Two further points will complete this introductory description of methods. From the simple relation set forth in equation (9) is derived a very convenient scheme for making Fig. 5 a diagram of steam per horsepower-hour also. For a given rate W_H,

$$k = \frac{P \times W_H}{13{,}750} \qquad (12)$$

which can be represented by an inclined straight line like *WW*, starting from the zero point or origin of coördinates. For instance, take $W_H = 10$; then on the ordinate at $P = 30$,

$$k = \frac{30}{13{,}750} \cdot \frac{1}{10} = 0.0218 \text{ lb.}$$

With this one calculated point and the origin, the line *WW* is definitely located. A series of these constant-rate lines forms a scale on which the values of any of the steam quantities represented can be read off in pounds per horsepower-hour.

In Fig. 4 the line *CC* is a similar constant-rate line for a consumption of 12 lb. per hp.-hr., but the calculation of a determining point is more complicated. As a general proposition, the steam weight w per revolution might be used instead of the weight k per cubic foot; the advantage of the latter is to make engines of various sizes more readily comparable. With machines of the same general class, as to range of limiting pressures and of mean effective pressure, results from a number of engines can be plotted in the same diagram, so as to give a striking composite impression of the behavior of the class.

The last thing to be noticed concerning Fig. 5 is the horsepower scale at the top. It is not always desirable to reproduce tables of numerical results; but to make the diagrams carry as much information as possible, without confusion, will be helpful. If the speed of an engine is constant, the indicated horsepower will vary directly as the mean effective pressure. For the mean speed of the engine, the mean effective pressure which corresponds to a certain indicated horsepower is readily calculable, so that it is a simple matter to get a unit of division and step-off points for the horsepower scale.

New McIntosh & Seymour Horizontal Engine

In designing the new type F engine the manufacturer, the McIntosh & Seymour Co., Auburn, N. Y., has taken advantage of its years of experience and has simplified many details of construction tend-

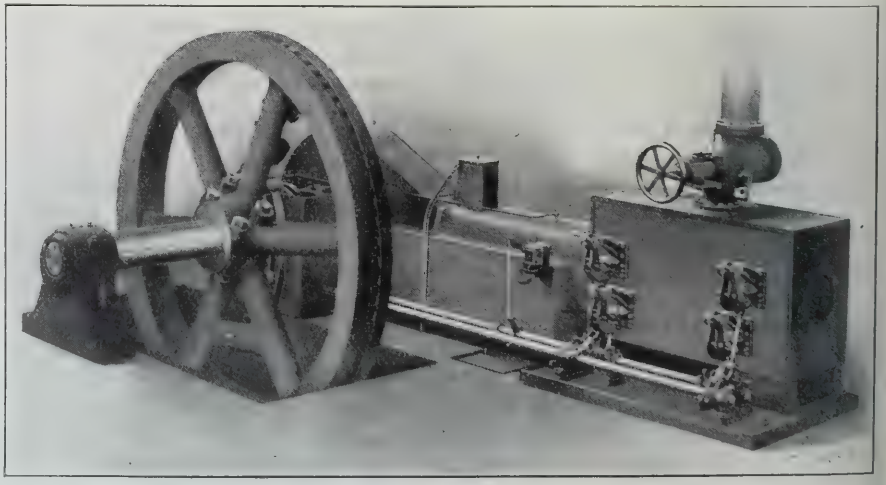


FIG. 1. NEW TYPE F MCINTOSH & SEYMOUR HORIZONTAL ENGINE

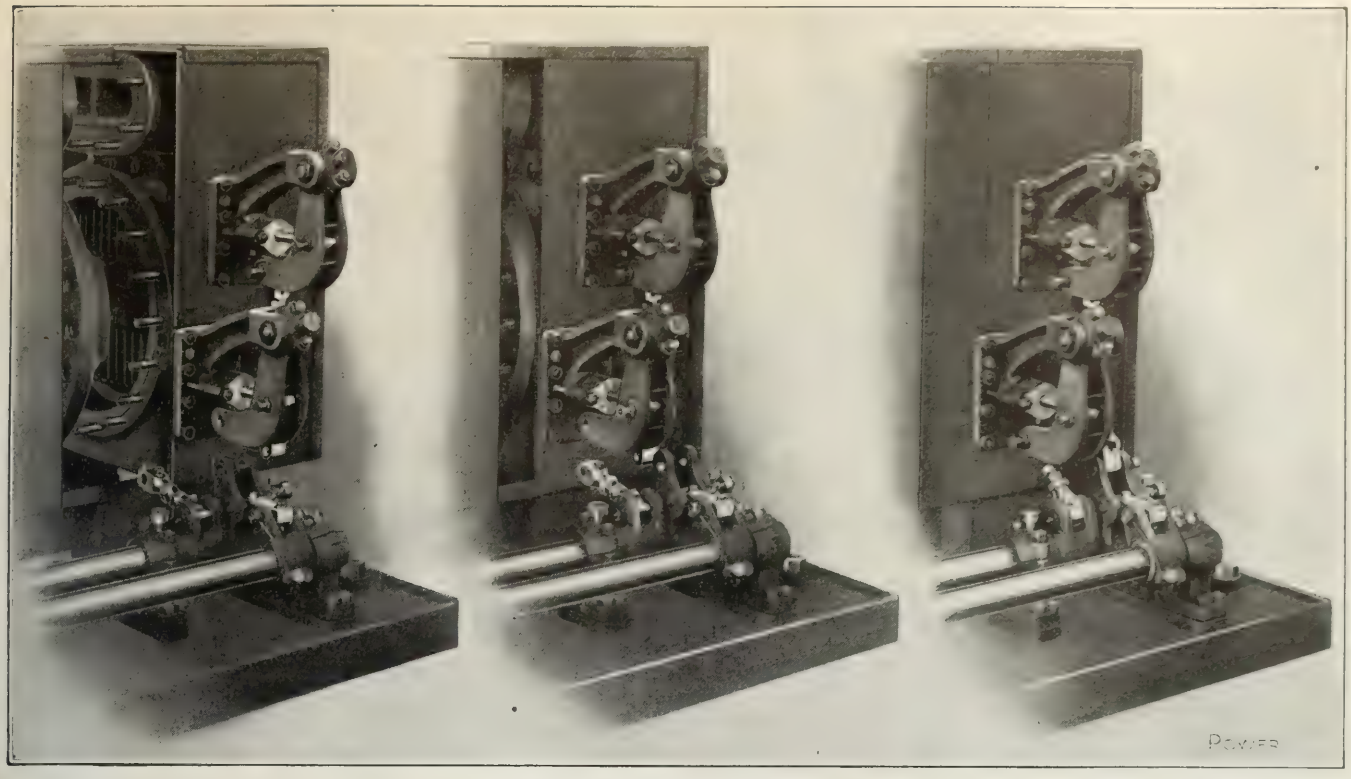


FIG. 2. VALVE GEAR AND REMOVABLE HEAD

ing to increase durability, economy and convenience in handling. The engine is illustrated in Fig. 1.

There are two positively driven steam valves and two exhaust valves, and an exhaust and steam eccentric, the latter controlled by a governor. The steam and exhaust valves are driven by links and bell cranks from rocker-arms on layshafts opposite the cylinder. The special feature of the gear, insuring durability and quiet running, is that the arms or other rocking members, turning upon journals, have means for taking up lost motion.

The four valves are made duplicate and interchangeable as are other parts of the valve-gear, which is arranged to counteract the effect of the angularity of the connecting-rod so as to make the cutoff equal at both ends of the cylinder.

The cylinder casting has a flange at each end through which the cylinder head, containing an exhaust and steam valve, is secured, Fig. 2.

The valve stroke necessary to insure ample opening is small and occurs chiefly when the valve is open and relieved of the pressure of the steam. As there is substantially no movement of the valve when closed the wear is extremely slow, is even and without shoulders since all wearing surfaces on both valves and seat "wipe over." Figs. 3 and 4 show this feature.

Short and direct ports are possible because of the location of the valves in the cylinder heads. The flow of steam is toward the center of the volume displaced by the piston and the form of port is similar to that of a venturi water meter, giving a maximum discharge, a minimum drop of pressure and a minimum final velocity of discharge for a given size of port. This prevents currents of steam impinging upon the walls of the cylinder, thereby causing heat transference and losses from bypassing of heat around the working cycle.

The valves have wipe-over grooves arranged so that steam is admitted at both sides of each port for loads up to nearly full rated capacity. The length of the port edge thus available, is about six times the diameter of the cylinder.

A gridiron valve is used, being peculiarly suitable for placing in the cylinder head, which is removable; practically no more work is necessary than in removing an ordinary cylinder head and its casing.

The quick and easy manner of removing the valve, Fig. 5, affords a means of inspecting, to some extent, the interior of the cylinder by taking out the exhaust valve without disturbing the cylinder head.

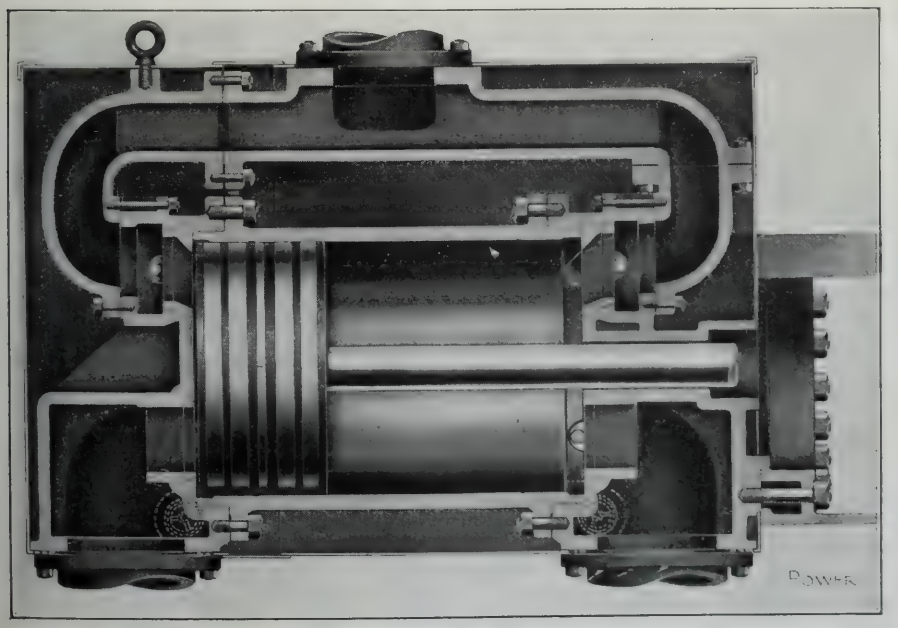


FIG. 3. VERTICAL SECTION THROUGH CYLINDER

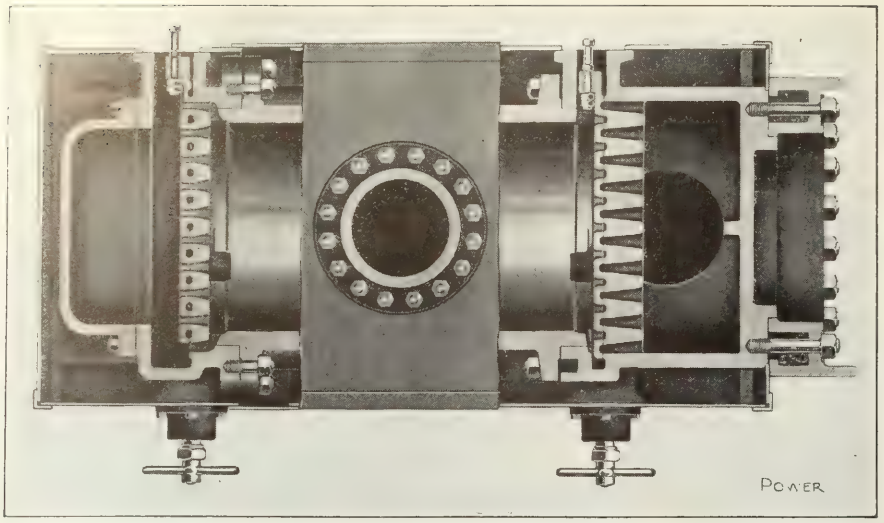


FIG. 4. PARTIAL SECTIONAL VIEW OF CYLINDER SHOWING HEAD-END STEAM AND CRANK-END EXHAUST VALVES

A combination release valve and drip cock is placed in each cylinder head. As the steam valves are free to lift a limited amount, the two features provide security against damage from water in the cylinder.

This engine has a complete oiling system consisting of a receiving tank, water separator and filter, elevating pump to the supply tank above and a series of distributing pipes leading to the sight feeds.

The governor differs from the standard type in that there is but one centrifugal weight which is balanced by the eccentric. The eccentric turns upon a journal ec-

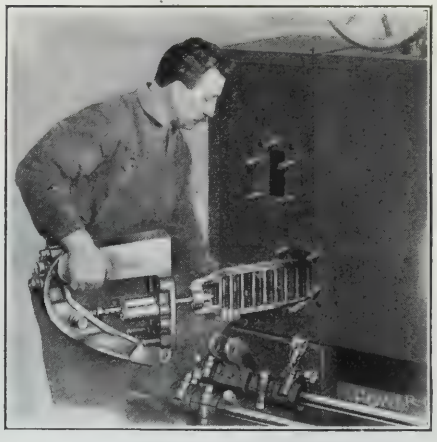


FIG. 5. SHOWING THE EASE WITH WHICH A VALVE CAN BE REMOVED

centric with the shaft instead of turning upon the shaft itself, so that a constant point of admission may be secured to the steam valves with all cutoffs.

The governor is so designed that the combined inertia effect of the different parts due to a change of speed, while it acts in the proper direction to assist the governor, is very slight.

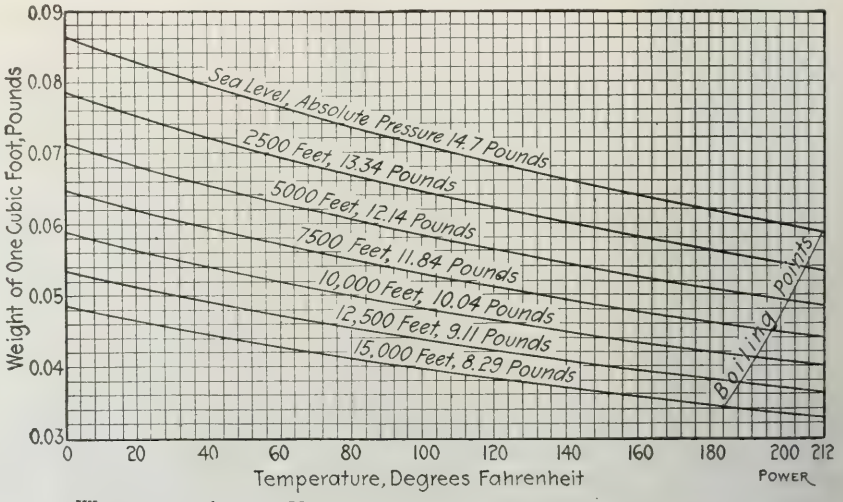
Compressed Air by the Pound

BY FRANK RICHARDS

In statements and computations having to do with the compression and transmission of air the basis or unit of quantity is the volume, usually in cubic feet, of free air, or air at atmospheric pressure, at the time and place under consideration. There are some inconveniences and uncertainties about this practice because the value of the unit is constantly varying with both the altitude, or normal local pressure, and with the temperature. This working volume can always be compared with what would be the volume at sea level and at average temperature, say 60 deg. F., but it is an unwelcome operation.

Knowing the actual quantity of air handled at any time and place would be working with more certainty. If the actual weight of the air is known, it remains the same, whatever the transformations of condition. Pressure, volume and temperature may all vary, but " a pound's a pound for a' that."

To facilitate using the weight basis for records and computations the accompanying table and diagram have been prepared. Computing the items of this table was simple. The weight of a cubic foot of air, whether compressed or not, is obtained by multiplying the constant 2.7093 by the absolute pressure and



WEIGHT OF AIR AT VARIOUS ELEVATIONS AND TEMPERATURES

WEIGHT OF 1 CU.FT. OF FREE AIR AT VARIOUS ALTITUDES AND TEMPERATURES

Temperature Fahr.	Sea level Abs. press. 14.72	2500 ft. Abs. press. 13.34	5000 ft. Abs. press. 12.14	7500 ft. Abs. press. 11.84	10,000 ft. Abs. press. 10.04	12,500 ft. Abs. press. 9.11	15,000 ft. Abs. press. 8.29
0	0.0863	0.0785	0.0713	0.0648	0.0589	0.0535	0.0487
10	0.0845	0.0768	0.0698	0.0635	0.0577	0.0524	0.0477
20	0.0827	0.0752	0.0683	0.0622	0.0565	0.0513	0.0467
30	0.0811	0.0737	0.0669	0.0609	0.0554	0.0503	0.0457
32	0.0807	0.0734	0.0667	0.0606	0.0551	0.0501	0.0455
40	0.0794	0.0722	0.0656	0.0596	0.0542	0.0493	0.0448
50	0.0779	0.0708	0.0643	0.0585	0.0532	0.0483	0.0439
60	0.0764	0.0695	0.0631	0.0574	0.0522	0.0473	0.0431
70	0.0750	0.0682	0.0619	0.0563	0.0512	0.0465	0.0423
80	0.0736	0.0669	0.0608	0.0552	0.0502	0.0456	0.0415
90	0.0723	0.0657	0.0596	0.0542	0.0493	0.0448	0.0408
100	0.0710	0.0645	0.0586	0.0533	0.0484	0.0440	0.0400
110	0.0697	0.0634	0.0578	0.0523	0.0476	0.0432	0.0393
120	0.0685	0.0623	0.0565	0.0514	0.0466	0.0424	0.0386
130	0.0674	0.0612	0.0556	0.0506	0.0459	0.0417	0.0378
140	0.0662	0.0602	0.0549	0.0497	0.0452	0.0410	0.0373
150	0.0651	0.0592	0.0538	0.0489	0.0444	0.0404	0.0367
160	0.0641	0.0583	0.0529	0.0481	0.0438	0.0397	0.0361
170	0.0631	0.0573	0.0521	0.0473	0.0431	0.0391	0.0358
180	0.0621	0.0565	0.0513	0.0466	0.0424	0.0383	0.0348
190	0.0612	0.0556	0.0504	0.0459	0.0417	0.0379	0.0343
200	0.0602	0.0547	0.0497	0.0452	0.0411	0.0373	0.0338
210	0.0593	0.0539	0.0489	0.0445	0.0405	0.0368	0.0332
212	0.0591	0.0538	0.0488	0.0444	0.0404	0.0366	0.0331

dividing the product by the absolute temperature. Thus the weight of 1 cu.ft. of air at an altitude of 10,000 ft. (absolute pressure 10.04 lb.) at 80 deg. F. is

$$\frac{2.7093 \times 10.04}{80 + 461} = 0.0502 \text{ lb.}$$

The diagram embodies all the data comprised in the table, while the curve at the right hand, where it cuts the altitude curves, gives the boiling points of water at those altitudes.

The diagram shows at a glance the differences in weight, or in the actual quantity of air handled, at the different elevations. Thus, to take the extremes, the weight of a cubic foot of free air at sea level and at 60 deg. is 0.0764 lb., while at an elevation of 15,000 ft. the weight of the same volume and at the same temperature is 0.0431, this latter being only 56 per cent. of the former.

In compressing this air to say 100 lb. local gage pressure, the compression at sea level would be

$$\frac{100 + 14.7}{14.7} = 7.8 \text{ atmospheres}$$

while at 15,000 ft. it would be

$$\frac{100 + 8.29}{8.29} = 13.06 \text{ atmospheres}$$

The volume must be reduced to 1/8 in. one case and to 1/13 in the other.

Tom Hunter, Hoisting Engineer

By Warren O. Rogers

Hunter explains how a pump handles mine water from the sump to the surface.

A visit to mine pump rooms shows that the pumps are of large size, having capacities up to 25,000 gallons per minute.

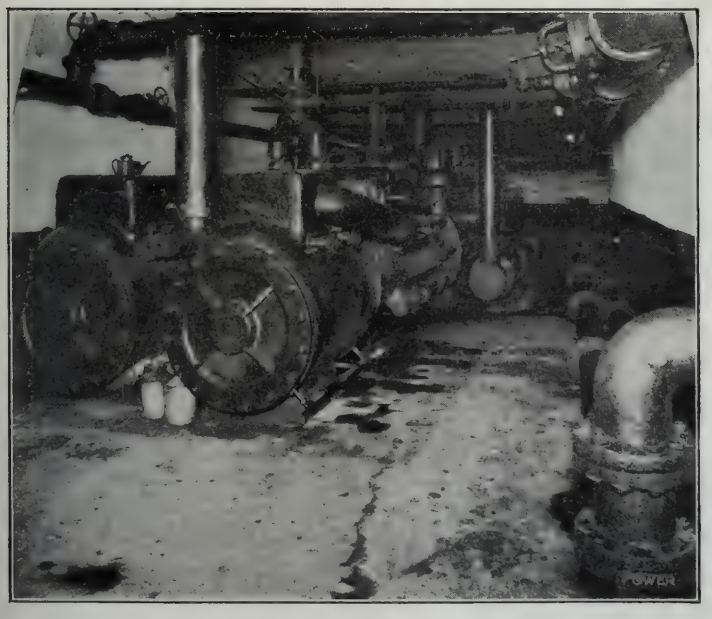
One rainy evening, while seated in the lobby of the hotel, I read an account of a cloudburst that had flooded a mine and drowned several miners. The next day, while out for a stroll, I came to a mine, the superintendent of which stated that the water from the river had broken through the roof, and that it had been in a flooded condition for months. All work had been abandoned in the inner workings, and it would require weeks to pump the mine dry and resume operations.

This subject opened up a field upon which Hunter had said nothing, and one which my talk with the superintendent indicated was of vital importance to mine operation.

It wasn't often that I saw Hunter during the day, but entering a street car that afternoon I chanced upon him. We were bound for the same town, quite a "twofer" as we went back to the smoking compartment. "I was at a flooded mine this morning, and it occurred to me that it is going to require some pumping before all of the water is removed. Now, how do they pump water out of a mine that may be anywhere from 50 to 1000 few miles distant, so there was plenty of opportunity to talk mine pumps.

"What next?" questioned Hunter, when I asked him what he knew about pumping mine water. "I thought you had about pumped me dry regarding mine matters."

"Just begun," I replied, handing him a



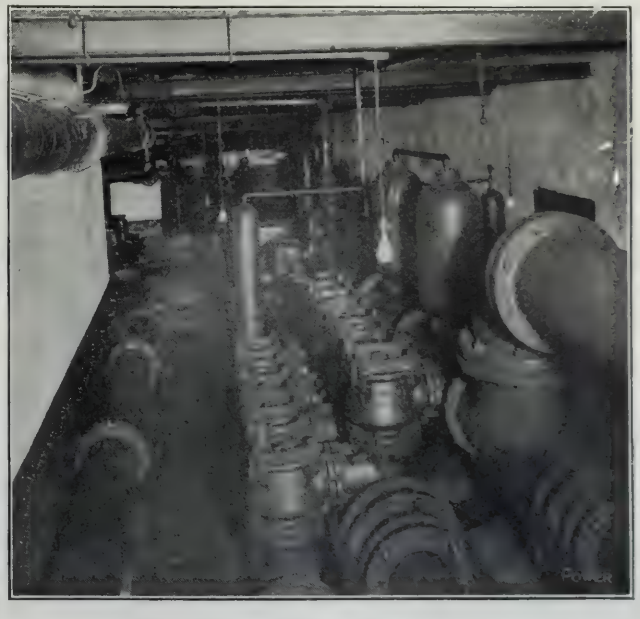


FIG. 1. VIEW OF TWO PUMPS AT THE NO. 9 COLLIERY OF THE PENNSYLVANIA COAL COMPANY

or more feet deep, when a pump will only lift water theoretically about 34 ft.?"

"And is never accomplished," rejoined Hunter. "On the other hand, a duplex pump, placed at or near the bottom of the mine, will easily deliver water in a single lift to the surface, regardless of the depth of the average mine."

"Kind of funny," I replied, chewing the end of my cigar.

"Nothing funny at all. The reason the pump will deliver water to the discharge pipe is because a partial vacuum has been formed in the water cylinder, and the pressure of air on the water supply forces water into the cylinder. If a perfect vacuum were formed by the pump plunger, and no leaks exist, a pump would work with a 33.7-ft. lift. As a perfect vacuum is not obtained a lesser lift of from 22 to 25 ft. is got, with a pump and suction pipe in good condition. A leaky suction pipe will reduce the lift in proportion to the leak, and a serious leak will prevent the pump picking up at all.

"Pumping against a head in a discharge pipe is a different matter. The pump water cylinder is filled with water, and as the piston is forced toward the cylinder head the water escapes through the upper or discharge valves, because there is no other outlet for it, the lower or suction valves being closed by springs as soon as the tension and weight of water is greater than the atmospheric pressure in the suction pipe. With steam

behind the steam piston, the water in the water end will be forced to the top of the mine through the discharge pipe."

"You say that these pumps are placed at or near the bottom of the mine. How can they be used when a mine is flooded? You might as well expect decent service from a waiter without tipping."

Hunter threw the stub end of his cigar out of the window at a stray dog and answered, "Easiest thing in the world. The submerged pumps are out of service, of course, unless air is used in place of steam, but what is known as a sinking pump is used. It is made in a vertical direct-acting type with a steam-actuated valve-gear. It is hung in a vertical position in the shaft, which permits of lowering it as the water recedes in the flooded mine. After the first level has been reached a separate pump can be put to work on that level until the flooded pump has been cleared, when it can be operated to take care of that level of the mine."

"I can see that, but you said that a duplex pump could pump in a single lift, regardless of the depth of the mine. As I remember, a column of water 1 ft. high and 1 sq.in. in area weighes 0.433 lb. Why don't the pump stall when the weight of water in the discharge pipe equals the steam pressure? For instance, if 100 lb. of steam pressure is used in the steam cylinder, why don't the pressure in the discharge pipe equal that in the steam cylinder when the water reaches a height of 231 ft. or a pressure of 100 lb. in the discharge pipe? With a mine 700 ft. deep a steam pressure of over 300 lb. per sq.in. would be required on the steam ends of the pump, which would be some pressure."

Hunter grinned, and pulling a small handbook out of his pocket asked if I had ever noticed the size of pump cylinders.

"You'll find," said he, referring me to the book, "that the water and steam cylinders bear a certain ratio to each other. A simple pump will have a steam cylinder of say 14 in. and a water cylinder of 8 in. in diameter. The first will have an area of 154 sq.in., and the latter a trifle over 50 sq.in. The steam piston would have a total working pressure acting upon it with 100 lb. pressure of 15,400 lb. The water end would have to work against a pressure of 308 lb. per sq.in. before the pressure would equalize, neglecting friction. That would mean pumping against a discharge head of over 711 ft.

"Taking a compound pump with a high-pressure cylinder of 14 in. in diameter, and a water cylinder of 9 in., a 20-in. low-pressure cylinder would probably be added to the steam end, giving an additional steam-piston area of 314 sq.in., which, at a steam pressure of say 50 lb. in the low-pressure cylinder, would mean an added pressure of 15,700 lb. Of course, the steam pressure in actual service is below these figures, the cylinder clearance, condensation, point of cutoff, etc., affecting them."

"Well," I remarked, "your explanation makes the matter plain, but there must be some pretty good-sized pumps to get away with the water from some mines."

"Suppose we take a look into a mine pump room," suggested Hunter, as we neared No. 9 shaft of the Pennsylvania Coal Co. "I think I can get permission to go down."

He did, and we had soon descended to the bottom of the shaft. The mine superintendent's office was visited, and the way it was fitted out was a surprise. The walls, roof and floor were of cement, a neat desk, table, chairs and a rug or two were the furnishings. It was lighted by electricity, and was as sweet and clean as if it had been on the surface, instead of 700 ft. under ground.

After passing through dark, gloomy passages, our way lighted by oil lamps, we finally came to one of the pump rooms, in which were two compound duplex outside-packed plunger pumps, one 16, 30 and 14 by 36 in. in size, the other of the same size but having a 48-in. stroke. (Fig. 1.) Both were run condensing on a 20½-in. vacuum. The condensing water was obtained from the sump and returned to it after passing through the condenser. The walls and floor of the pump room were concrete, and the ceiling natural slate formation.

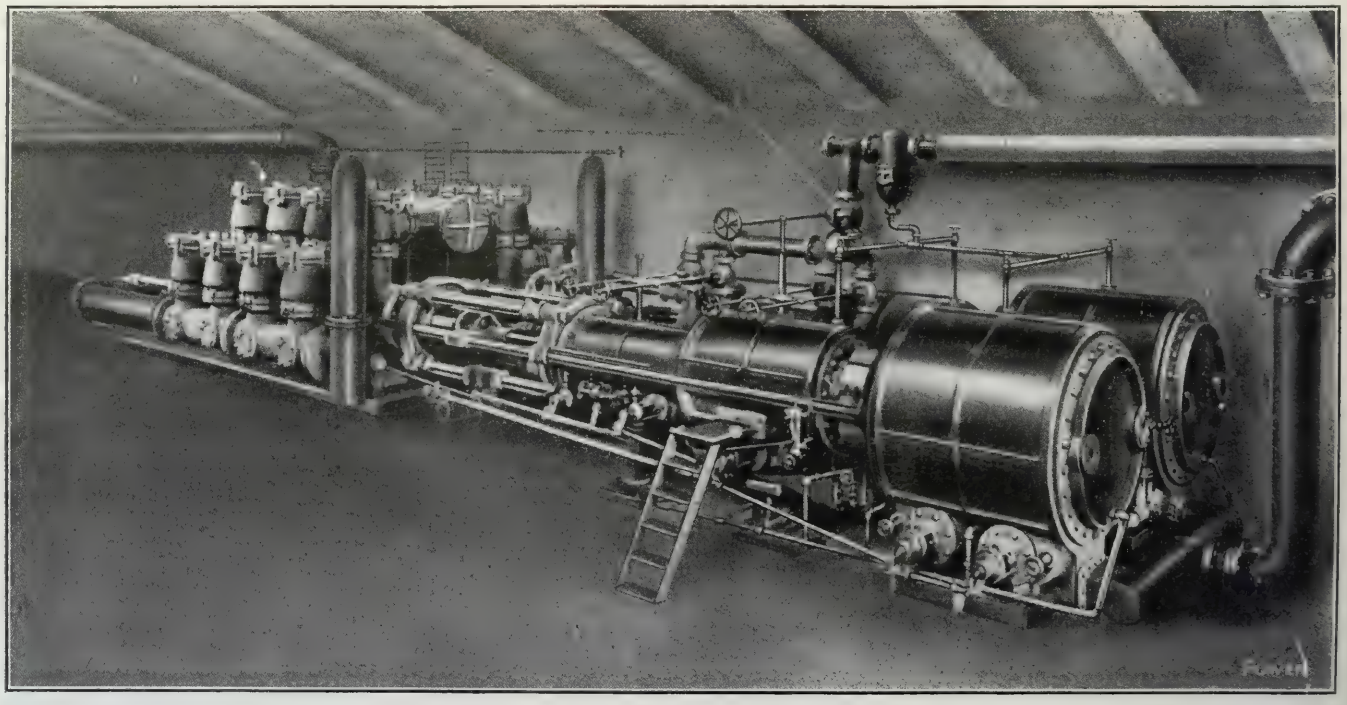


FIG. 2. NEW SCRANTON TRIPLE EXPANSION DUPLEX OUTSIDE END PACKED MINE PLUNGER PUMP

"Where does the steam come from?" I asked, noting the large steam pipe leading into the pump room from a gangway.

"Why, from the boilers, on the surface," answered Hunter. "Although the boilers are sometimes placed underground, near the pump, the general practice is to take steam from a surface boiler plant. The pipes are run down the shaft, and if the pump runs noncondensing, the exhaust steam is carried to the surface, the pipe running in the same shaft. This long exhaust pipe increases the back pressure in the exhaust pipe and

on the pump steam piston. If the exhaust is discharged into the sump, the water is so heated that the temperature of the mine and the humidity are greatly increased, making it difficult to preserve the mine timber. Condensing the steam prevents this trouble, but the heat from the steam pipes cannot be eliminated unless some other power is used to drive the pump."

"What other power could be used?" I asked, as I watched a pump runner fill a lubricator.

"Compressed air is used in a few mines for operating pumps, but as an air-compressing plant is necessary, not more than one in ten mines use it."

"I should think there would be several advantages with the air system, as all leaks would help ventilate the mine, and the heat of the steam pipes would be avoided. Air would cost more than steam, wouldn't it?"

"In large plants, steam would have an advantage. Motor-driven pumps are fast coming into use, however, and electricity is an ideal pump drive. The one drawback is the danger of sparks, in gaseous mines, and the cost of the generating station. But you see, instead of long lines of steam piping, with losses by condensation and back pressure, electric wires, which are easily installed, take their place."

While we were indulging in this conversation we had been making our way to a new pump room where a 16, 25, 42 and 14 by 36-in. standard, duplex, triple-expansion outside-packed pump with pot chamber water ends was being installed. (Fig. 2.)

"How did they ever get this pump down here?" I exclaimed, as I sized up its immensity. "It must weigh 60 or 70 tons."

"It was brought down in parts, one at a time," the foreman of the erecting gang explained. "That pump weighs 85 tons, and has a capacity of 25,000 gal. per min., at a piston speed of 320 ft., and operates against a 500-ft. head."

Seeing that I was interested, he showed me a blueprint of the machine. (Fig. 3.) Continuing, he said, "The water end is completely wood-lined, so that not one particle of cast iron is exposed to the action of the acid mine water. The steam-jacketed cylinder has semi-rotary valves with a special cutoff gear provided with a handwheel so that the point of cutoff can be regulated and adjusted while the pump is in operation. It will run condensing, of course."

"Where is the sump located?" I asked.

"It is not always in the same position in relation to the position of the pump," answered Hunter. "The sump must be placed where conditions in the mine require, which may not be at a convenient point to put the pump. In practice, the pump should not be more than 20 ft. above the bottom of the sump, and the length of the suction should be as short as possible. The main pumping engine is placed as near the shaft as conditions will warrant, so as to reduce the length of piping and the back pressure on the water end."

Split vs. Solid Pulleys

By H. A. Jahnke

A bad mistake made by some engineers, owners and managers of factories and shops is to use solid cast-iron pulleys for all line, jack or countershafting for driving machinery. The advantage of a split pulley over that of a solid pulley is that it can be placed in position without interference with any other part of the shafting equipment, such as removing couplings, hangers and pulleys in order to get the new pulley into place. Especially is the split pulley valuable where an old machine is moved to a new location. Where the pulley driving the machine is of the split type it can easily be removed from one section of shaft and placed on another. A split pulley is also of special value when adding new machinery, because it is not necessary to shut down the plant to cut keyseats in the shaft or to take down a section of shaft to get a pulley in position. The split pulley can be placed on the shaft during the regular noon hour or after closing time in the evening.

Another point in favor of the split pulley is that there are no keyseats or setscrews to mar and spring the shaft; and still another advantage is the ease with which this pulley can be changed to fit the different sizes of shafting through the plant; all that is necessary is to change the bushing for the different diameters of shafting.

Many owners of factories and shops when installing a new machine look over the old pulleys in stock and select one of the required size to drive the machine. The writer had a case of this kind some time ago where a machine was added to the factory equipment and in the stock of old solid cast-iron pulleys, one was found of a size to drive the machine from the line shaft. To get this pulley in position on a section of shaft, a keyseat had to be cut in the shaft and it was decided to do the job on a Sunday. A flange coupling and one solid cast-iron pulley had to be taken off, which made it necessary to take the section of shaft out of its bearings. It took from seven to eight hours to get the coupling and pulley off because it was a very tight fit and rusted on, and it was late in the evening before the job was completed.

It surely would have paid the owner to have left the solid cast-iron pulley in stock or the scrap pile and to have gotten a new split pulley, which could have been placed in position in a short time.

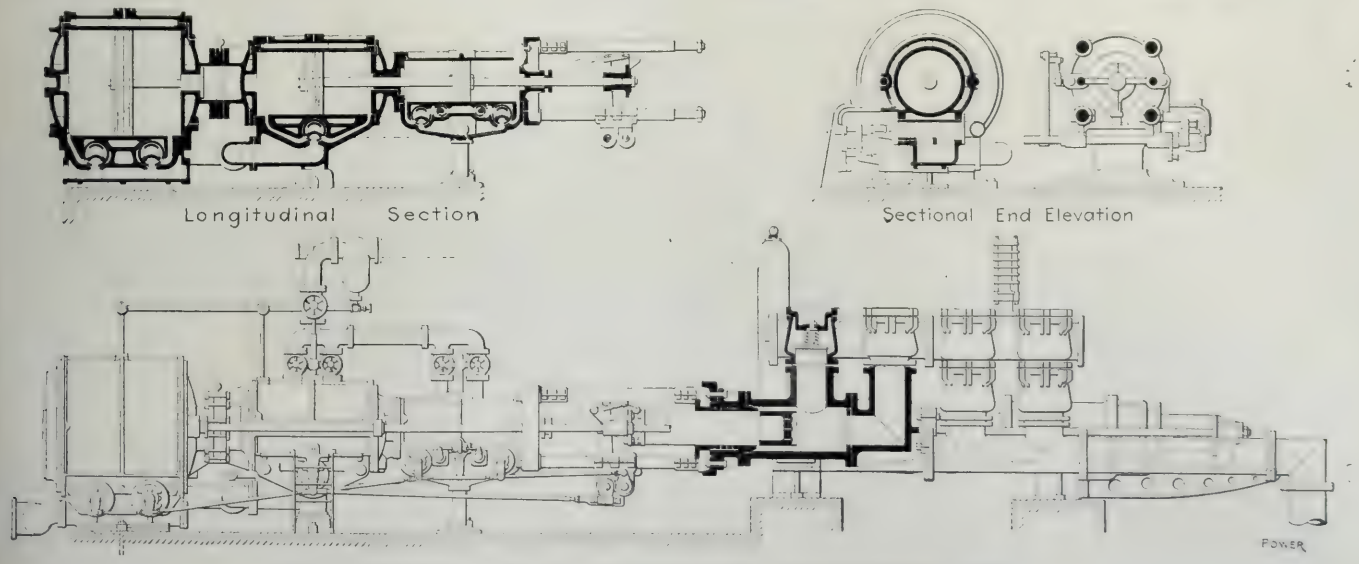


Fig. 3. Sectional Side Elevation of the Triple Expansion Pump

In another instance a machine was moved to a new location. For driving this machine two solid cast-iron pulleys were located on a $2\frac{15}{16}$-in. shaft. To get these pulleys off, this section of shaft had to be disconnected, thereby leaving the other machines which received power from this section idle for several days. Had these two pulleys been of the split type, the machine could have been moved and set up and the two pulleys taken off during the noon hours or after closing time, thus saving the other machines from idleness and loss of time.

Still another instance was that of a solid cast-iron pulley, all the arms of which broke near the hub while running. To get the hub off the shaft in the regular way, a solid cast-iron pulley and one coupling had to be removed. As this would have taken some time, the shafting was shut down long enough to break the rim of the broken pulley to get it off when the shaft was again put in operation. Then a split pulley was ordered of the required size. This pulley was on hand at shutting-down time, when holes were drilled in the hub of the broken pulley with a ratchet, the hub split with a chisel, and the split pulley placed in position. This job was done in a few hours, while if another solid pulley had been used it would have taken at least a day.

Split Pulleys on Countershafting

In most manufacturing plants countershafting with tight and loose pulleys is used for starting and stopping some machines. After the loose pulley has been run for a number of years it becomes worn in the bore of the hub, and the pulley will not run smoothly, causing unnecessary noise. In most cases, to babbitt or bush the hub, the entire countershaft will have to be taken down and the machine shut down for some time, causing a loss of production. There was a case of this kind some time ago where both tight and loose pulleys were of the solid type, and the entire countershaft had to be taken down to get the loose pulley off. Then the pulley was sent to the machine shop to have the bore of the hub bushed. It took several days to get the pulley back again, because the machine shop was rushed with work; all this time the machine had to lie idle.

It is good practice to use pulleys of the split type for a tight and loose pulley system on countershafting or any other shaft where a loose pulley has to be used. They may be of either iron or wood and have the loose pulley provided with an iron sleeve in the bore of the hub. Should this sleeve get worn, the sleeve may be renewed without much labor and expense. If the shaft is worn where the loose pulley has run, the pulleys can be reversed—that is, the tight pulley can be put in the place of the loose pulley.

The writer has made repairs this way without taking down the entire countershaft by first removing the split tight and loose pulleys. Then, should the sleeve in the loose pulley be solid, the hanger and bearing next to the tight or loose pulley is removed, when the shaft is clear at this end and the old sleeve can be taken off and a new sleeve put in its place.

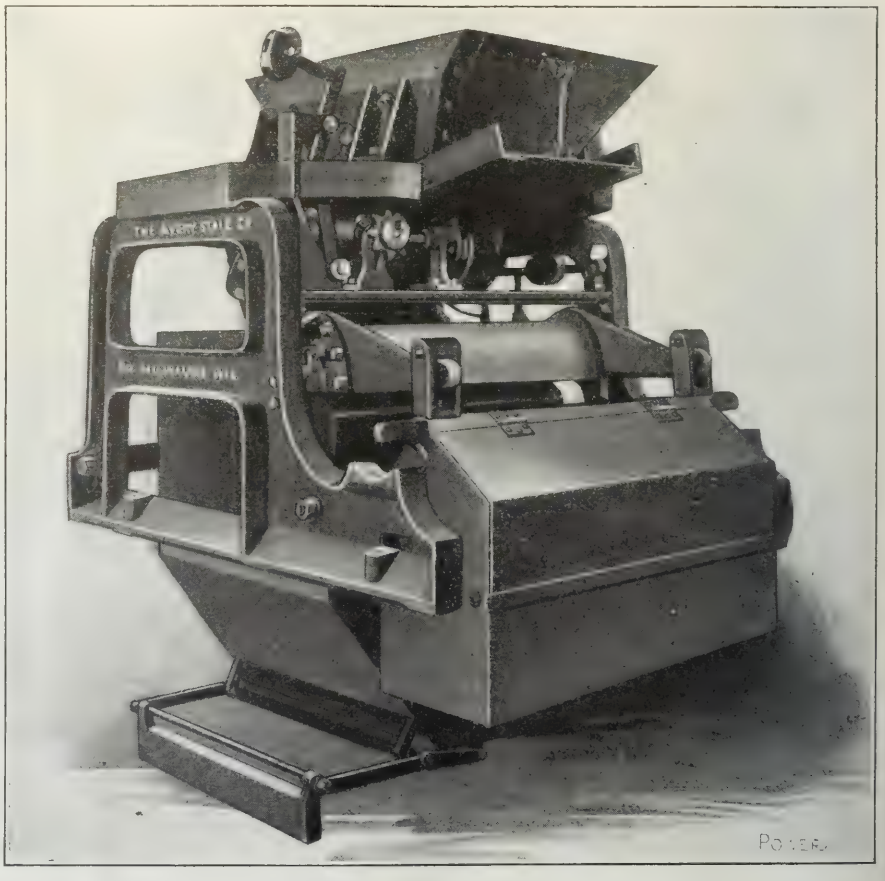
A still better plan is to use a split sleeve for the loose pulley, doing away with taking down any hanger or bearing. Where a split sleeve is used in the loose pulley, repairs can be made on the countershaft without shutting down the machine during working hours. This cannot always be done where solid pulleys are used, as the tight pulley may be a very close fit on the shaft so it will have to be pounded off. If the pulleys are too large to go by some timber or other obstacle, then the only way to get the pulleys off is to take the entire countershaft down.

It is good practice to carry an extra sleeve in stock for all loose pulleys which are split and have an iron sleeve, as an accident is likely to occur at any time to a loose pulley due to the neglect of the machine operator or the regular oiler to oil the pulley properly.

Avery Automatic Coal Scale

One type of automatic coal scale manufactured by the Avery Scale Co., North Milwaukee, Wis., is shown herewith.

This type has a patent soft-coal feeding device and operates on the even-arm beam principle. The beam is double-sided and has six knife edges, two in the center and two at each end. The two center knife edges form the fulcrum of the beam and are supported by steel bearings fixed to the two main frames of the scale. The bottom-discharge weight hopper is suspended from one end of the beam and the weight box, in which standard United States weights are placed, is suspended from the other end. When the weight box and the weight hopper are empty one exactly balances the other. The scale can be tested, without the necessity of checking with another scale, by using known weights.



Avery Automatic Coal Scale

The feeding and the discharging mechanisms are controlled by the weigh beam, although none of their parts is directly connected to the beam which is entirely free to do the weighing at the crucial point. Putting the weights into the weight box tends to depress that end of the beam, but descent is prevented by a projection from the inlet mechanism, the latter being held in a closed position by an ordinary toggle-joint.

To start the machine, it is only necessary to break this toggle-joint when the weights in the box set the inlet mechanism operating. Coal then passes to the weigh hopper, but, before the required amount has descended, the weight of the inlet mechanism is lifted off the weigh beam and supported by a trigger. As soon as the exact weight of coal required is in the hopper that end of the beam descends and pulls aside the trigger which controls the inlet mechanism and by this action the supply of coal is instantly shut off.

Connected to the toggle-joint are two levers which drop when the trigger is pulled aside, and one of them liberates the toggle-joint controlling the weigh hopper, allowing the weighed coal to discharge.

As soon as the contents of the weigh hopper are discharged the balance weight returns the door and the toggle-joint to their original positions. In its travel the projection on one of the toggle-joints hits the same lever that caused the weigh-hopper door to open, which liberates the feeding mechanism so that coal is again delivered to the weigh hopper. While the weigh hopper is filling, the toggle-joint on the hopper is locked, preventing the discharge door from opening and similarly, while the weigh hopper is discharging, the toggle-joint attached to the inlet mechanism prevents coal from being delivered to the hopper.

The feeding mechanism has been especially designed to overcome trouble experienced in getting a machine to handle soft coal containing an appreciable amount of moisture. This feeding device is an oscillating tray moved back and forth by an ordinary crank, and by the shaking motion it automatically clears itself.

The crank is operated by a clutch governed by the weigh beam through the feeding mechanism. The scale is also arranged so that it will automatically regulate itself to the rate at which the coal is taken away. If the machine were feeding mechanical stokers, it would only fill up the spouts which feed the coal to the stokers and then stop working until the coal cleared the outlet door of the weigh hopper.

These machines are made in capacities ranging from 100 to 6000 lb. and are usually made to make one discharge a minute, but the speed can be arranged to accommodate the conditions under which the scale is to work.

Four thousand tons of coal were put into the "Olympic" in 14¼ hours at Southampton recently. This is a record coaling feat for the port and a world's record. Thirty-two gangs of men were employed, an average rate of 288 tons an hour being maintained throughout the night.

An Unusual Boiler Accident*

By Arthur L. Ormay

A rather unusual accident occurred in one of a group of four double-flue boilers (Lancashire type) located in a Viennese plant, the general design of which is shown in the illustration.

The heating surface was 860 sq.ft. and the boiler had a rated working pressure of 180 lb. The boiler was erected and tested in June, 1910, and on Nov. 3, 1911, was examined for corrosion on the shell plate. Shortly before this date, however, the boiler underwent a hydrostatic test previous to applying automatic stokers to the furnaces.

This modification necessitated changing the water gage-glass from a slanting to a vertical position in the center of the front boiler plate.

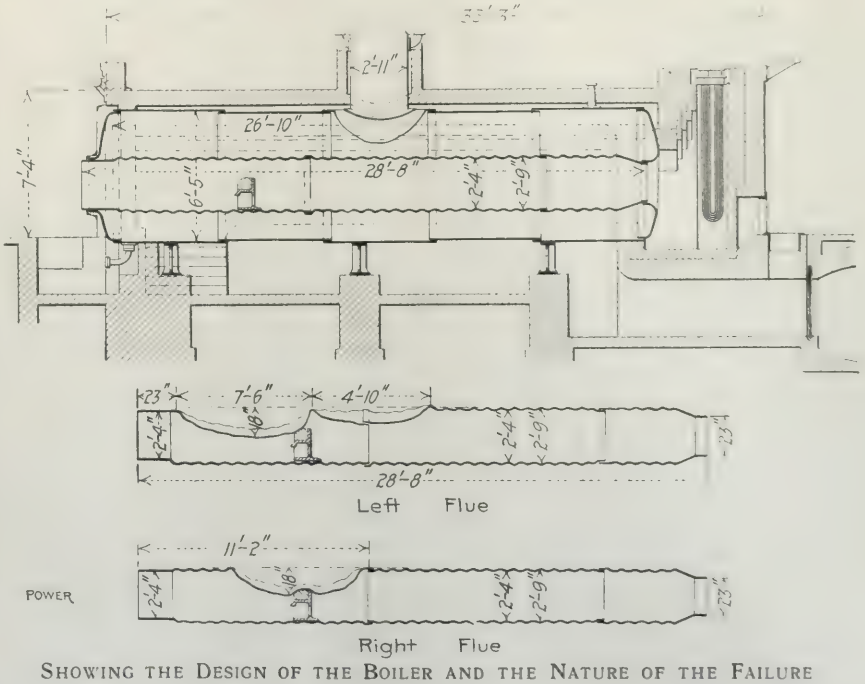
On the afternoon of the day of the accident a fireman noticed steam escaping from the blowoff-pipe pit in front of the boiler, which was covered with an iron plate, and suspected it was due to a leaky gasket in the blowoff line. Later, however, hot water began to run out of the ashpit of the right flue, and a defective flue was suspected. On opening the fire-door the fire was found practically extinguished and the corrugated flue sheet had sagged, as shown. The escape of the hot water soon ceased and steam began to escape from the flue, filling the boiler room, the boiler pressure being 180 lb.

It is believed that, although the water-column connections and feed apparatus were in perfect condition, the boilers had suffered from low water, resulting in the deformation of the flue sheet. The corrugated flues were pressed down to the bridge-wall, causing a sidewise deflection, and in this position the iron bridge-wall support pierced the flue.

Over the upper left edge of the bridge-wall a bulge was formed which showed cracks on the outside flue surface, but not a complete rupture. Similar bulges were noticed in the left flue in the same relative positions, and the bulge was more acute and greater than in the right flue. The middle of the bulge of the right flue coming directly over the bridge-wall led to a greater deformation sidewise which accounts for the rupture of the corrugated flue.

The fireman's negligence in not giving attention to the water level in the boiler was reported to be the cause of the failure.

*Translated from "Zeitschrift der Dampfkesseluntersuchungs- und Versicherungs-Gesellschaft" a. b.



Showing the Design of the Boiler and the Nature of the Failure

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Flexitallic Gasket

This gasket is made of layers of waterproofed asbestos and copperized-steel built up at right angles to the line of pressure, in one continuous length, beginning on the inside and ending on the outside. Thus, the gaskets cannot blow out and, owing to its flexibility, the joints are said to be tight regardless of pressure or conditions.

The gasket is also designed to successfully take care of vibration or expansion and contraction. It can be used over and over again, because it is indestructible and does not stick to flanges. It is suitable for superheated steam.

The gasket is manufactured by the Flexitallic Gasket Co., Real Estate Trust Building, Philadelphia, Penn. It is made in all shapes and sizes for use under various conditions.

Steam Power Plant Piping

By William F. Fischer

The merits and demerits of cast-iron pipes and flanges, and riveted steel, brass and copper pipes are set forth. Several tables dealing with the weight and bursting pressure are also given.

Light-weight cast-iron fittings, up to and including 10 in., are recommended for steam pressures up to 50 lb. and water pressures up to 75 lb.; the larger sizes for steam pressures up to 25 lb., and water pressures up to 35 lb.

Standard weight cast-iron fittings are recommended for steam pressures up to 125 lb., and water pressures up to 160 lb. Extra-heavy cast-iron fittings are recommended for steam pressures up to 250 lb. and water pressures up to 300 lb.

Extra-heavy hydraulic fittings are recommended for water pressures from 800 to 2500 lb., depending upon the size and thickness of the metal used.

Pipe fittings are made with a very high factor of safety as they must withstand not only the working pressure for which they are intended, but also the strains set up in the pipe line by water-hammer, vibration, expansion and contraction, weight of piping, settling, etc.

Good gray cast iron has an average ultimate tensile strength per square inch of from 18,000 to 20,000 lb.; gun iron, 30,000 lb.; ferro-steel and semi-steel, 35,000 lb. This metal is tougher and more uniform in strength than cast-iron, or gun iron.

Steel castings should have an average ultimate tensile strength of 60,000 lb. and a minimum elastic limit of 30,000 lb. per sq.in., an elongation of not less than 20 per cent. in 2 in., and a reduction in area of not less than 30 per cent.

Steel fittings and valves are largely used for superheated steam at high pressures and temperatures.

Malleable-iron fittings have an average ultimate tensile strength of 35,000 lb. per sq.in. Such fittings are made in sizes up to 12 in. and are desirable for general service, especially in screwed type.

Brass castings average about 18,000 to 20,000 lb. per sq.in. ultimate tensile strength, and bronze castings about 30,000 lb.

Table 1 gives the average bursting pressure of standard, cast-iron, flanged pipe fittings, designed for 125 lb. working pressure, and Table 2 that of extra-heavy cast-iron and ferro-steel flanged pipe fittings, all designed for 250 lb. working pressure.

Stop valves used in power-plant work may be had in the straight way or globe types with screwed or flanged ends.

The straight way or gate valve offers, when wide open, a straight, unobstructed passage for the fluid through it. Gate valves are made in straight body and angle types with rising or nonrising stem. The angle gate valve is like the straight-body type, except that the body beyond the gate chamber is cast in the form of a 90-deg. elbow. Gate valves are made with single-piece, solid-tapered wedge disks and tapered seats, with double-tapered wedge disks and tapered seats, or with double, parallel disks and parallel seats.

Owing to the rigidity of the solid-wedge gate or disk valve, the gate must be jammed home after it comes in contact with the tapered seats, to make a tight fit; otherwise the valve will leak under pressure. This jamming action has a tendency to score and cut the seats, especially if dirt or grit lodges on them and is ground into the soft metal. When a valve body becomes distorted or strained by the piping settling or by expansion and contraction strains, which tend to spring the valve seats, it is almost impossible to keep the valve tight without first truing up the seats, which necessitates removing the valve from the line.

*TABLE 1. BURSTING PRESSURE STANDARD CAST IRON FLANGED FITTINGS DESIGNED FOR 125 LBS. WORKING PRESSURE

Size Inches	Thickness of Body in Inches	Average Bursting Pressure in pounds per Per Sq.inch	
		Tees	Elbows
6	9/16	1600	2000
8	5/8	1150	1500
10	11/16	1100	1200
12	13/16	775	900
14	7/8	700	900
16	1	750	850

*Crane Co.

*TABLE 2. BURSTING PRESSURE EXTRA HEAVY FLANGED FITTINGS DESIGNED FOR 250 LBS. WORKING PRESSURE

Size Inches	Thickness of Body in Inches	Average Bursting Pressure in Pounds per Square Inch			
		Extra Heavy Cast Iron		Extra Heavy Ferro-Steel	
		Tees	Elbows	Tees	Elbows
6	1	1687	2275	2733	3266
8	1 1/16	1350	1625	2250	2725
10	1 1/8	1306	1541	2160	2350
12	1	1380	1275	2033	2133
14	1 1/8	1100	1075	1825	1700
16	1 3/16	1025	1250	1700	2100
18	1 1/4	600		1450	
20	1 5/16	750		1275	
24	1 1/2	700		1300	

*Crane Co.

The double-disk gate valve is flexible, as the disks readily adjust themselves to any inequalities or distortion of the seat rings. Should the valve body become distorted and throw the seat rings out of alignment the double disks when closing will rock into position for perfect seating.

Gate valves over 6 in. in diameter should have bypass connections to equalize the pressure on both sides of the gate, and also permit steam to be admitted slowly into the cold end of the line, warming it up gradually before opening the main valve. This prevents water-hammer in the main and tends to distribute the expansion strains more uniformly throughout the piping system.

Globe Valves

A globe valve is divided by an L-shaped partition, which offers considerable resistance to the fluid flow through the valve and over the partition, making two turns at nearly right angles.

The greatest strain on the threaded stem is when screwing the disk down hard against the seat, but nearly always the stem could be twisted off before the threads would fail.

Globe valves in horizontal steam lines should have their stems horizontal to prevent condensation water flooding the line and causing water-hammer. With the valve stem vertical, the condensation will fill the pipe about half full of water before it begins to flow over the valve seat and through the valve.

Globe valves are seldom used as stop valves on water or steam mains because of the excessive friction they create. They are chiefly used as throttle valves, placed near the machines they control. A gate valve is unsuitable as a throttle valve as the disk tends to chatter when partly open; it will not regulate the supply as well as a globe valve, which can be opened full, or closed much quicker.

All screwed fittings, valves, or flanges up to and including 12 in. should be tapped to conform to the Briggs or U. S. standard pipe gage as adopted by all the leading manufacturers of piping materials in the United States. When properly machined, and made up, a screwed pipe joint should be steam- or air-tight against any pressure the pipe will safely withstand.

To avoid difficulty and delay in erection, new fittings and valves to replace old ones should have exactly the same dimensions and drilling as those they are to replace; a point frequently overlooked by the man who places the order.

Flange joints are advisable on all high-pressure lines above 2 in. The first cost for flanged valves and fittings is slightly greater than for screwed ends, but the many advantages warrant the additional cost.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

An Old Motor

By C. V. Hull

A short time ago a large manufacturing concern purchased a number of 220- and 250-volt motors from an electrical power company. These were direct-current motors which had been replaced by alternating-current equipment. Most of the direct-current machines were modern, but among them was a 10-kw. Edison motor built by the General Electric Co. under patents issued in 1882; the serial number is 428.

Fig. 1 shows the switch side and gives a good idea of the construction. This bipolar arrangement is similar to the first Edison dynamo, which was exhibited as a relic at the Buffalo Exposition. The armature is surface wound, and instead of the modern soldered connections, the leads are fastened to the commutator bars by screws. The brush rigging is evidently of later date as carbon brushes

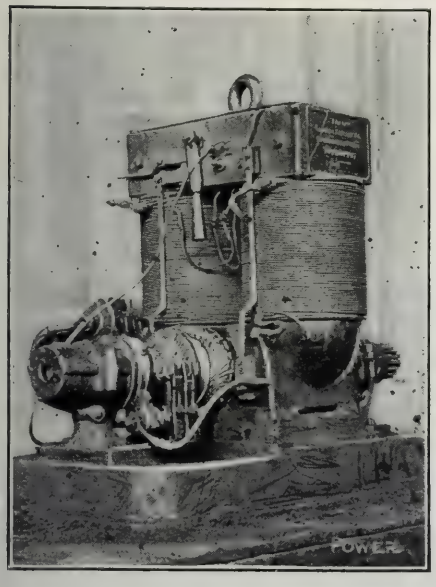


Fig. 1. Old Bipolar Motor

are used, with holders having adjustable flat springs.

The sleeve bearings are of high-grade bronze and each is oiled by two rings. As compared with later 10-kw. motors, the large size of the bearings is noticeable.

The disconnected ends of the series windings are easily seen. These were fitted with clips which were hand-made, judging, at least, by their appearance. The shunt connections may be seen on the side of the upper part of the frame, as well as the corded insulation about the field winding and some of the rubber tubing on the ends of the series winding.

On the side of the frame is the switch used to short-circuit the original starting rheostat, which was a cube-shaped affair with a simple contact handle moving over the contacts arranged in a circle,

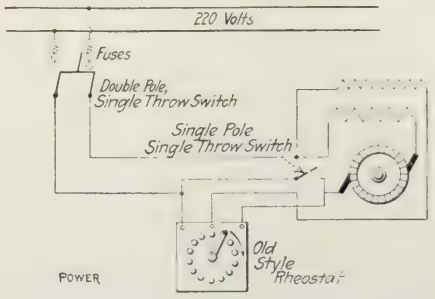


Fig. 2. Line Connections

similar to a field rheostat construction. When starting up, the resistance was cut out by moving the handle of the rheostat around the circular row of contacts. The switch on the side of the motor frame was then closed and the handle of the rheostat returned to its original position for starting; see Fig. 2. In case the operator forgot to bring this handle back, the fuses would blow when the line switch was closed to start the motor again. Or, if the operator failed to pull out the switch on the frame, fuses were liable to be blown when starting again. It is apparent that the whole operation of starting was nonautomatic in every step.

Owing to a shortage of motors when this old machine was received, it was used for some time on a hoist, raising a heavy weight for breaking scrap iron. A modern magnetic release rheostat was used. The series windings were not connected as the motor ran well with only the shunt field in service.

A few months ago, however, the motor was taken out of service because the field was so strong that pieces of iron, broken off by the blows of the weight, sifted through cracks in the roof of the motor shelter, and were drawn into the space between the pole pieces and the armature. These chips cut off so many wires on the armature that another motor of modern construction was installed in its place.

Selecting Faulty Cables

At the recent convention of the American Electric Railway Engineering Association, the committee on power generation submitted a report upon "Methods of Selecting Faulty High-Tension Cables at the Power House."

To ascertain present practice in this line the committee first sent letters to a number of companies operating large plants and the replies indicated that many had not taken advantage of developments in the art and therefore employed no satisfactory method of selection. Others employed methods satisfactory to their own local conditions, but the replies in general indicated the need for further development.

The report goes on to say that electrostatic ground detectors will often indicate a weakening of the insulation before the current increases to a destructive value, but they do not indicate nor select the faulty cable and their use is limited to systems operated without the grounded neutral.

Torchio-Varley System

The Torchio-Varley system, based on the relative phase insulation resistance to ground, will indicate the feeder at fault, if the feeders are not in parallel, in which case it will indicate only the group to which the faulty cable belongs. Moreover, this system can be applied only to underground cables and cannot be used with a grounded neutral. It has been successful on high-tension underground systems of 6600 volts where the high-tension feeders are not operated in parallel.

Relays

Overload relays are in general use for both overhead and underground feeders. Some of these when properly set are fairly successful in selecting a faulty cable where a ground exists with the neutral of the system grounded, but where there is a "short" between phases, these relays are often unsatisfactory and sometimes cannot be depended on for actual selection.

Where feeders are operated in parallel, either with or without grounded neutral, it would seem that the faulty cable could be selected by means of inverse time element overload relays, having characteristics between the ordinary inverse time limit and definite time-limit relays.

on the generator end of the cables and satisfactory reverse-current relays on the receiving end. Alternating-current reverse-current relays have in general been unsatisfactory, being inoperative on low power factor loads, and very likely to trip the breaker on direct overloads. Reverse-current relays to be satisfactory should operate practically instantaneously on a reversal of power only, not on overload, and should not operate on slight reversals, such as might be occasioned by throwing a rotary converter on the line when slightly out of phase.

If such a reverse-current relay did not actually select the faulty cable it should at least indicate that it has operated on a reversal of power.

Mertz-Price System

The Mertz-Price system and its modifications are based on balancing the currents at the two ends of the apparatus to be protected, and the system as developed cannot be used without pilot wires—two wires per cable for magnetic balance and one wire per phase per cable for balancing the electromotive force. On a large system the cost of pilot wires is excessive.

This scheme, however, will select and isolate the faulty cable under practically all conditions of service. It is especially suited to the selection of faulty high-tension tie-line cables between two generating stations or between two substations, each supplied directly from a common source. This system has been in use in England and on the Continent for more than two years.

Mr. Stott's Suggestion

H. G. Stott offered a suggestion for overcoming these difficulties which appears to be quick and at moderate cost applicable to practically all operating conditions. His scheme is to install, in the main station having double busbars, a three-phase transformer with a 1-to-1 ratio having high resistance and poor regulation and rated somewhat higher than the charging capacity of the longest cable of the system; under these conditions the short-circuited current would be only slightly in excess of its capacity.

To select the cable at fault, when all or some of the switches have gone out, it is necessary only to connect one end of this transformer to the high-tension busbar in service and then connect, one at a time, the doubtful cables to the high-tension busbar which is energized through this transformer. This applies full voltage to all phases of the cable simultaneously and its condition will be indicated by ammeters connected in the circuit. Each cable after being tested and found satisfactory can be connected to the busbar supplying normal service. Such a test can be applied by the switchboard operator without leaving his station and requires but a few minutes to complete with a minimum service interruption.

Largest Direct Current Generators

The new plant of the Southern Aluminum Co., now under construction, at Whitney, N. C., will contain seven vertical waterwheel-type electric generators, representing units of the largest capacity ever built for generating direct current. They will be of G. E. make, and each machine will have a rating of 5000 kw., delivering 20,000 amperes at 250 volts and operating at a speed of 170 r.p.m. Two smaller direct-current generators of the same type, rated at 2500 kw. when running at 300 r.p.m., and two 1250-kv.-a. alternators, having a speed of 514 r.p.m., will be included in the equipment.

The machines will be directly connected to vertical shafts of S. Morgan Smith turbines by forged-steel flanged couplings. Each 5000-kw. generator will weigh about 150 tons, measure 22 ft. in diameter and extend 13 ft. above the floor level. The entire rotating element of the generators will be supported from an overhead thrust bearing. Although the normal speed will be 170 r.p.m., the generators will be designed with provision for a safety runaway speed of 75 per cent. above normal. The wheel governors will have remote electric control for both hand and automatic operation.

CORRESPONDENCE

Blown Fuse

In a large department store where a refrigeration system is employed, there is a 5-hp., 550-volt, three-phase motor belted to a centrifugal pump for the brine circulation. One morning the motor refused to start, but when the belt was taken off and the motor turned by hand it got up to speed, but would not carry the load; moreover, the solenoid on the starting box for the automatic cutout was dead. The electrician was directed to take the starting box apart, but after working 5 hr. the trouble still remained. The engineer then blamed the motor, but this was new.

Finally it was suggested that the voltage might not be the same on all three phases. The illustration shows the transformer connections to the motor. The voltage from *A* to *B* measured 550 volts, from *B* to *C*, 275 and not any from *A* to *C*. One of the primary fuses was blown.

Can any readers tell which fuse blew on the primary circuit?

J. G. Koppel.

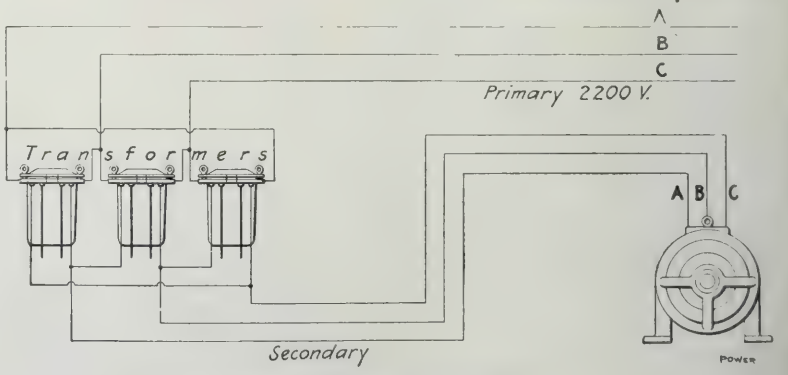
Montreal, Que., Canada.

Neutral Brush Position on Interpole Machines

The method of locating the neutral brush position described by J. S. Kinney in the Oct. 15 issue is correct, and will give the desired result provided residual magnetism in the main magnet cores can be kept out of existence during the test. Unless the entire magnet is of laminated construction, however, this is difficult to do and entails a tedious repetition of "killings."

The neutral position can be located more quickly and with equal certainty by the following procedure:

Drive the armature at full speed and excite the main field winding from some separate source until a moderate voltage —say one-fourth the normal voltage of the machine—is obtained at the brushes, with the interpole winding dead. Then excite the interpole winding fully, from a separate source, and note whether the voltage at the brushes increases or decreases. If it does either, the brushes should be adjusted to a point where closing the circuit of the interpole winding makes no difference in the brush voltage. The speed must be kept the same when the interpoles are active as when they are dead, and if this cannot be done, then correction must be made in the voltage readings by dividing the voltage by the speed each time the field excitation is changed. The quotient of volts divided by revolutions per minute must be the same with and without interpole excitation.



Transformer and Motor Connections

Cecil P. Poole.

Atlanta, Ga.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

The Junkers Engine—III

By F. E. Junge

Conditions Influencing Operation

To form an opinion of the Junkers engine it is essential to discuss those features of operation and construction, upon which the merits of large oil engines are preëminently dependent. These are: conditions influencing the cycle of operation, action of forces, frame and support, power-transmitting mechanism, reliability, overhauling, erection, dismantling and general economic aspects. In the following discussion in which the views of Professor Junkers are given in support of his claims, the normal single-acting Diesel two-stroke-cycle type of engine has been referred to for comparison.

Combustion Chamber

In Fig. 13 the Junkers engine is compared with a single-acting two-stroke-cycle valve engine, the comparative simplicity of the combustion space of the former being apparent. A better idea of the complexity of the cylinder castings for Diesel engines of the double-acting two-stroke-cycle type, which must be used for large work, is conveyed by Figs. 14 to 16, of which Fig. 14 is a section through the combustion space of a Junkers engine, Fig. 15 a section through the upper cylinder cover, and Fig. 16 the lower dead space of a Diesel engine.

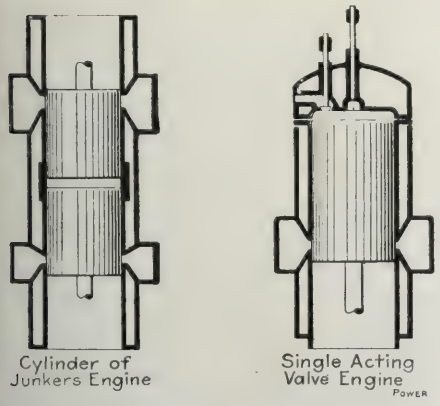


Fig. 13. Junkers Engine Compared with Single-acting Valve Engine

The extreme simplicity of the Junkers engine, as shown, admits of the complete control of combustion with large diameters, and has a favorable bearing on the heat transmission. A result of this is the possibility of running the engine at slow speeds, thus insuring flexibility.

In contrast to the Junkers engine the combustion chamber of the Diesel engine is most intricate, as can be inferred from the illustration. In this case the control over combustion becomes difficult with large diameters, the heat transmission is influenced to an unfavorable extent, and permanent control over the high temperatures with large diameters is out of question. As a result of this the fracture of cylinder heads is not an unusual occurrence, and slow running is unsatisfactory in large Diesel engines.

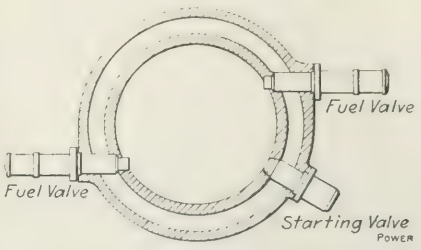


Fig. 14. Section through Combustion Space of Junkers Engine

Heat Transmission to Cylinder Walls

The work lost by the communication of heat to the cylinder walls does not so much depend upon the aggregate value of this heat dissipation, but rather on the nature of its distribution over the stroke of the piston. For the heat lost during combustion and during the early part of the expansion stroke a corresponding amount of power becomes unavailable for useful work. The heat lost toward the end of the expansion stroke, however, has already expended its expansive power available for performing useful work; hence does not impair the fuel consumption. As far as the utilization of the initial heat within the cylinder is concerned, it is immaterial whether the last quantity is communicated to the cylinder walls or leaves with the exhaust. Therefore, it is necessary to keep the ratio of area to volume of combustion space as small as possible. This is necessary also for reasons quite apart from the foregoing, since during combustion the dissipation of heat is greatest, the pressure and temperature attaining their highest values.

The most practical way of procuring a small area of the combustion chamber is to decide upon a large ratio of piston stroke to diameter of piston. In the Junkers engine such a ratio is the natural result of its peculiar arrangement. It therefore differs from other engines in possessing the smallest heat dissipation during combustion and in this respect is thermally superior to others. The practical tests confirm this theoretical consideration. Owing to the favorable configuration of the surface of the combustion chamber the final compression temperature required for the ignition of the fuel is positively attained at a comparatively low piston speed. Therefore the speed of the Junkers engine can be considerably reduced without fear of a failing ignition.

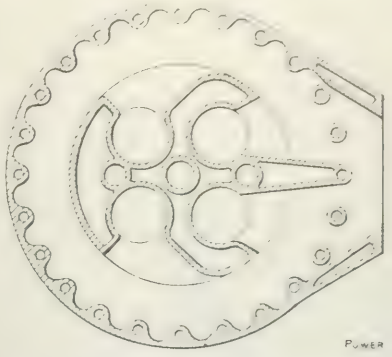


Fig. 15. Section through Upper Cylinder Cover of Diesel Engine

Scavenging Features

The success of a two-stroke-cycle engine is largely dependent upon its scavenging features. Complete scavenging (cool and pure charge) is not only a desideratum with this type, but its realization is of vital importance; for on the scavenging depend the absolute command of positive ignition, an economical,

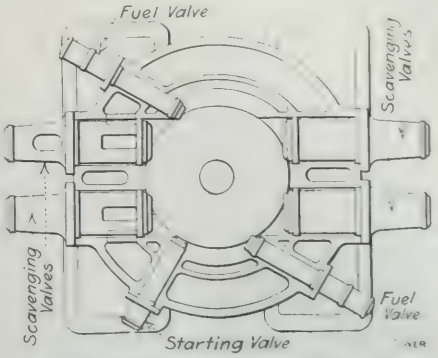


Fig. 16. Section through Lower Dead Space of a Double-acting Two-stroke-cycle Diesel Engine

soot-free combustion and the output of the engine. Difficulty in attaining perfect scavenging in the two-stroke-cycle engine is due to the fact that the events for which two full strokes are provided in the four-stroke-cycle engine, namely, the exhaust and the suction strokes have to be effected during part of one stroke. If, notwithstanding the short time available a pure and cold charge is to be procured the following conditions must be fulfilled: A long cylinder of as small diameter as possible; a perfectly smooth scavenging space devoid of all reëntering or protruding parts, which are liable to retain burnt gases; exhaust and scavenging ports of ample size; precise and positive governing and rapid opening of the exhaust and the scavenging ports; and ports at opposite ends of the cylinder calls for the use of two pistons as governing elements. This disposition is by no means incidental, but is the outcome of systematic experimental research, extending over many years. The line of thought underlying these researches is shortly characterized in the following:

1. Complete expulsion of the products of combustion at the end of the power stroke is essential to the life of every engine of this type, and dependent upon this is the realization of positive ignition, of a perfect, soot-free combustion, of low-grade fuels, and of a high mean indicated pressure.

2. To insure rapid exit of the waste gases with a small consumption of energy during the short time available, large exhaust-slot areas must be provided. Furthermore, in order that these large slot areas may be precisely and rapidly uncovered and covered, gas-tight, close-fitting, and rapidly acting governing elements are necessary. The best way to attain large port areas is, of course, to arrange the port openings along the whole circumference of the cylinder, and the governing element satisfying these conditions in the most perfect manner is, obviously, the piston itself.

3. The same considerations apply to the admittance of scavenging air as to the expulsion of the exhaust gases. Hence, the arrangement of the ports and the choice of the governing elements is the same, essentially, as in the first case, namely, admission slots disposed over the whole circumference of the cylinder and the use of the piston for governing.

4. To avoid a mixing of the scavenging air and the burnt gases both must be conducted in the same direction of flow, that is, the exhaust ports must be at one end, and the inlet ports at the other.

5. The additional features, such as the simple and reliable combustion chamber of relatively small heat-transmitting surface, the high aggregate piston speed, the balance of reciprocating parts and the absence of all delicate parts exposed to high temperatures, all guarantee a working efficiency far superior than the seemingly complicated method of inter-connecting the pistons appear to guarantee.

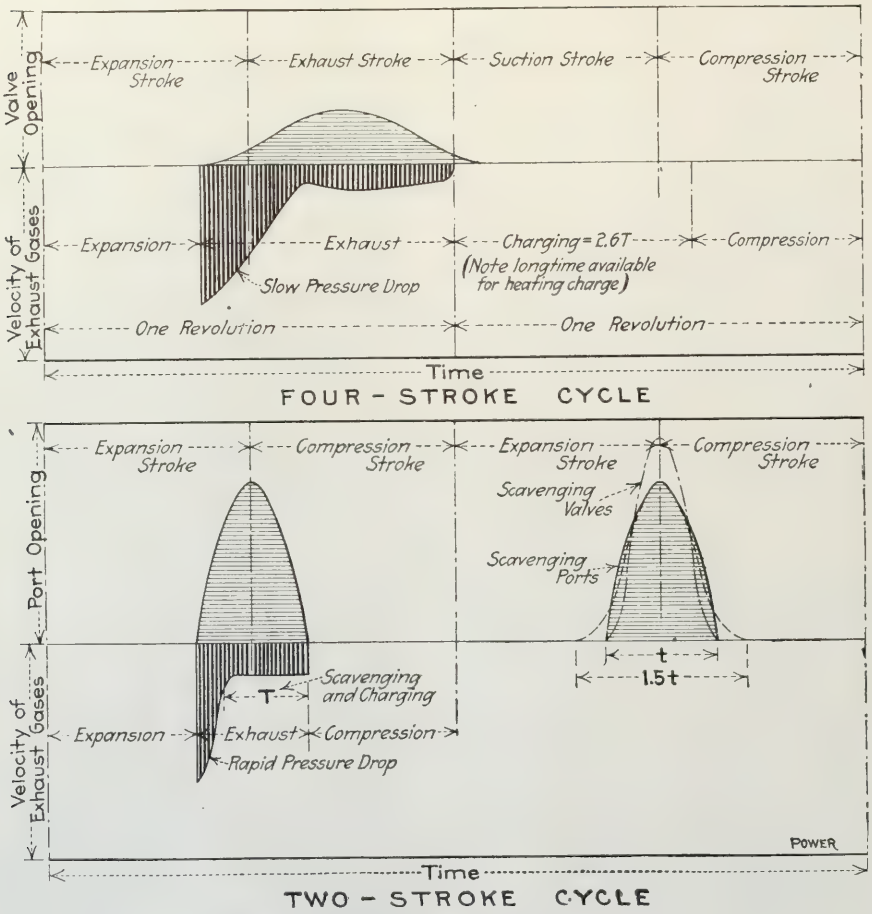


FIG. 5. COMPARISON OF JUNKERS TWO-STROKE-CYCLE ENGINE WITH A FOUR-STROKE-CYCLE ENGINE

Another point to be considered is that the velocity of the air has a great influence on the effectiveness of scavenging. The velocity conditions during scavenging of two- and four-stroke-cycle engines have often been the subject of comparison, and, to some extent, the unfavorable criticism advanced against the two-cycle engine has been justified. The diagram shown in Fig. 5, in which the Junkers engine is graphically compared with a four-stroke-cycle engine of equal cylinder volume, shows that in the Junkers engine these conditions are decidedly favorable. The net port openings of the valves are plotted as ordinates against the time elapsing as scavenging progresses. Below the base line, but in the same direction, are plotted the corresponding velocities of the exhaust gases. Here the difference of time, and, therefore, of pressure drop available for the escape of the burnt gases is plainly shown. It requires much more time in the valve-governed engine and, moreover, is responsible for much greater throttling and wire-drawing losses in the valve seats.

The unfavorable conditions resulting from the necessity of accelerating and retarding the valves used as governing elements are also existent in the two-stroke-cycle valve engine. The diagram also shows the necessary time of opening and the net opening area required by the two-stroke-cycle valve engine, if the same favorable conditions of operation are to be attained which are inherent in the Junkers engine. The requirements are either 1.5 times the time with equal area, or 1.3 times the area for equal times.

PREHEATING OF AIR BY SCAVENGING VALVES

It is often maintained that the scavenging valves of valve-governed oil engines present no difficulty in actual running service. This opinion is based upon the supposed cooling of the valves by the scavenging air. Should this cooling actually take place, there could only be an

omission of valves or other inlet devices for admitting scavenging air in the combustion chamber. All of these conditions are met by the Junkers engine.

At the end of the expansion stroke when the pistons have about reached the outer dead-center position, the products of combustion leave by a row of ports at the one end, extending over the whole circumference of the cylinder. Simultaneously the scavenging air enters through corresponding ports at the other end, expelling the waste gases in the direction of their flow. The characteristic arrangement of the exhaust and the scavenging

illusory advantage attending it. For, taking it for granted that the scavenging air cools the valves, a heating of the air becomes inevitable. This, however, would not be keeping within the fundamental conditions to be met in oil engines, namely, employing a pure and cold charge.

On the other hand, the scavenging air can only enter without being heated, if it does not cool the scavenging valves. In this case the latter are exposed to the deteriorating effects of intense heat. One of the above contentions must surely be correct. Probably what actually happens is that the scavenging valves are not cooled by the scavenging air to an extent that would conform with absolute reliability of operation, whereas the air is heated to a degree making it impossible for a cold charge to be attained. Assuming that only 5 per cent. of the total heat of combustion is imparted to the air charge, the latter will sustain a rise of temperature of from 80 to 100 deg. C. This would correspond to a diminution in the weight of the charge of from 20 to 25 per cent. The mean pressure of the working diagram will show a corresponding decline, while the temperatures and pressures during combustion will not be lower.

When it is considered that large oil engines of the double-acting two-stroke-cycle type, in addition to the two stuffing-boxes, have from four to six valves in each cylirder head, making eight or twelve valves per cylinder—all of which, except the starting valves, are constantly operating under severe temperature and pressure conditions; and that where used in marine work, repairs on shipboard are difficult to execute, it must be admitted that the simplicity of the Junkers cylinder is not too dearly bought by mechanical complications outside.

Producing Gasoline from Natural Gas

By Frank P. Peterson

In 1911 the world's yield of crude petroleum is believed to have been about 520,696,316 bbl.. While difficult to say just how this was split up into the various commercial forms, it is safe to say that not less than 10 per cent., or over 32,000,000 bbl., was used in the form of gasoline. This is notable in contrast to the fact that 20 years ago there was no appreciable demand for gasoline.

An important step in the production of gasoline has materialized during the past 2½ years in the form of stripping waste gases of suspended heavy hydrocarbon vapors and using these condensates as gasoline.

Theoretically, the process of recovery is very simple, namely, the application of pressure and cooling of the gas, provision being made for separating the condensates by precipitation under pressure.

Nearly all the oil wells yield some gas; many yield great volumes while others yield gas only from the strata in which the oil is produced. Again, some yield one gas from their oil strata and another gas from other strata through which the well passes and which is not "cased off."

The gases which issue from "dry" or nonoil-producing strata are often largely pure methane (marsh gas, CH_4), which has a critical temperature of 115 degrees F., and will not condense at any pressure above that temperature.

The gases which reside in the strata along with the oil, however, are in contact with the oil in a finely divided state in the interstices of the sand. There is every opportunity for the methane gas, when issuing from the strata at low pressure along with the oil to pick up and carry in suspension any vaporous oil. Further than this, it is now known that the whole family of saturated hydrocarbons are found together right through the series from CH_4 to a heavy wax structure. This complicates matters, for there are many gases that have and many that have not sufficient suspended heavy compounds to yield paying quantities of gasoline concensates. Hexane (C_5 H_{14}), petane (C_5 H_{12}) and butane (C_4 H_{10}) are the factors which, when present in sufficient quantity, warrant reduction to a liquid state for market. Hence, the prevalence of these condensible constituents must be determined before attempting to compress the gas for their recovery.

One of the first steps is to find the specific gravity of the gas, although this cannot be depended upon entirely. For instance, family A, from one well or group of wells might consist of 10 per cent. methane, 30 per cent. ethane, 30 per cent. propane and 30 per cent. butane. The specific gravity would be about 1.43 (air = 1), and nothing could be produced from it marketable as gasoline.

Family B, from another well or group of wells might consist of 60 per cent. methane, 5 per cent. ethane, 5 per cent. propane, 5 per cent. butane, 10 per cent. pentane and 15 per cent. hexane, in which case its specific gravity would be 1.26 and the 25 per cent. comprising the two heaviest constituents would yield a gasoline condensate, marketable as gasoline of at least 9 gal. per 1000 cu.ft. of gas.

The condensates yielded by the different gases are classified commercially into two general groups: natural gas gasoline and gasol. The first is the unrefined bulk of condensate as it is recovered from the gas and the second is the refined compounds derived by splitting the condensates into groups for different uses.

John Lathrop Gray secured the first broad patents relating to the production of gasoline from natural gas, and the writer, perceiving the need for further refinements, deviated from Grays' constant-pressure, constant-volume method by breaking the operation into steps for the purpose of drawing off fractions differing in specific gravity and boiling points. More recently, however, Dr. Snelling has added what appears to be the final touch of refinement which respects both the critical pressure and temperature characteristics of different compounds and enables the greatest nicety to be realized in the fractionation of the complex gaseous compounds into their several liquid states. It is needless to add that the gasol fractions must be retained and handled under restraint of considerable pressure when they approximate propane and the lighter compounds.

An important phase of the recovery of gasoline from natural gas is that separation is effected under physical conditions maintaining a close equilibrium between the liquid and the gaseous states. Because of its lightness as compared to the mediums ordinarily handled by baffling and trapping, much difficulty has been experienced in eliminating uncertainties in the operations; hence, a vast amount of "cut and try" application.

Physically the product in appearance is white. It carries, usually, a very small (negligible) percentage of heavier oils, sometimes lubricant from the compressor cylinders. In time it evaporates to complete dryness comparable to and usually quicker than a refined oil product of equal heaviness. Its specific gravity varies widely, in different regions and with different pressures applied to obtain it, ranging from 0.625 to 0.71 (water = 1), or from 96 to 68 deg. Baumé.

As a fuel for gasoline motors its carburetting qualities are excellent. Consequently, the impression has become more or less general that its power value is much greater than ordinary refined gasoline. This is not true, except in the sense as the hydrogen ratio of a fuel increases, so does its gross fuel value.

Refined stocks, because much less volatile, can be stored without loss, while the new product cannot, even in winter, be stored in open tanks without loss. As fuel for light gravity gas lighting machines it is good, but its most fitting place seems to be in mixture with the refiners' low gravity gasoline or naphtha fractions whereby a motor fuel of maximum power and the best carburetting characteristics may be had.

Many instances have already occurred in which the abandonment of a few small oil wells in oil fields has been saved by the installation of equipment to recover gasoline from their heretofore wasted gases. And it is not unusual for such wells to yield a greater amount of gasoline condensates than they do of crude oil. And as the condensates have a value at the well of about twice that of the oil, it can readily be seen how great a boon it is to the small oil operator whose revenue cannot be maintained sufficient for livelihood from the oil alone. The present production is about 2500 gal. per day.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Raw Water Can Ice Making Systems

By Samuel Sydney

Distillation of the water for freezing in can ice-making plants has generally been considered essential to secure clear crystal ice that would equal in appearance the best natural product. Recently a number of methods of making clear ice in cans without prior distillation of the water have been tried out, and most of them with a fair degree of success. Clear plate ice has always been produced without distillation, but there are numerous objections to the plate system, and the fact that about 90 per cent. of all the ice-making plants installed are can plants indicates the superiority, or at least the belief in the superior fitness of the can system.

Raw-water ice-making plants in use at the present time may be broadly divided into two general classes: those employing stationary ice cans, from which the ice is lifted out of the can, when the freezing is completed, and those utilizing the ordinary removable ice cans used in making distilled water ice. Of the first class there are two distinctive types: the Jewell system, with a "nonfreezing zone" at the bottom of the can, and the Pownall system with a siphon-connected precipitating tank adjoining the freezing tank.

Raw-water freezing systems of the ordinary can type have been in use for many years with more or less success, but only within the last year or two have they been so perfected as to appeal to prospective ice manufacturers sufficiently to induce any considerable number of installations. During 1912, however, a score or more of such plants have been installed and appear to be operating successfully.

Experiments in freezing raw water in cans by agitation with air were made years ago, there being records of successful experiments in Geneva, Switzerland, in 1877, in England in 1879 and in Brooklyn, N. Y., in 1882. Victor H. Becker made clear raw-water ice with an air blast in the cans in this country in 1874, antedating them all. But these remained as experiments and were soon abandoned for the distillation process.

The early experimenters who succeeded and are still operating on their plans are J. E. Simon, of Louisville, Ky., who secured a patent on his system in 1898 and who operates a 100-ton raw-water ice plant in Louisville, and E. J. Ulrich, of Colorado Springs, Colo., whose patents were issued in 1901 and who operates plants in Colorado Springs and in Salt Lake City.

The extra labor required in the tank room of a raw-water plant adds to the cost from 4 to 9c. per ton of ice made as compared with distilled water ice, but the saving of labor and fuel at the steam end of the plant not only counter-balances this extra tank-room labor cost, but considering also the saving in space requirement and in original investment, raw-water ice is actually being made at from 10 to 20c. per ton less than in distilled-water plants of equal capacity.

Simon System

In the Simon system previously referred to, air pipes are laid in the partitions of the freezing tank, depressed in alternate rows, with air cocks attached facing both ways. The air cocks are screwed into the bottom of the air line, and each has an orifice $\frac{1}{16}$ to $\frac{3}{32}$ in. in diameter. A flexible connection is made between these air cocks and an air tube extending to within about 6 in. of the bottom of a 300-lb. can. This depending tube is made double, consisting of a brass pipe with an ½-in. bore within which is another pipe with an ⅛-in. bore. The outer pipe is closed around the inner pipe with a ½-in. bore within which is another pipe with a ⅛-in. bore. The pipes. As soon as the ice can is filled with cold water from a well or city main one of these depending pipes is hung in the water at the center of the can and sustained by a tube holder held to the eye of the can by a spring. Air pressure at 1½ to 2 lb. is then turned on, keeping the water in constant agitation until all of it except about two quarts at the center of the can is frozen. The air tube is then withdrawn, the water sucked out and distilled water placed in the pocket and allowed to freeze. A clear crystal cake of ice with scarcely any visible core results. It should be added that the air before it is admitted to the main air pipes on the tank frame is passed through a dehumidifier which removes excess moisture in the air and prevents danger of snow or frost forming in the depending tube interfering with the free flow of the air to the water in the can.

Ulrich System

In this system each can is made with a pipe running down the outside of the can and across the bottom, entering through a small expansion chamber and check valve at the middle of the bottom of the ice can. For the air supply an air header, with suitable laterals is laid in the framework of the ice tank, or under it. The laterals are connected with the pipe on the outside of the ice can by means of a flexible hose connection. Inserted in the air laterals are ¼-in. valves for each can, to control the air supply. In the early experiments with this system the air pipe passed through a reservoir at the bottom of the can, but it was found impossible to prevent freezing of the orifice in the air pipe and the reservoir plan was abandoned. Instead, the air after leaving the air compressor is cooled by passing through water-covered coils and then through an air drier. After that no difficulty was experienced from freezing of the air pipe at the pressure carried about 10 lb. The ice is pulled and harvested as in the ordinary distilled-water can plants.

Ashton's Modification

A modification of the two preceding systems is that in use in a large ice factory in St. Louis, Mo., devised by its superintendent, W. A. Ashton. In this plant the air is conveyed through a header along one side of the tank with laterals branching off across the top of the tank between each two rows of cans. The laterals are laid in the tank frame and communicate by means of a hose connection with the separate galvanized-iron air pipes provided for each can. A valve on the lateral controls the air supply for each row of cans. The air pipe for insertion in the ice can is provided with a short horizontal piece at the top for hose attachment and at the bottom has a small piece of flexible rubber tubing with an opening of only $\frac{3}{32}$ in. The action of the air keeps the flexible rubber end in constant motion which aids in agitation of the water and tends to prevent the possible closing of the orifice by frost. In operation the cans are filled with city water by means of a special can filler, the air pipe, which is supported at the top with a small cleat, is dropped in, and air from the compressor at about 8 lb. pressure is turned on. The air, however, is first passed through a copper-tube cooler and its temperature reduced with water, after which it is passed through an air drier and deprived of its excess moisture. After about 85 per cent. of the water in the can is frozen the air pipe is with-

drawn, the water remaining in the can is sucked out by means of a vacuum pump and distilled water run in through a hose to take its place.

Some Chicago Installations

An improvement on the St. Louis method is that employed at the Lincoln Ice Co.'s plant in Chicago. Here the air mains and laterals are laid, as in the St. Louis plant, in the ice-tank frame, but between each two rows of cans brass castings with a reamed socket on each side are fastened into the air laterals. A tee pipe is provided for each can, one end of the tee being beveled to fit snugly into the reamed opening of the brass casting. The other end of the tee is pressed into a spring socket at the opposite side of the tank frame, the spring holding the tee firmly to the brass casting. A pin hole in the casting admits the air into the tee, its expansion serving to cool the air to near the temperature of the water in the can. The ice cans are held down firmly in the tank by a single cleat adjusted in an instant. The setting in of the tee pipes is also done quickly, one end of the horizontal part of the tee being pressed against the spring socket and the other allowed to spring into the reamed opening of the brass casting. The spring holds the air pipe firmly in place. There are no flexible connections and the air pipe stays where put until time for removal. Air pressure at about 6 lb. per sq.in. keeps the water in rapid agitation and prevents freezing of the orifice of the tee pipe. The latter is thrown out when all but from 10 to 15 per cent. of the water is frozen, the nozzle of a suction hose is dropped in and the water exhausted and replaced by distilled water. A practically coreless clear cake of ice results.

In another raw-water ice-making plant in Chicago, a somewhat different arrangement for supplying air to ice cans is used and no distilled water is required for the core. In this plant a tank containing 20 rows of ice cans is installed and a main air header is laid lengthwise through the center of the tank in the tank frame with a threaded nipple provided with an air cock opposite each transverse row of cans at each side, this main serving ten rows of cans the length of the tank. Similar air mains are laid at the extreme sides of the tank, these having nipples and air cocks to one side only and each serving five rows of cans.

Instead of a separate tee pipe for each can the tees are prepared in rows of five. They consist, in fact, of five pipes depending from a common cross pipe. One end of this cross pipe is sealed, the other end is threaded, the threaded end fitting into a union on the end of the threaded nipple of the air main. The cans are filled five at a time, the five-branched air pipe is set in, the cross pipe being supported on the cleats that hold the ice can rigid, and the connection at the union is made by hand without tools. At the same time the air cock or faucet is opened and the air flows into the five cans at once.

At this plant 400-lb. ice cakes are frozen and tee sets with depending fingers about 46 in. long are first inserted, but after the cake is about half frozen these are lifted out and tee sets with depending ends about 28 in. long inserted in their stead. When the water is frozen to within about 1½ to 2 in. the short-arm tees are removed, the center core of water sucked out and cold raw water from the forecooler tank is run in to fill the cavity. The covers are then replaced and the remaining water allowed to freeze. The result is a 400-lb. cake of clear ice with no more core than distilled water ice and with only a very small cup or depression at the top of the cake.

Jewell System

An entirely different proceeding is followed in the raw-water ice-making systems using stationary ice cans. In the Jewell system there is no freezing tank or brine tank such as is used in the ordinary can ice-making plant. Instead the ice can itself performs the function of the ordinary brine tank. The ice can first used had a double wall, with a ⅝-in. space between walls. The can has an extension at the bottom, termed a nonfreezing zone, which is not jacketed as is the rest of the can, and is only about one-third the width of the can. A recent modification of the system does away with the jacketed can, using instead an ordinary can with the nonfreezing zone extension at the bottom. These cans are permanently fixed in a tank, or "cell," with common connection through suitable headers, to brine cooler, air system, water supply, etc. As many "cells" are installed as the size of the plant may warrant.

The cans are arranged in rows, and the cells are insulated with 8 or 10 in. of granulated cork or other good insulating material. A ¼-in. air pipe from an air compressor or blower system is connected to the bottom of each can, the air pipe terminating in the upper part of the nonfreezing zone, and thus obviating any danger of freezing the orifice and blocking the air current.

After the cans are filled with water and the air current is turned on, cold brine from an adjacent brine cooler is circulated through the ⅝-in. space between the can walls or around the cans in the cell tank, by means of a small brine pump. The brine used is generally of low temperature, near zero in the jacketed-can type, but about 12 deg. F. in the cell type. The water in the jacketed cans is frozen rapidly, the ordinary sized 300-lb. cans taking from 26 to 34 hr. The rapid freezing makes it necessary to allow more time for tempering the ice before it is pulled, or cracking will result. When a row of cans is frozen, the brine flow is shut off and the brine remaining in the can jackets is put in circulation through the coils in the forecooling tank which contains the water for the next charge of the can. This partly warms the brine in cooling the water tank and at the same time gradually tempers the ice. This circuit is continued for from two to three hours or until the ice in the cans is loosened so the ice can be pulled. To facilitate the final thawing the water remaining in the nonfreezing zone is drained out and the next charge for the cans is turned on. The water, it should be added, flows to the cans by gravity, the quantity admitted being regulated by an automatic stop cock or valve.

For convenience in harvesting, small strips of iron with a hook at the lower end and an eye at the top are hung over the sides of the can and are frozen into the ice. The ordinary can dogs of an ice hoist fit into the eyelets and lift out the cake. A quick tap knocks the iron strips loose so they can be used over again.

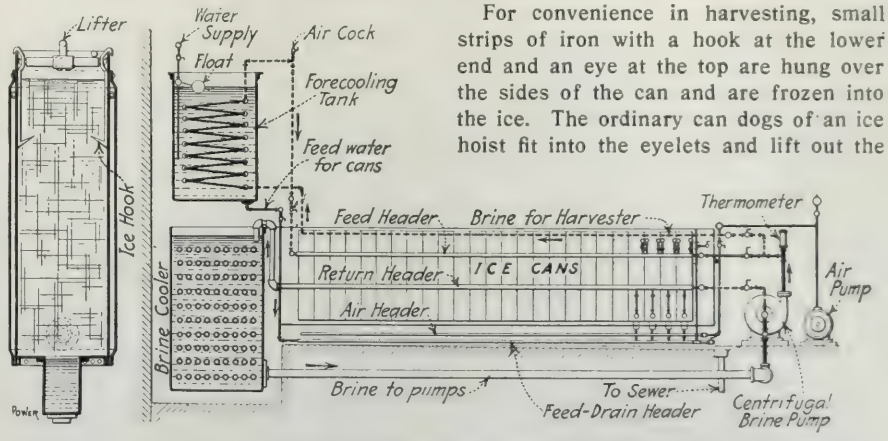


Fig. 1. Jewell Raw-water Ice-making System

During freezing all the impurities in the water are rejected and sink to the bottom into the nonfreezing zone. The writer has seen ink thrown into the water in the can yet a perfectly clear block of ice pulled. The liquid in the nonfreezing zone was fairly good writing ink.

Pownall System

In the Pownall system stationary ice cans are employed as in the Jewell system, but there the similitude ends. The cans are permanently fixed in the tank and are placed very close together, or with only about ⅞ in. space between the

cans. Well insulated covers are provided, as the cake is frozen from the top downward. There are no pipes or coils in the tank but brine from a brine cooler, usually placed alongside of the tank is circulated around the cans by means of propellers, the temperature of the brine being carried somewhat lower than is customary with distilled-water ice. At the bottom of each ice can are fixed flanges and connections to a water-inlet header and a water-outlet header and an air-supply pipe. All the cans in a tank, if of moderate size, or in a section, if the tank is large, are connected as a unit for water inlet and outlet, air supply and ice harvesting. That is, all the cans in one tank or one section of a tank are filled at the same time and pulled at one time.

Adjoining each ice tank, or section, is a precipitating tank having connection with the bottom of the cans through a common header. The precipitating tank extends somewhat lower; that is, is deeper than the freezing tank. The air current is admitted at the bottom of the can between the water inlet and outlet pipes through a ½x3½-in. nipple. The air pipe is so connected to the water pipes as to create a siphonic action, and cause the water in the can to rise to slightly higher level than in the precipitating tank, flowing then by gravity through the other water connection to the precipitating tank, thus maintaining circulation of the water. The purpose is to cause the impurities in the water, expelled during the crystallizing process, to collect in the precipitating tank where there is no air agitation and settle therein. Sometimes after freezing a tank full of water 4 or 5 lb. of solids are found deposited in the precipitating tank.

The air pipe mentioned is a small galvanized-iron pipe with ⅛-in. opening carrying about 2½ to 3 lb. air pressure. As a rule the air is not cooled or dried or expanded as in other systems, as there is no danger of freezing the orifice of the air pipe.

When the water in the cans is nearly all frozen, or say to within 3 per cent. in a 300-lb. can, the air pipe is shut off, the small amount of water remaining in the cans is drained off through a small pipe provided for the purpose and distilled water is let into the opening through another connection controlled by a valve. The freezing is continued until all the water is frozen into a clear coreless cake of ice. The brine circulation is then stopped and the ice is allowed to temper for an hour or so. Switched over to another connection the brine is then circulated through the coils in the water-storage tank holding the next charge for the cans, which cools this water and at the same time warms the brine enough to thaw loose the ice frozen in the cans. The ice is then lifted out of the cans by a hoist, small iron hooks having been frozen in, as in the Jewell process to furnish a hold for the cross-bar of the hoisting apparatus. It is claimed that from 10 to 12 tons of ice per ton of coal burned is possible with this system.

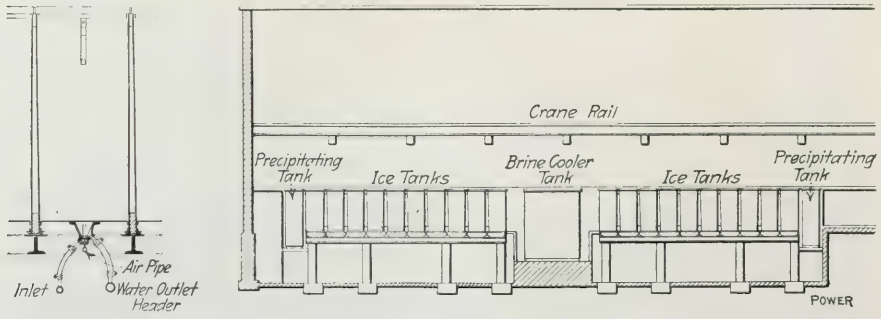


FIG. 2. POWNALL SYSTEM WITH ITS BRINE COOLER AND PRECIPITATING TANKS

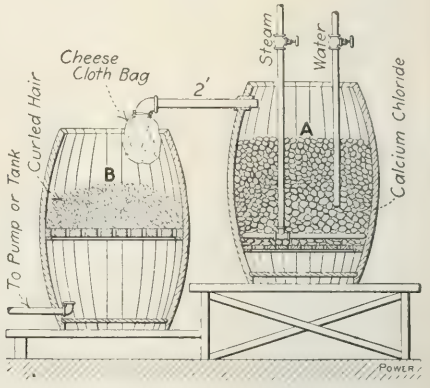
large tierces such as are used around packing houses for pickling purposes. The barrels hold about 300 gal. each and are on stands close together, one about 1 ft. higher than the other.

About 12 in. from the bottom of tierce *A*, I fitted a round cover the exact diameter of the tierce at that point and perforated it with 1-in. holes on 4-in. centers. This strainer is to allow all the nondissolvable matter in the calcium chloride to drop through to the bottom of the tierce, to be removed later.

A ¾-in. line supplies steam for boiling the solution. A 1x¾-in. tee on the end of the pipe is fitted with two semi-circular 1-in. pipes bent to fit the shape of the tierce and capped on the ends. The pipes are well perforated with $\frac{3}{16}$-in. holes close together which give the steam a free outlet into the solution with practically no noise even with the steam valve wide open.

A 1¼-in. water pipe extends down to within 1 ft. of the strainer, and about 4 in. from the top of the tierce is a 2-in. overflow pipe long enough to allow the brine to pass over into tierce *B*, where it is strained and delivered to the pump or direct into the freezing tank, as the case may be. Halfway down in tierce *B* is a strainer similar to the other one, covered with curled hair to catch any foreign matter or dirt that might pass over from tierce *A*.

At the bottom of the tierce is a 1¼-in nipple, with an elbow on the inside turning up, as an outlet for the brine. This prevents any sediment that might filter through from going to the pump and thence into the system.



BRINE-MAKING OUTFIT

In making the brine I first cut up the drum of calcium chloride with an old rip saw into about six or eight slabs and break these up with a sledge into sizes about as large as an apple.

Tierce *A* is then filled about three-quarters full of the calcium chloride and water turned on until within 4 in. of the overflow. Turning on the steam I let the solution come to a boil and when it is strong enough, which is determined with a Baumé hydrometer, I open the water valve enough for the solution to slowly rise and flow over into tierce *B*, testing the solution often so that it will not get weak from adding too much water. The density of the brine can be regulated very closely with the water valve. As fast as the calcium chloride is dissolved more is added so that the tierce is always half full of calcium-chloride lumps. This method gives good results for the density we use, which is 30 deg. Baumé.

C. E. ANDERSON.

Chicago, Ill.

CORRESPONDENCE

Making Calcium Chloride Brine

A simple apparatus for making calcium-chloride brine for refrigerating purposes, which I have used for some time with entire satisfaction, is shown in the accompany sketch, where *A* and *B* are two

The smoke nuisance costs the United States $500,000,000 a year, is the estimate given by the department of industrial research at the University of Pittsburgh. This damage consists in the defacement of buildings, tarnishing of metals, damage to merchandise, injury to both human and vegetable life, increased cost of housekeeping and loss to manufacturers through the imperfect combustion of coal. The loss in Chicago alone is estimated at $50,000,000 a year.—*Gas Review.*

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POWER

Non-Return Stop Valves

A boiler tube in a large plant recently pulled out of the drum into which it was expanded, a number of men were seriously hurt and the service was interrupted. Had it not been for the heroism of an attendant who rushed through the steam-filled room, and closed the stop valve, thereby horribly burning his hands, every boiler would have lost its pressure, hours of idleness for the entire establishment would have followed, and possibly those in the boiler room would have been killed.

It requires an iron nerve and an extraordinary sense of duty to close a stop valve under such circumstances. Hardly ten men in a thousand would risk it. The nonreturn stop valve in such cases allows only the one boiler to be rendered unserviceable and the time saved thereby is an item usually in excess of the cost of the valves.

And this is not their only good feature. As with some forms of insurance policies it is not necessary to "die to win," with these valves it is not essential that a boiler be disabled to prove their advantages. We all know the care that must be exercised in cutting a boiler in on the line with others under pressure. The pressures must equalize before the valve is opened and if the steam gage on which the engineer solely depends for guidance is defective, disaster may follow the opening of the valve. The automatic nonreturn valve obviates this danger by not opening communication between the boilers until the pressures are equal.

Those who have advocated the compulsory use of nonreturn stop valves have met stubborn opposition. Plant owners are sometimes reluctant to believe that they will be benefited by installing such valves. It is learned that they protect the fireman in the event of a ruptured tube or seam. This is surely their paramount feature, but it is talked of so much that their other advantages are forgotten. This is unfortunate.

Railroads are liable to damages if a passenger slips on a car step not provided with an adhesive mat. This is because the mat conduces to safety and is a commodity on the market, the price of which is not prohibitive. Should not owners of boilers be liable for boiler accident, the dangers of which can be minimized by the use of the nonreturn stop valve?

The Engineers' Study Course

With this issue we commence a study course for engineers. The idea is not a new one. A similar course added to the popularity of POWER in the early nineties and many engineers who are occupying positions of trust and responsibility today owe their start to the opening thus afforded for self-instruction, and their initial impetus upon the road to advancement to the interest awakened by this course.

The engineer deals with quantities which must be computed, "figured," in their own magnitude and in their relation with other quantities. He ought to be an adept in arithmetic, he must have a clear conception of ratios and proportions and know all about percentage. The initial lessons of the course will lay this groundwork, but they will be made to teach something more than plain arithmetic by the selection of examples dealing directly with power-plant subjects, so that the student will be learning engineering as well as arithmetic.

Another purpose which will be persistently followed is that of teaching the reader to analyze a problem, to see the relations of various quantities to each other and know the unknown can be determined from the known rather than to load him up with a collection of rules to be learned by rote, or filed away to be hauled out, not always with the best discrimination, and applied with more or less precision to the case in hand.

As the problems in connection with these lessons will serve to a large extent the same purpose, the Study Questions which have been running for several months will be discontinued with this issue.

Snap Judgment

Some engineers are much too hasty in their condemnation of what might prove to be a highly useful article in their technical paper. They merely glance at its contents or cast it aside as worthless before they have given careful consideration to its purpose and endeavored to ascertain if it may not contain a solution of an operating problem that has been causing them hours of hard work and study.

The same critical judgment should be used by the engineer in reading his technical paper as is expended in investigating the claims made for a machine that is recommended for economical operation and efficiency. The careful and thorough engineer looks before he leaps, and he knows that while a machine may be worthless under certain conditions, under others it has often proved to fully answer its purpose.

He knows, for instance, that a mechanical stoker which will effect a good economy with a large-size coal is often a failure when smaller sizes are employed. A low-pressure steam turbine may increase the plant efficiency if added to an installation of reciprocating engines run noncondensing, but if operated with a low vacuum or run directly from a low-pressure steam system, its use may be highly inefficient.

Snap judgment is studiously avoided by those engineers who are always seeking for the best, whether it is a machine or an article in their technical paper. Both of the latter have their right uses, and, in determining what such may be, one is deserving of careful consideration as well as the other.

The Junkers Engine

For the past two or three years the Diesel engine has been a central figure in engineering discussions due largely to its exceptionally high efficiency and to its extended use abroad.

In spite of its many advantages, however, it possesses certain inherent features which tend to make the first cost high, the construction somewhat complicated, and the employment of skilled labor necessary.

The Junkers engine, which is now attaining some popularity in Germany, operates on the Diesel principle, but is entirely different in construction from the latter engine. Its internal parts are much simpler as no valves nor cylinder heads are employed and the castings are very plain. Moreover, its reciprocating parts are perfectly balanced. On the other hand, its external parts are somewhat complicated.

In the series which is now running in our "Gas Power Department," the author sets forth the contentions of Professor Junkers in support of his engine, which include interesting comparisons with the Diesel engine. Our pages are always open to useful discussion, and we trust that our friends, especially those who are Diesel adherents, will take this opportunity of expressing their views. Professor Junkers will be given an opportunity to reply to all comments and as far as possible we shall endeavor to print the comment and the reply in the same issue.

Self Overvaluation

It is but natural that the engineer, who by dint of years of study and hard labor has become head of a power plant, should be inclined to overvalue his services to his employer, especially when his plant operates a factory.

Admitting that industrial success is in a way dependent upon the economical operation of the power plant, it is a fact not to be lost sight of that the average factory manager regards the power plant as an incidental. In some industries the entire cost of manufacture has been placed as low as two or three per cent.

If such is the case, the engineer becomes but a slim spoke in the industrial wheel. He may claim that "nothing can run without him," but he must also know that without the factory and its product his services are valueless.

The factory manager welcomes his returning salesman with a warm grasp of the hand, because through him is found a field for the factory output. But this same manager will frown on the approach of his chief engineer and brusquely inquire: "Well, what is it now, more repairs, supplies, or is it another breakdown? Every time I see you coming it means some additional expense."

The estimation in which an engineer is held by his superiors largely depends upon the former's personality, and while he should cultivate a certain amount of self-assertion, he must also bear in mind that but few superiors will accept him at his own valuation. Managers have apparently taken it for granted that the power plant is a necessary evil, and if the engineer hopes to escape humiliations and rebuffs, his best plan is to try to reduce the "evil" to its lowest possible denomination.

Dealing with Meddlers

Blocking the meddler is one of the disagreeable, but often very vital duties of the power-plant engineer. The very breadth of engineering activity offers a tempting field for the man who delights in finding fault and thinks he can show the engineer up in a poor light.

Between legitimate inquiry and meddling the line is very flexible. It changes according to persons and circumstances and is fixed sometimes by tact. The owner may be a meddler instead of a judicial supervisor. The meddler may be a poorly selected representative of the owner, incapable of carrying out his intended functions, or one of the office boys looking for a chance to gain favor of the owner.

Competent, authorized persons are at all times entitled to the fullest knowledge of power-plant conditions. Correctly made-out reports from the engine room, with the ability to elaborate upon them if required to, meet this situation. When the inquiring is inquisitiveness or just plain meddlesomeness, it is still best to be able to explain, and if necessary over the head of the meddler. Every failure here is a reflection upon the profession; every victory a credit to all engineers.

The man with education, not necessarily collegiate, tempered with tact, seasoned with observation and study, tied with hard knocks, with experience behind him to back up his statements is the man who can block the meddler.

Here is an apparent excuse for using a big word. The literary fellows tell us that using big words only because they are big, is a vice, but a big word, when it is the best word, needs no apology. Besides, if we were sticklers for "the simple folk vocabulary of our Saxon forebears," we could not always find words to express our meaning, particularly in engineering experience.

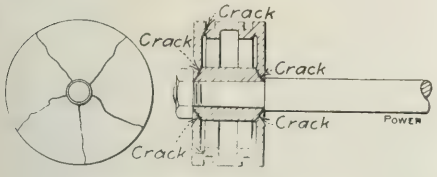
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Mysterious Piston Fracture

Recently I had charge of a 12x12-in. duplex single-stage air compressor working against 100 lb. receiver pressure, to which three accidents of exactly the same nature happened that I have not been able to account for, to my satisfaction, and would like to hear from readers as to their theory of the cause

The illustration shows the nature of the accidents very clearly. A vertical section and an end view of the head are shown, indicating the nature of the fracture. Nothing was left on the rod except the hub or eye through which the rod passed. It would appear that the fracture was caused by a force acting from within the piston. My conclusion from the first accident was that water had accumulated in the head in some manner and during high compression had generated into steam and exploded the head, but when the head on the other side broke in exactly the same way a short time after, I lost confidence in this conclusion, and when in less than a week the piston which we had renewed fractured likewise, it weakened still more, for it would hardly seem probable that all three could be faulty in this manner. After the third accident I had solid heads put in and had no further trouble. What was the real cause?



BADLY CRACKED PISTON

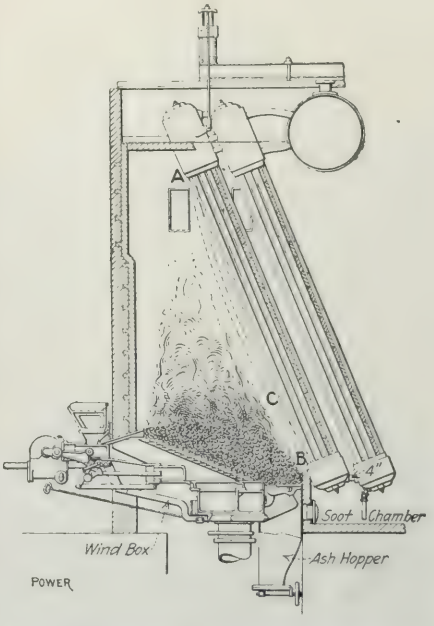
H. R. ROCKWELL.

Alten, Ill.

Interesting Boiler Experience

When our four 500-hp. water-tube boilers, equipped with mechanical stokers and using balanced draft, supplying steam for an induction-motor load varying from zero to 4000 kw. were started about two years ago, everything went well for a few months; then one night a tube broke at *A*. Very little was thought of this, and it was "logged" that the tube had been cut when expandedfi thereby causing the leak. In a couple of days a tube broke in another boiler. This continued until it got so that when one boiler was cut in, another boiler was ready to be cut out, but we were fortunate in not having two boilers cut out at once. It was the front tubes of the front units that always gave out. Each unit has twenty-one 3¼-in. tubes 18 ft. long. Two units and part of the steam drum are shown herewith. The tubes would crack at either *A* or *B* and frequently a blister would be raised as at *C*.

Numerous causes and their proper remedies were suggested by those about the plant. Some said the only remedy was to install two more boilers. Others said charcoal-iron tubes would prevent further trouble; while another claimed that the ends of the steel tubes should be annealed. Another contended it was the fireman's fault for carrying the fires too heavy and getting the tubes too hot, while another stated that the fires were carried too light, allowing holes to be blown in them, creating a blow-pipe effect on the tubes, thus forcing the water away, causing steam "bubbles" and overheating of the tubes. Another blamed the damper regulator for not being sensitive enough. It used to vary about 7 lb. from closing to opening. By the time the fan came up to speed a sudden load came on, and the pressure would drop 10 to 15 lb. This, he claimed, caused the boilers to prime. With the sudden drop of pressure the temperature of the steam would also drop, but the water would retain its temperature and tend to flash into steam at the reduced pressure. While this action was going on the front of the front tubes being in contact with the hottest gases, would overheat, causing them to sag as shown by the dotted lines.



PARTIAL SECTION OF WATER TUBE BOILER SHOWING DEFORMATION OF TUBE

Another said the 4-in. nipples which connected the units at the bottom, did not allow sufficient water to pass through, especially when they were coated with scale. He held that the great fluctuation of the water level every time a load was thrown on suddenly proved his claim. This was brought to the attention of the boiler manufacturer whose representative said that when the draft fan was stopped the induced fan would circulate cold air around the tubes, especially if one of the cleaning doors was opened, and this caused the tubes to warp.

Discussion continued and the tubes kept breaking until the company shut down an old street railroad station and transferred its load after installing three 500-kw. rotary converters. There was an immediate improvement; the synchronous motors raised the power factor, a big decrease in coal consumption per kilowatt-hour resulted, and the peak load could go to 5500 kw., without a decrease in steam pressure. The load seldom drops less than 700 kw. This keeps the fires hot all the time, and at present the peak load does not go over 4500 kw. Three boilers are used, the fourth being banked the greater part of the time.

THOMAS SHEEHAN.

North Adams, Mass.

Passing of Some Notable Engines

Only a few years have passed since the late P. W. Willans, of Willans & Robinson, Ltd., Rugby, England, came forward with his high-speed engine. It was to revolutionize the engineering business and provide an ideal motive force. Licenses for its manufacture were granted to American, German, Swiss and French engineers. Now, not only is the engine no longer made, but that it ever existed is almost forgotten, even at Rugby.

In conjunction with Mr. Willans, two others should be named as pioneer builders of high-speed engines, W. H. Allen, of W. H. Allen & Son Co., Ltd., now of Bedford, England, and who is still actively engaged as the head of a huge business, and the late Peter Brotherhood, of Brotherhood, Ltd., now of Peterboro, England. At this period these firms were all established in London and no doubt

the success of this engine was the prime cause of all of them establishing modern works in the provinces. It was quite common to see a long line of these engines in British power stations some fifteen to twenty years ago.

There are still several firms engaged in making high-speed engines in Great Britain. The Allen generating sets (which are made complete at the Bedford works, and for which Mr. Allen may also be termed a pioneer of electrical engineers, the firm going into the electrical business almost before it was a commercial success) are extensively used, especially in the British Navy, and upon ships generally. They were installed on the "Olympic" and her ill-fated sister-ship. But this firm for some time has been engaged upon the turbine and oil engine, as are most of the other British firms connected with this branch of industry. As it is doubtful if any firm outside of Great Britain is making this type of engine now, its end may shortly come to pass, and its place be taken by something more reliable and economical.

Another event was closely connected with the introduction of this engine. Prior to this time, shops were run on more or less secret lines—under old conservative ideas. There was little interchanging of thought between shops, much less between nations. After inventing his engine, Mr. Willans read a paper before the Institution of Engineers in London on "The Willans Engine and Its Construction" which marked the opening of a new order of things. Criticism was plentiful, but Mr. Willans convinced most that his ideas were correct, and the secret method was dead. Today we rejoice in the "open shop." Papers are read and transmitted broadcast by the press, representatives of various nations, especially British, American and German visit each other and are received with open arms wherever they go. In this America may be said to lead.

We give publicity to our ideas and court the criticism of the world, but for the late Mr. Willans to do so some twenty years ago, was then thought worse than folly.

F. P. STRACHAN.

Belfast, Ireland.

Flywheel Fan

Engineers having engine rooms with low ceilings and poor ventilation can improve conditions during the hot months by adding a fan to the flywheel or other wheels that may be handy. I use two sheets of medium-weight galvanized sheet iron about 10x30 in. clamped to the flywheel arms with U-shaped clamps having one end about 1¼ in. longer than the other, so as to give the blade the proper pitch. The speed of the engine is 170 r.p.m. Care should be taken to draw the air from the direction in which it will not take up dust.

A fan as arranged will cool the room and also the main bearings and eccentrics. Before attaching the blades it must be determined how large they should be and whether or not the engine will properly carry its load with blades so attached.

C. V. COOPER.

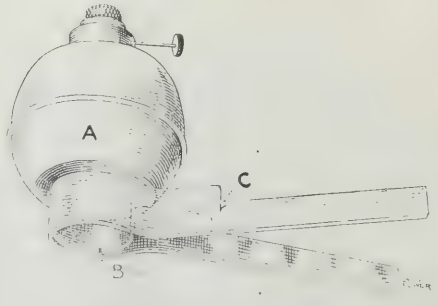
Greenfield, Mo.

Useful Hand Lamp and Holder

The usual type of oil lamp is inconvenient and clumsy in many places. Especially is this the case when expanding tubes or working inside a firebox or smoke box of a locomotive or other type of boiler. Shops where portable electric lights are available do not need such lamps, but they are very good in other plants.

A foreman boiler maker of a portable-engine works making locomotive-type boilers is the inventor of the lamp and holder shown. The device is simple; a small lamp with a globular tin container *A* fitted with a stem to turn up the wick is the type chosen and purchased as required. The lamps have a short piece of 1-in. diameter thin brass tube *B* soldered to the center at the bottom.

The holder is made of $\frac{1}{16}$-in. spring steel 1 in. wide. The part which encircles the tube is $\frac{7}{8}$ in. diameter, so that when the holder is sprung on the tube at the bottom of the lamp it grips it and leaves the two free ends of the holder about 3 in. apart.



HANDY LAMP FOR USE WHILE CLEANING BOILERS

The free ends of the holder when pushed into a tube exert a pressure on the tube walls, which holds the lamp from turning. It will be noticed that the free ends are reduced in width. The shoulder thus formed prevents the ends being pushed too far into the tube.

When stood on a flat surface the splayed ends of the holder prevent the lamp from being readily overturned. It may be seen that any lamp of suitable shape can be converted into this type with little trouble.

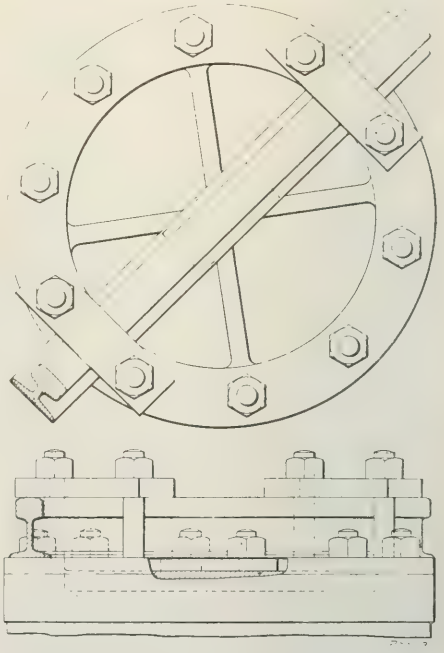
A. L. HAAS.

London, England.

Poor Gaskets Served Good Purpose

Recently we installed an 18½x17-in. engine belted to two generators. This outfit was second-hand and has caused all manner of trouble.

One morning this engine was started in the usual way, allowing about 10 min. to get up speed by admitting steam very slowly. Everything seemed all right, but as the engineer was about to cut in the generators the cylinder cracked in four



STEEL RAIL HOLDING FRACTURED CYLINDER HEAD

places. I immediately shut off the steam and started to investigate. Water was all over the floor and some was still dripping from the cylinder-head joint. The cracks were about 1½ to 2 in. long, all centering at the ⅜-in. eye-bolt hole. The breaks showed plainly on the outside, but could not be seen on the inside of the head. Upon removing the head we found that the gasket was nearly all gone, the pieces being found later on the floor.

This damage was done by entrapped water, notwithstanding that the cylinder has two 1½-in. relief valves and both were wide open. There is also a ½-in. bleeder on the steam chest, which was open. I have a very warm spot in my heart for that gasket, because I believe that if it had not blown out, the cylinder head would have done serious damage.

Fortunately the head was not damaged beyond repair, so we removed four of the studs from the cylinder and put in their place four 9½-in. studs, and then bolted a piece of track steel across the face of the head, as shown. This head is recessed so we had to put an iron block in between it and the rail. After this

repair the engine was started and has been running every day since with no indication of giving away.

This only serves to warn the engineer that he cannot be too careful about thoroughly draining the steam pipe before starting the engine.

J. B. DANIEL.

Chicago, Ill.

Favors Non-Return Stop Valves

In nearly every power-plant paper that one may pick up is cited an account of some accident to boilers or steam piping under pressure in which life is lost or endangered from the escaping steam and water. Many of these accidents could be avoided by using non-return valves on the boiler branches which would close at the time when it is most important that they be closed.

I have been through three of these accidents, and a fourth happened about 15 min. after I was relieved by the night man. The first two were almost identical and were each caused by a split tube in a water-tube boiler. There were six 250-hp. boilers on the line under 150 lb. pressure and nothing but common high-pressure gate valves on the branch between the boilers and header. The only damage was to the foreman who was burned. The mill was shut down for 1½ hr., but the cost of the accident in each instance, in time lost in the mill getting started again, was enough to equip all the boilers with non-return stop valves. The third was caused by a 10-in. steam pipe pulling out of a flange; no one was hurt, but it was impossible to close the valve as it was near the break, and each boiler had to be cut out by hand and the fires drawn.

The cost in this case would have nearly paid for the installation of these valves. The fourth was caused by water-hammer breaking the stop valve while the boiler was being cut in on the line, and nearly cost one man his life, and shut the whole plant down for about two hours. If there had been automatic valves on the boilers in these instances the trouble would have been local, and would not have affected the whole plant as it did.

Some of the nonreturn valves only close in one direction, that is, against the header pressure. This will cut out the boiler if a tube bursts, but would not close against the boiler pressure if the steam pipe burst. I believe they should close either way. Boilers are not any too safe at their best after they have been in use a few years, and I consider the non-return stop valve just as necessary from a point of safety as any other appliance except the safety valve. The inspection laws should be so amended that their use will be compulsory where two or more boilers are connected to a common steam line.

A use for the non-return stop valve on the boiler, other than its operation in case of accident, is in cutting the boiler in or out of service. The usual procedure where hand-operated stop valves are used is to watch the steam gage until the pressure appears to be about equal to that of the boilers in service, then hurry to the top of the boiler and open the valve. Sometimes, due to the gage being inaccurate or other reasons, there may be 5 to 15 lb. difference in the pressures, or the fireman's attention may be called to something else at that time and the pressure in the boiler may become too high, which makes cutting in more dangerous. This would be avoided by using these valves.

These conditions help to increase the number of Monday-morning accidents. With the non-return stop valve in use the hand-operated stem may be screwed up (if it has been closed) at any time while raising steam, and when the pressure in the boiler is equal to that in the steam line the valve will open of its own accord whether the fireman is watching it or not.

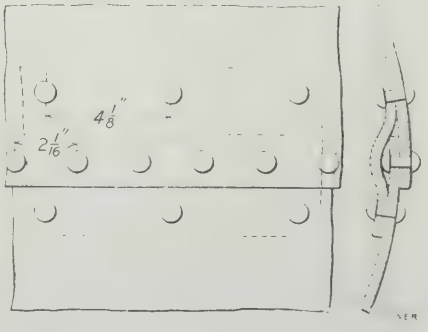
J. C. HAWKINS.

Hyattsville, Md.

Boiler Joint Problem

I submit this longitudinal boiler joint to the readers with the idea that some of them will figure its efficiency and so engineers in other parts of the world will realize what the applicant for a second-class engineer's license is "up against" in Massachusetts.

This joint is supposed to be a locomotive joint, but we in Massachusetts call it a "preventive joint" as it was put in a single-riveted lap-jointed boiler to prevent the boiler going to the scrap heap.



REINFORCED BOILER LAP JOINT

The style of joint is single-riveted lap-seam with one cover strap. The dimensions are:

Plate thickness, 5/16 in.; tensile strength, 55,000 lb. per sq.in.; inside pitch, 2 1/16 in.; outside pitch, 4⅛ in.; rivets, ¾ in. diameter after driving; shearing strength, 42,000 lb. in single shear, 78,000 lb. in double. I took this up with our boiler inspector and we found the weakest part of this joint to be the three rivets in single shear, their strength being only 0.785 that of the solid plate.

The cover strap is pounded in a little snugger to the rivet in the shell lap than my sketch shows.

JOHN H ARMSTRONG.

Adams, Mass.

Effect of No Lubrication on High Speed Engine

An engineer at a coal-mine power plant neglected to open the feed valve on the lubricator of a 16x18-in. automatic engine running 225 r.p.m. and belted to a direct-current generator which furnished power to the mining machines. As his duties included the blacksmith work and repairs around the mine, no further attention was paid to the engine until it suddenly stopped shortly after noon, with the rocker-arm broken and the screws loose in the rocker-arm brackets. The broken rocker was replaced by one removed from an engine at another mine near-by, and the engine started. After running a few minutes it was necessary to stop as the bracket could not be kept in place.

The cylinder head and steam-chest covers were then removed and the valve and seat were found so badly cut and worn that a new cylinder, piston and valve were needed before the engine was again ready for service. These parts were removed from the engine at the other plant and after being placed in position the engine ran without trouble. The damaged engine had been in service only two years and was purchased new.

C. C. SIDINGER.

Salem, Ohio.

Zinc Paint as Corrosion Preventive

The drums of our water-tube boilers showed signs of rapid corrosion, due undoubtedly to the presence of sulphuric acid in the feed water. To stop this action we painted the interior of the drums with graphite. The results were not wholly satisfactory, so we painted the parts affected with zinc paint.

Although we have not been using this paint long, the results so far are very pleasing.

J. E. ASHBAUGH.

Clarksburg, Penn.

[If the feed water is highly corrosive, coating the drum with paint containing zinc may give good results while the paint lasts, which will not be long. Zinc slabs hung in the boiler would give better results with less work on the part of the boiler attendant.—EDITOR.]

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Equalizing Cutoff Throughout Governor Range

Mr. Low's article on the Corliss engine in the issue of Sept. 10 was indeed interesting and instructive.

On page 379, he suggests that some means of equalizing the cutoff on the two ends of the cylinder throughout the range of the governor should be devised.

About ten years ago I was trouble man for a firm building four-valve and Corliss engines and had a call to a 30x42-in. noncondensing Corliss engine, which was giving trouble with the dashpots, among other things. If the cutoff was equalized for about quarter stroke, at no load the head end got practically no steam, and the dashpot would not handle the valve under those conditions without special adjustment which was unsatisfactory at normal load.

The knock-off cams were arranged similar to Fig. 1. To try the action of the valve-gear with the cutoff equalized, or nearly so, from lead to about one-third stroke, I moved the knock-off steel on the crank end so that the cams stood similar to Fig. 2, when cutting off at the same point as represented by Fig. 1, which was for no load. As will be noticed, the lever arm *A*, Fig. 2, is considerably shorter than when the cam stood as in Fig. 1, and on this account, when the governor reach-rod moves in the direction of the arrow for a longer cutoff, the crank-end cam will move through a greater angle than the head end and thus tend to correct for the angularity of the connecting-rod.

The builders adopted the idea at once and determined in the drafting room the proper ratio between the cam arms, and sent a new cam with a longer arm for the head end, as that was more easily changed, and the crank-end cam was put back to the original position. This arrangement was adopted for all engines and it worked closely throughout the range of the governor.

The disadvantage of this method is that the head-end valve will have a smaller opening than the crank-end valve for a given point of cutoff, especially if it is late, and the higher velocity of the piston combined with the smaller opening will tend to show more wire-drawing on the head-end indicator diagram than on the crank end. To overcome this I proposed to make both knock-off cam-arms the same length and either make the head-end hook lever shorter or the wrist-plate radius longer to give practically the same valve opening and cutoff on the two ends throughout the range of the governor.

The builders preferred to keep the heavy parts of the valve-gear symmetrical rather than take advantage of the better admission line on the head end and so adopted the unequal length knock-off cam levers as the simplest means of equalizing the cutoff. Changing the cams did not cure the dashpot trouble entirely, but helped appreciably.

E. P. HAINES.

Philadelphia, Penn.

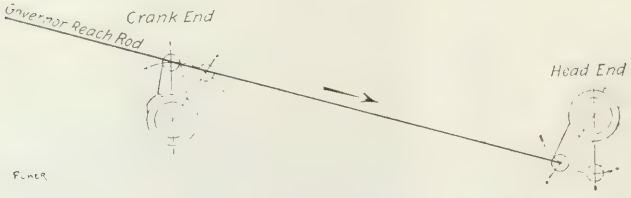


FIG. 1. REACH ROD AND VALVE-ARMS AS ORIGINALLY SET

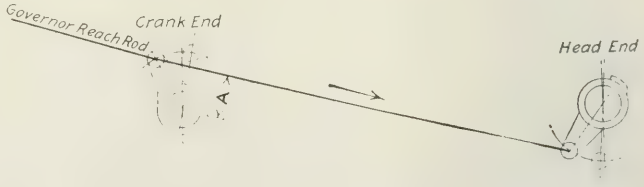


FIG. 2. SETTING TO EQUALIZE CUTOFF THROUGHOUT GOVERNOR RANGE

Zinc Slabs as Corrosion Preventives

Mr. Stafford, who inquires, in the Oct. 1 issue, as to hanging zinc slabs in boilers to prevent corrosion, should locate them so as to have a good electrical contact between the zinc and iron, and have the zinc where scale or deposit will not cover it. The idea is to make an electric couple out of the zinc and iron, and hence corrode the zinc rather than the boiler. It follows that as the zinc is effective only insofar as it is itself eaten away, that it will not last forever, but must be renewed, that many thin plates located at various points are better than one large chunk of twice their weight, and that any actual free acidity of the feed water should be neutralized with soda. Some sugar-plantation engineers using bad feed water report the zinc plates as very effective. Some of these plates should be anchored near the feed-water inlet.

DONALD M. LIDDELL.

Elizabeth, N. J.

Care of Baffles in Boilers

Mr. Wing's remarks as to the importance of caring for the baffling in water-tube boilers, in the Sept. 3 issue, I wish to corroborate. The least one can do is to replace the absent bricks and even then the baffling often is not put in good condition. owing to the springing apart of the tubes.

While the holes caused by such breaks are not large, they are many. If the surface of the brick is fused, it helps to close the holes, in which case it would seem unwise to use a spreader, as spreading will reopen them. Trimming the corners of new brick with a half-round file, striking a glancing blow does not weaken the brick and conduce to disintegration later on as is the case when unskillfully trimmed. Little trimming is needed to make a brick turn into place.

From long service baffles become unable to resist a crushing stress but are still tight as baffles. To spread the tubes to replace one brick might therefore crush several adjoining ones. There is a time when it is best to renew a whole set of baffles, but experience leads me to defer that as long as it is possible to patch the old, as rebaffling leaves so many holes. My experience has been mostly with double-deck boilers, when most of the holes have to be patched by applying the refractory material with the end of a stick 3 ft. long, although with the proper tools a new brick can easily be put in.

Another evil of rebaffling is that the tube bricks that are partly set into the side walls are sometimes broken out or disturbed, making patching at long range very difficult, especially as the flame

plates do not always extend near enough to the side wall to afford backing for the plastic material that must be used in patching.

Another place to watch in double-deck boilers is the space between the tube nests. The bricks used here being thin and light for their flat area, fall down, due probably to using steam-blowing apparatus for cleaning the tubes, the flame plates not being close enough to prevent the steam knocking them down.

I remedy this by laying a platform of baffle brick on top of the tubes and laying on this ordinary firebrick and refilling the space. This is done by a thin-chested fellow, who can crawl in, or by using boards on which to slide the brick in and sticks to guide them into place. This requires a hole to be made through the side wall.

All these things and the fact that some of the openings can be stopped only with difficulty out of all proportion to their size, has convinced me that no baffles are as tight as they should be. I find I am not alone in this belief as I have heard of forms being fitted to the tubes leaving a space into which the refractory material is run around the tubes, but with what success I do not know. It shows, however, that someone wanted a better baffle.

If it was my job to keep the baffling in order, I should not be so afraid of the chief crawling into the setting. as of his looking into the cleaning holes when the flames flowing through show where the holes exist, even revealing defects in the back baffles if the draft is strong. Looking into the holes is interesting to me aside from what can be learned. Through the top hole of the middle pass when the damper is closed and not too soon after coaling, the slow burning of the gases gives a peculiar vivid glow. When the damper opens and the gases take on motion and roll up, over and down, filling the whole space and beating against the bridge-wall, a very avalanche of flame, it is a grand sight to see and teaches the engineer that it is inexcusable to neglect the baffles and walls when the best that can be done is to keep the soot out and the setting in the best possible condition.

A. N. BOGART.

New York City.

Emergency Repair to Air Compressor

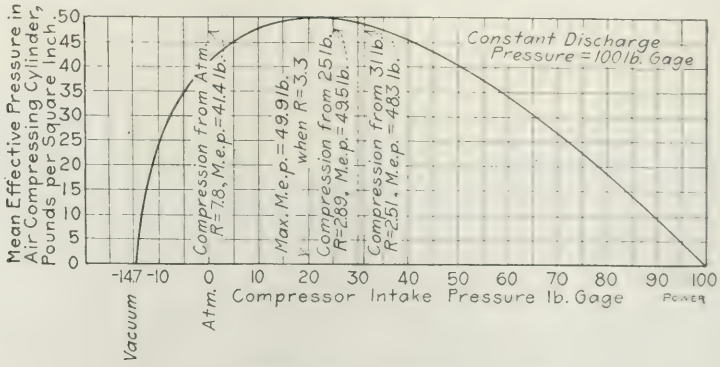
W. M. Fleming's letter on this subject, on page 431, Sept. 17 issue, brings out some curious facts about the compound or stage compression of air.

No doubt those who coupled up the 10x10-in. single-stage machine to act as the high-pressure side for the 20x8-in. low-pressure cylinder were not aware that when they caused the 10x10-in. machine to pump air from 25 lb. to 100 lb. they were subjecting it to a 20 per cent. overload, if designed for pumping from atmosphere to 100 lb. as it probably was.

The reason for the curious fact of increased horsepower when pumping through the smaller range of pressure is inadvertently given in Mr. Fleming's last paragraph where he says that the indicator cards of the 10x10-in. machine showed much more air compressed when pumping from 25 lb. pressure than when pumping from atmosphere. As a matter of fact it was handling more air; just about

$$\frac{25 + 14.7}{14.7} = \frac{39.7}{14.7} = 2.7$$

times as much. I say "about," because the decrease in compression ratio would materially increase the volumetric efficiency of the 10x10-in. machine and it would really be handling somewhat more than 2.7 times its usual atmospheric intake capacity.



INTAKE-DISCHARGE PRESSURE CURVE

This increased quantity of air pumped naturally takes more horsepower to compress it, hence the overload of 20 per cent. in power required. This horsepower increase is nowhere near the air capacity increase, and there was a considerable net gain. The fact that the 10x10-in. machine was able to stand the overload for continuous running speaks well for the design. I am wondering if some of the bearings did not heat.

Mr. Fleming says they adjusted the unloader on the 20x8-in. cylinder, to bring the intercooler pressure to 25 lb. This probably was done because the original intercooler pressure of the 20 and 12 by 8-in. machine was about this point and any increase would overload the 20x8-in. cylinder and its bearings. If, however, the 10x10-in. machine successfully withstood the 20 per cent. overload, the 20x8-in. might stand some, too, and by not closing the regulator, or unloader, to such an extent they might pump still more air.

In pumping from atmosphere to 25 lb. gage, the mean effective pressure is close to 16.98 lb. per sq.in. If now they had opened the regulator or unloader a little, allowing the intercooler pressure to rise to 31 lb. gage, the mean effective pressure in the 20x8-in. cylinder would have been about 19.82 lb. This is a 16.7 per cent. overload above 25 lb. operation. The mean effective pressure in the 10x10-in. cylinder pumping from atmosphere to 100 lb. gage, is close to 41.4 lb., and when pumping from 31 lb. to 100 lb., the mean effective pressure in this cylinder would be about 48.3 lb., which is about 16.7 per cent. overload above 41.4, the usual mean effective pressure for which the machine probably was designed. Thus with 31 lb. gage in the intercooler we have 16.7 per cent. overload in the 20x8-in. cylinder and 16 per cent. overload in the 10x10-in. cylinder, the same in the two cases, and we know already that the 10x10-in. machine is good for at least 20 per cent. overload.

The advantage of this is that by decreasing the overload in the high-pressure 10x10-in. compressor from 20 to 16.7 per cent. and putting 16.7 per cent. overload on the 20x8-in. cylinder, there would have been pumped about

$$\frac{31 + 14.7}{25 + 14.7} = \frac{45.7}{39.7} = 15 \text{ per cent.}$$

more air.

Thus, if they had needed the air, balancing the overload to about 17 per cent. on each machine would have obtained about 15 per cent. more air than when simply overloading one machine 20 per cent., as they did. No doubt they had enough air, as I believe Mr. Fleming says, but these facts, if understood, might have helped them if more air was needed.

Many may not see why increasing the intake pressure of the 10x10-in. compressor from atmosphere to 25 lb. gage causes a 20 per cent. overload on this machine, and a further increase of the intake pressure from 25 to 31 lb. will reduce the overload to 16.7 per cent. The fact is that with a given discharge pressure, an increase in the intake pressure will produce an increase in the mean effective pressure and horsepower, until the compression ratio reaches 3.3; that is, when the absolute discharge pressure is 3.3 times the absolute intake

pressure. For 100 lb. gage discharge, this point occurs when the intake pressure is

$$\frac{114.7}{3.3} = 34.75 \text{ lb. abs.}$$

or about 20.06 lb. gage. Above this intake pressure the mean effective pressure decreases again, as may be realized by observing that with 100 lb. intake and 100 lb. discharge no compression occurs and no work is done; there is no mean effective pressure. This may be seen in the accompanying diagram of mean effective pressure with 100 lb. gage discharge and various intake pressures from vacuum to 100 lb. Zero mean effective pressure occurs at two points, when intake is at absolute vacuum, because the cylinders would then have no air to pump, and when the intake pressure is equal to the discharge, as before stated. This occurs with a dry-vacuum pump also, and with 30-in. vacuum, at sea level, the pump would develop no horsepower. It would also develop no horsepower if the vacuum were completely broken. The maximum indicated horsepower occurs at ratio 3.3 as before, which would be at about 20.8 in. of mercury.

With the air compressors under discussion, when the intake of the 10x10-in. machine was made, 25 lb., the mean effective pressure, although 20 per cent. greater than when pumping from atmosphere, was less than the possible maximum occurring at 20.06 lb., and a further increase to 31 lb. intake pressure would actually decrease the mean effective pressure and overload on this compressor, although it would equally overload the 20x8-in. machine.

This problem of variable intake pressure considerably complicates the work of pumping natural gas, which often occurs at high well pressure and is subject to considerable variation, with ultimate decrease as pumping continues, sometimes resulting in increased horsepower with reduced capacity and sometimes a decrease of both. It depends upon the compression ratios.

Upon again reading over Mr. Fleming's article I find a peculiar situation. He says the 10x10-in. compressor was a duplex (two-cylinder) machine. Assuming that both the 8-in. and 10-in. machines ran at the same speed (no speeds being specified), the cylinder ratio produced by pumping from one 20x8-in. cylinder into two 10x10-in. cylinders would be as their piston displacements, or about:

$$\frac{20^2 \times 8}{2 \times 10^2 \times 10} = 1.6$$

Assuming also that the broken intercooler tube was plugged and the cooler working, the intercooler pressure produced should have been about

$$1.6 \times 14.7 = 23.5 \text{ lb. abs.}$$

or 8.8 lb., say 10 lb. gage if the cooling effect were poor.

If the 20x8-in. machine ran at a greater speed than the 10x10-in., say in the inverse ratio of their strokes, the equivalent cylinder ratio produced would have been about

$$\frac{20^2}{2 \times 10^2} = 2$$

This would produce an intercooler pressure of about

$$2 \times 14.7 = 29.4 \text{ lb. abs.}$$

or 14.7 gage, say 16.5 gage for poor cooling.

These figures are based upon using the entire output of the 20x8-in. cylinder, and if, as Mr. Fleming says, they throttled the intake to this cylinder, its capacity would have been decreased and the intercooler pressure further lowered. How they obtained 25 lb. gage in the intercooler is a mystery to me. He does say they adjusted the unloader of the 10x10-in. machine, but choking the intake to this machine would have resulted only in piling up the pressure in the 20x8-in. cylinder, only to be reduced again to the pressures already figured—either 10 or 16.5 lb.—at the intake to the 10x10-in. cylinders. No amount of throttling between stages can affect the intake pressure to the second stage short of absolute closure, and the pressure will accommodate itself to do this independently of throttling between stages. This pressure depends only upon the ratios of cylinder volumes and the temperatures. The high-pressure cylinder must handle all the low-pressure cylinder gives it. If they throttled the second-stage intake, why did they throttle the 20x8-in. intake? These two operations would have opposite effects.

In view of these facts and that Mr. Fleming definitely states that they throttled the 20x8-in. intake, I think he must mean they entirely closed the suction of one of the 10x10-in. cylinders (or otherwise stopped its action) when they "adjusted the pressure regulator" on that machine. That would give a ratio of

$$\frac{20^2}{10^2} = 4$$

(assuming equal piston speeds), and this would produce an intercooler pressure of

$$4 \times 14.7 = 58.8 \text{ lb. abs.}$$

or 44.1 gage, say about 46 lb. for poor cooling. Then by throttling the intake to the 20x8-in. cylinder, as he says, they could have brought this pressure down to 25 or 31 lb., and obtained the results in horsepower and capacity described. Even then it would have been better, if they needed the air, to use both 10x10-in. cylinders and the entire unthrottled capacity of the 20x8-in. cylinder, letting the cooler pressure come where it would, 10 lb., or 16 lb., or 20 lb., depending upon the actual relative speeds of the two machines. This also would have added only about the same overload as before to the 10x10-in. machine (see the curve) and would have greatly reduced the power of the 20x8-in. cylinder by the reduction of its discharge pressure. Balancing the work in the two stages results in minimum horsepower, but when the balance is obtained by throttling the intake there is still another loss, as a partially closed intake controller, while it does reduce the air capacity, it scarcely reduces the horsepower at all until almost entirely closed.

I think the 10x10-in. machine had but one cylinder and was not a duplex. At any rate the problem is of unusual interest to figure upon in the light of the facts set forth in the accompanying curve.

R. S. BAYARD.

New York City.

The Man Who Wins

Through its forewords and editorials POWER always seems to be trying to help the engineer better his conditions by close attention to business, and study of the books that teach the ways of overcoming the difficult problems that are encountered in this work. A few engineers as well as other employees still have the distorted idea that they should do no more than they think they are being paid for, and who wonder why their salaries are not increased as fast as that of their neighbor, who is always trying to do what he can to cut down the cost of operation a little, or to make himself worth more to his employer.

Most employers, when it comes to paying out good money for wages are from Missouri, and have to be shown that the engineer is worth the money. The man who gets the largest salary is usually the one who is trying to reach the top of his chosen profession whatever it may be, and who labors hard at his work instead of trailing along a week or two ahead of the yellow discharge ticket.

Recently a friend sent me a post card on which the following very pertinent verses were printed:

The man who wins is an average man,
Not built on any particular plan,
Not blessed with any particular luck,
Just steady and earnest and full of pluck.

When asked a question he does not guess—
He knows, and answers "no" or "yes."
When set to a task the rest can't do,
He buckles down till he's put it through.

Three things he's learned; that the man who tries
Finds favor in his employer's eyes,
That it pays to know more than one thing well,
That it doesn't pay all he knows to tell.

So he works and studies till one fine day
There's a better job with bigger pay,
And the men who shirked whenever they could
Are bossed by the man whose work made good.

For the man who wins is the man who works
Who neither labor nor trouble shirks,
Who uses his hands, his head, his eyes.
The, man who wins is the man who tries.

J. C. HAWKINS

Hyattsville, Md.

Our Repudiation

The subjoined correspondence explains itself. Let the American Carbonine Co. appoint an engineer, we will appoint another; these two to select a third. Let the three engineers so selected make an exhaustive test of the value of carbonine.

If by the use of carbonine a given quantity of coal can be made to evaporate 20 per cent. more water than can be evaporated under the same conditions of temperature and pressure with the coal untreated, we will pay the entire expenses of the test and acknowledge that we were mistaken and did carbonine a serious injustice when we published the correspondence and the comment herewith reprintedin the left-hand column below.

If, upon test, the coal treated with carbonine fails to evaporate 20 per cent. more water per unit of weight than can be evaporated under the same conditions of temperature and pressure with the same coal untreated, the American Carbonine Co. shall pay the entire expense of the test and sign an acknowledgment to be printed in the pages of POWER that the attitude of incredulity assumed by this paper in the treatment of carbonine reproduced below was wise and warranted.

Each party to the agreement shall furnish a bond acceptable in amount and terms to the board of experts conditioned upon the faithful carrying out of his portion of the contract.

If carbonine or anything else will save 20 to 30 per cent. of coal and abate the smoke nuisance at the same time, it is worth spending money to find it out.

If carbonine is only another of the various nostrums which have been bobbing up from time to time for the past 20 or more years for the same purpose, exploited in the same way, and failing to sustain the test of real investigation and continued use, it is right that the public should know it.

Editor POWER:

On page 266 of POWER for Feb. 20, there is an editorial headed "Get Rich Quick Schemes," the latter part of which tells about "Carbonine" and how the use of that same powder mixed with water and sprinkled on the coal is supposed to produce a saving of 25 to 26 per cent. in the coal bill. Salem is not the only city in Massachusetts where the promoters of Carbonine have caught some victims, as Fall River has two or three plants which its promoters claim are using Carbonine and for which they claim a saving of 26 per cent. in the coal pile.

It seems that the right way to use Carbonine is to buy one ton, which costs $90. Then take one pound of Carbonine powder and mix it with one gallon of water (hot or cold) it makes no difference. Then to each ton of coal sprinkle eight gallons of this mixture and then you save 26 per cent. of your coal bill. One of the promotors was asked how it was figured out that adding 67 lb. of water to a ton of coal aided the combustion of the coal, and he answered that he did not know. "You could use Carbonine without water," he said, but adding water made the mixture go farther. You could get the same results by using it dry. The writer agrees with him, and also believes he can get the same results by leaving out the Carbonine, ...

Fall River, Mass. MICHAEL H. HARRINGTON.

[The sample accompanying the letter above was sent without comment to Dr. C. E. Lucke, of Columbia University, for analysis, and by him submitted to a firm of industrial chemists whose report is reproduced herewith.—EDITOR.]

M C. WHITAKER, M Sc F J METZGER, Ph D.

CONSULTING
INDUSTRIAL AND ENGINEERING
CHEMISTS

BROADWAY AND 116TH STREET
NEW YORK CITY

June 11, 1912.

Prof. C. E. Lucke,
Columbia University.

Dear Dr. Lucke:

We have made a qualitative analysis of the sample of material forwarded on June 5th with your letter from Power. We find that this material is practically all common salt mixed with some potassium permanganate and some zinc compound. We also note small quantities of carbonaceous material and a little sulphate.

The stuff looks to us very much like some of the fake ash burning compounds which are put out and which we have had occasion to examine before.

Hoping that you will find this satisfactory,

Very truly yours,

M. C. Whitaker

AMERICAN CARBONINE CO.

CARBONINE THE COAL ECONOMIZER

170 SUMMER STREET

BOSTON, MASS.

October 22, 1912.

Mr.F.R.Low
Editor Power Magazine
505 Pearl Street
New York, N.Y.

Dear Sir:-

It has just been brought to my attention,the article regarding Carbonine in your issue of July 9th 1912, signed by a Michael H.Harrington,Fall River,Mass.

It is very evident that you have not looked into this to any great extent. Now in the first place I want to set matters right.

The American Carbonine Co.Inc.,is not a get rich quick scheme,nor a fake. Carbonine is not a fad nor an experiment. It has been used in Canada for more than two years, would refer you to advertisement appearing in the Canadian Manufacturer,September issue 1912.

Carbonine is sold on its merits. We do not claim it is a substitute for coal,neither do we claim any heat units in Carbonine alone. We do claim this,when combined with coal, "Carbonine" a catalytic chemical which has an affinity for coal, which affiliates with it,and facilitates its combustion,consuming the volatile gases at the proper ignition point, utilizing them as heat efficiency instead of allowing them to pass out un-burnt and un-consumed as un-altered hydro-carbon and carbon monoxide.

It produces a practically smokeless combustion,saving from 20 to 30% in the amount of coal required for the production of a given result.

The so-called coal experts report is incorrect. There have been about forty chemists analyze Carbonine,and their reports vary, yet Carbonine is made the same to-day as it was nearly three years ago.

You undoubtedly know that the Pears Soap people offer a considerable sum of money to anyone who will tell them exactly the ingredients in Pears Soap. It is an undisputed fact that there are chemicals that cannot be analyzed. The important question is not what or how an article is made,but the results it produces, the practical is what the public demand. A thing should not be condemned until a fair trial or test is given it. Sometimes people are sore-heads, so called, because they cannot have their own way.

Carbonine is an article that will stand alone on its merits,if given a fair and practical test,this had to be proven before patent rights could be secured. We want you to repudiate the incorrect statements in the issue of July 9th in "Power" or else prove the statements.

Yours very truly,
American Carbonine Co.
W. L. Dunham
Treas.

WLD/I

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Deflection of Line Shafting

What amount of deflection is allowable for line shafts?

W. H.

For continuous line shafting the bearings should be spaced near enough together so that the deflection will not exceed 0.01 in. per ft. length of shafting.

Danger with Siphon Condenser

Is there any danger of getting water in the cylinder of an engine with a siphon condenser?

S. M. O.

Yes; from stoppage of the tail pipe or a reduction of the effective head of the tail water, to less than 34 ft., which might happen from air backing up in the tail pipe.

Slip of Pump

What is meant by the percentage of slip of a pump?

P. M.

The difference between the volume of water-piston displacement and volume of water actually pumped is the slip of the pump; and the per cent. which this difference is of the volume of water-piston displacement is the percentage of slip.

Flywheels for Jackshaft

In driving a belted electric generator from an engine jackshaft which has unsteady speed, due to lightness of the engine flywheel, can the speed of the generator be made steadier by placing a flywheel on the jackshaft?

W. H. S.

Uniformity of the jackshaft speed would be improved by any addition of flywheel weight within the limits of disturbance of speed that might be introduced by creating an unsteadiness of the engine belt.

Tobin Bronze

What is the composition and tensile strength of Tobin bronze?

J. B.

The composition is usually about as follows:

	Per cent.
Copper	59
Zinc	38.4
Tin	2.16
Iron	0.11
Lead	0.31

As cast metal it has a tensile strength of about 65,000 lb. per sq.in. of sectional area; when rolled, about 79,000 lb.; and when drawn down by cold rolling a tensile strength of over 100,000 lb. per sq.in.

Loss of Heat by Steam Pipe

With the average temperature of a boiler room at 85 deg. F., what is the loss of heat per running foot of bare 7-in. iron steam pipe containing steam at 110 lb. boiler pressure?

A. M. W.

The rate of transfer of heat from bare wrought-iron or steel steam pipe is about 2.7 B.t.u. per sq.ft. of radiating surface per hour, for each degree Fahrenheit difference of temperature between the steam inside and air outside of the pipe. Steam at 110 lb. boiler pressure would have a temperature of 344.4 deg. F. and as the difference of temperature would be

$$344.4 - 85 = 259.4 \text{ deg. F.}$$

the loss would be

$259.4 \times 2.7 = 700.38$ *B.t.u. per sq.ft.* of external pipe surface per hour. The outside diameter of 7-in. pipe being 7.625 in. its circumference is 23.94+ in. and the external area

$$(23.94 \times 12) \div 144 = 1.995 \text{ sq.ft. per running foot}$$

The loss is therefore, about

$$700.38 \times 1.995 = 1397.2$$

or in round figures, 1400 B.t.u. per hr. per running foot of pipe.

Air Bound Suction Pipes

What causes a suction pipe to become air bound and how can it be prevented?

A. K.

Water generally contains air or other gas in solution at atmospheric pressure. When the pressure is relieved the gases, which are held in solution or in suspension are liberated in small bubbles, as may be observed in the boiling action of water when placed in a vessel under the glass receiver of an air pump. In like manner small bubbles of air are liberated from the water of a pump suction pipe and attaching themselves to the walls of the pipe they are not readily disengaged from cavities and rough spots of the pipe surface, and increase in size by contributions of air from additional water flowing through the pipe, often resulting in complete stoppage of the pipe area. Air bubbles are likely to be thus formed in any siphon or pump suction pipe, but can generally be dislodged by sharp taps of a hammer and discharged through petcocks placed at the highest points of the pipe line. The best method of preventing the liberation of air is to employ large size suction pipes and pumps large enough to be operated at a slow rate of piston speed.

Coal Required

How many pounds of coal will be required to raise the temperature of 200,000 lb. of water from 50 deg. F. to 180 deg. F. the coal having a calorific value of 13,000 B.t.u. per lb.?

E. S.

To raise the water from 50 deg. F. to 180 deg. F. would require, approximately

$$200{,}000 \times (180 - 50) = 26{,}000{,}000 \text{ B.t.u.}$$

Allowing a boiler efficiency of 70 per cent. the heat received from combustion of each pound of coal would be 70 per cent. of 13,000 = 9100 B.t.u.; hence the coal required would be

$$\frac{26{,}000{,}000}{9100} = 2857+ \text{ lb.}$$

Power Absorbed by Friction of Shafting

What is the formula for the number of horsepower absorbed by the friction of shafting?

J. G.

The average coefficient of friction of iron or steel shafting in horizontal bearings is about 0.07 for ordinary oiling and about 0.043 for continuous oiling. Where d = the diameter of the journal in inches, the space passed over for one turn would be

$$\frac{3.1416 \times d}{12} = 0.26\, d \text{ ft.}$$

hence for each journal bearing where

W = Total weight of shafting and pulleys, plus the resultant stress of belts, in pounds;

d = Diameter of the journal in inches;

T = Number of turns per minute;

h = Number of horsepower absorbed, then for ordinary oiling the formula is

$$h = \frac{0.07 \times W \times 0.26\, d \times T}{33{,}000} = \frac{W d T}{1{,}813{,}187}$$

and for continuous oiling

$$h = \frac{0.043 \times W \times 0{,}26\, d \times T}{33{,}000} = \frac{W d T}{2{,}951{,}699}$$

Engineers' Study Course

The promised Engineers' Study Course commences with this issue. The necessary mathematical foundation will be laid by a brief review of fractions, percentage, proportion, roots and powers, the use of logarithms, formulas and equations, mensuration, simple trigonometry, etc. These subjects will be carried along in an easily understandable way and be replete with engineering examples so that the student will learn not only how to read and understand mathematical expressions and to perform the computations arising in his work, but will gain much practical engineering knowledge in the process. The course will then take up elementary mechanics, simple chemistry, heat physics, boiler and engine testing, power plant economics, etc. As space permits and it proves practicable, treatment of the more advanced topics may be undertaken in the same issue with the more elementary lessons, that a wider range of readers may be interested and the beginner see where his studies are leading him.

As the course progresses the student will be referred to books, papers and articles which may be consulted for further knowledge upon the particular subject in hand.

In order that one may understand later lessons and perform the examples which will be given in this course, he must have a working knowledge of arithmetic. It is to be presumed that anybody who will wish to follow it can already add, subtract, multiply and divide, but it may be well, even at the expense of keeping most of the class waiting, to brush up a little on fractions.

A shaft that was one whole piece, that was "integral" as they say, not made up of a lot of separate parts but a complete unity, breaks into two parts; *fractures*.

The safety valve sticks and a boiler is blown into *fragments*.

These words, signifying breaking up, dividing, come from the same Latin word as "fraction" which means also, when we use it to represent an action, the dividing or breaking of a thing into parts, and when we use it to represent a thing, one of the parts into which the object is divided.

In Fig. 1 is a circle divided into 12 equal parts. Each of these parts is "one-twelfth" or $\frac{1}{12}$ of the whole circle. If we want to specify how much of the circle is shaded we see that five of the subdivisions are so treated and say that $\frac{5}{12}$ of the circle is shaded.

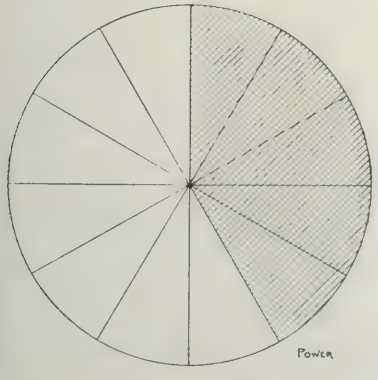


FIG. 1

This expression $\frac{5}{12}$ is a fraction. It expresses a part of something, a *fragment* of something, and tells what part this fragment is of the whole.

We have just been through the excitement of naming, or *nominating* candidates for the presidency. Denominate means a little more strictly to call by name, to classify, to specify. What *denomination* does he belong to? When you draw money from the bank you tell the teller whether you want it in bills of large or small *denomination*. The figure under the line is called the "denominator" because it names, specifies, tells, denominates what kind of parts we are dealing with, whether they are halves or quarters or $\frac{1}{8}\frac{1}{0}$ths or what. It is the number of parts into which the whole object, the "integer," may be assumed to be divided in order that by specifying a certain number of these parts we may express the quantity desired. When we said above that $\frac{5}{12}$ of the circle was shaded, what we virtually said was that if the circle were divided into 12 equal parts, the part which was shaded would equal five of these.

The number above the line (5 in this instance) is the *numerator* because it numbers, counts off or numerates how many of these parts are to be taken.

The expression $36 \div 12 = 3$ means that 36 divided by 12 equals 3.

Put the numbers themselves in place of the dots over and below the line, placing the number to be divided on top, and you will have a simpler expression which means the same thing, thus

$$\frac{36}{12} = 3$$

A fraction then is simply an expression of division, and means that the numerator is to be divided by the denominator.

When the numerator is less than the denominator the quotient will be less than one, as in

$$\frac{\text{numerator}}{\text{denominator}} = \frac{5}{12} =$$

Here the quotient of 5 divided by 12 is less than *one*. It is only that part of one that the shaded part of the little circle after the fraction is of the whole circle. The expression therefore signifies a fragmentary, a fractional, part. It is properly a fraction and is called a "proper fraction."

When the numerator is greater than the denominator, the quotient will be greater than one, as

$$\frac{36}{12} = 3 \qquad \frac{25}{6} = 4\frac{1}{6}$$

Divisions expressed in this way in which the numerators are greater than the denominators, are called "improper" fractions, which, however, is quite a different thing from "vulgar" fractions, which means fractions written with a numerator and denominator as $\frac{1}{2}$, as opposed to decimal fractions whose denominators are implied by the position of the decimal point and must be multiples of 10. The vulgar fraction, notwithstanding its vulgarity or commonness, can express any known quantity with absolute precision, while the decimal fraction can express it exactly only when its denominator will divide its numerator without a remainder when sufficient ciphers are added to the numerator. So it comes about that the common or vulgar fraction is used for the most refined calculations, and its pretentious relative, the decimal, is discredited altogether when it comes to the higher mathematics.

I have given you two conceptions of a fraction, the first when I cut a circle up into 12 equal parts and called this number of parts the denominator and then took a certain number of these parts and called this number the numerator; and the other when I said that it was a sign or expression of division; a direction to divide the numerator by the denominator. Just to show that it makes no difference which way we look at it, here, Fig. 2, is a foot rule divided into inches or 12ths. We want to express or specify the part of this foot which is occupied by three of these inches.

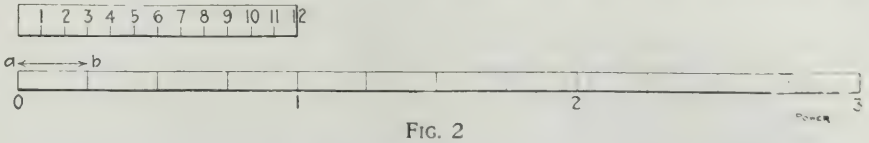


FIG. 2

Well, in one way we conceive of the rule being divided into 12 equal parts, and 12 is the denominator, and we are

talking about three of these parts, so 3 becomes the numerator and the expression becomes $\frac{3}{12}$ or the length *ab*.

In the other conception of the fraction, that of an expression of division, we take three foot rules or a three-foot rule and divide this into 12 equal parts, as in the lower part of the same figure. The result *ab* is the same as before. In one case we divide a 1-ft. rule into 12 parts, and multiply by 3 (i.e., we take one part three times) and in the other case we multiply by the 3 first and then divide the product by 12. In both cases the numerator is the multiplier and the denominator the divisor.

Remember that! Quantities above the line are multipliers; quantities below the line, divisors.

Mixed numbers are those consisting of a whole number or integer and a fraction as $3\frac{1}{2}$; $68\frac{3}{4}$; $79\frac{11}{12}$.

We must first learn to express these mixed numbers as improper fractions. It is very simple.

To Reduce a Mixed Number to an Improper Fraction

Rule: *Multiply the integer or whole number by the denominator of the fraction.*

To the product add the numerator of the fraction, and write the sum as a numerator over the denominator.

Example: Express $3\frac{1}{2}$ as an improper fraction.

Multiply the integer (3) by the denominator (2)

$$3 \times 2 = 6$$

add the numerator 1

$$7$$

Write the sum (7) over the denominator (2), then

$$3\tfrac{1}{2} = \tfrac{7}{2}$$

Example: Reduce $329\frac{19}{64}$ to an improper fraction.

329 integer
64 denominator
1316
1974
21056 integer × denominator
19 numerator
ans. $= \frac{21075}{64}$ int. × denom. + nu. / denominator

One addition may be saved by adding the numerator right in with the addition for the multiplication, thus

329
64
1316
1974
19
21075
64

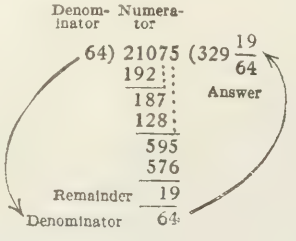
To Reduce an Improper Fraction to a Mixed Number

Rule: *Divide the numerator by the denominator for the integer part and write the remainder over the denominator for the fractional part.*

Example: Express $\frac{7}{2}$ as a mixed number.

From a group of seven articles we may take two away three times and there will be still one left. This is what is meant when it is said that 2 will "go into" 7 three times and leave one as a remainder. We write the quotient 3 as the whole number, and write the remainder (1) over the denominator (2) for the fractional part, giving $3\frac{1}{2}$ for the answer.

Example: Express $\frac{21075}{64}$ as a mixed number.



Examples for Practice

1. The length of an indicator diagram is $3\frac{19}{64}$ in. What is its length in 64ths?

2. A pump rod is $\frac{35}{16}$ in. in diameter. What is its diameter in inches and 16ths?

3. A boiler tube contains 1680 sq.in. of heating surface. Since there are 144 sq.in. in a sq.ft., this will equal $\frac{1680}{144}$ sq.ft. Reduce this to a mixed number.

4. A vessel contains 4275 cu.in. of water. There are 231 cu.in. in a gallon. How many gallons of water does the vessel contain?

5. An engine uses 6000 lb. of steam per hour and develops an average of 324 hp. How many pounds of steam does it use per horsepower-hour?

6. A boiler evaporates $8\frac{18}{37}$ lb. of water per pound of coal. Express this as an improper fraction.

7. What mathematical operations are indicated by the expressions:

$$\frac{12}{4} \qquad \frac{circumference}{diameter} \qquad \frac{a}{b}$$

8. What is a whole number, an entire quantity, called?

9. In a fraction, what is the number above the line called?

10. In a fraction, what is the number below the line called?

11. Write the expression $21 \div 5$ in the form of a fraction.

12. In the expression $\frac{17 \times 4}{3}$, what is the numerator and what is the value of the expression written as a mixed number?

Study Question Answers

On account of the Engineers' Study Course, beginning in this issue, the Study Questions will be discontinued. The answers to the last five questions given in the Oct. 29 issue follow:

(121) The electromotive force of each group will be

$$5 \times 2 = 10 \text{ volts}$$

and the internal resistance

$$5 \times 0.04 = 0.2 \text{ ohm}$$

But since there are five groups in parallel the internal resistance of the battery is

$$\frac{0.2}{5} = 0.04 \text{ ohm}$$

Hence the total resistance, both internal and external, is

$$0.5 + 0.04 = 0.54 \text{ ohm}$$

Therefore the current in the circuit will be the total electromotive force divided by the total resistance, or

$$\frac{10}{0.54} = 18.5 \text{ amp.}$$

(122) 1 hp.-hr. = 1,980,000 ft.-lb. of energy. As

$$Energy = pressure \times volume = PV$$
$$1{,}980{,}000 = PV$$

As the volume is to be expressed in cubic feet, the pressure must be in pounds per square foot. Hence

$$P = (60 - 2)\ 144 = 8352 \text{ lb.}$$

Then

$$V = \frac{1{,}980{,}000}{8{,}352} = 237.07 \text{ cu.ft. per hr.}$$

Dividing this by the specific volume given

$$\frac{237.07}{7.037} = 33.69 \text{ lb.}$$

of steam used (or water evaporated) per horsepower per hour.

(123) Let

x = Time first outlet is open;
y = Time second outlet is open.

Then

$$x + y = 10 \text{ min.} \qquad (1)$$

Discharging 14 and 9 gal. per min. respectively

$14x$ = Quantity discharged by first outlet;
$9y$ = Quantity discharged by second outlet.

From the conditions of the problem

$$14x + 9y = 100 \text{ gal.} \qquad (2)$$

Multiplying equation (1) by 14

$$14x + 14y = 140$$

Subtracting equation (2) from this

$$5y = 40$$
$$y = 8$$

and substituting in (1)

$$x + 8 = 10$$
$$x = 2$$

(124) The length of the tank, head to head, will be 6 ft. minus 3 in., or

$$6 - \tfrac{1}{4} = 5\tfrac{3}{4} \text{ ft.}$$

The diameter will be that of a circle having a circumference of

$$6 - \tfrac{1}{4} = 5\tfrac{3}{4} \text{ ft.}$$

As the circumference of a circle equals the diameter multiplied by 3.1416

$$5.75 = 3.1416\ D$$

$$D = \frac{5.75}{3.1416} = 1.83\ ft.$$

The volume of the tank will be that of a cylinder of the dimensions now determined, for the reduction of volume due to the concave head is equal to the increase of volume due to the convex head. Hence the volume is the cross-sectional area multiplied by the length, or

$$0.7854 \times (1.83)^2 \times 5.75 = 15.124\ cu.ft.$$

(125) The uniform load of 800 lb. per ft. of length may be regarded as a concentrated load of

$$800 \times 24 = 19{,}200\ lb.$$

applied through the center of gravity, which will be the middle of the beam. Taking a summation of moments around the right support, its moment then will be

$$19{,}200 \times 12 = 230{,}400\ lb.\text{-}ft.$$

and the moments of the concentrated loads will be

$$5000 \times 24 = 120{,}000\ lb.\text{-}ft.$$

and

$$9000 \times 8 = 72{,}000\ lb.\text{-}ft.$$

The opposed resistance is that of the left support, call it X and its moment will be $X \times 20$. Equating

$$20X = 230{,}400 + 120{,}000 + 72{,}000$$

$$X = 21{,}120\ lb.$$

As the sum of the loads must equal the sum of the reactions

$$21{,}120 + Y = 5000 + 19{,}200 + 9000$$

$$Y = 12{,}080\ lb.$$

and is the load on the other support.

ERRATUM: In the answer to study question No. 95 in the issue of Sept. 24, a typographical error appeared, the figure 1 being lost before the 8 in the number of seconds for the second acute angle. The answer should have been 51 deg., 20 min., 18 sec.

Exhibition of Products "Made in Minneapolis"

The Civic and Commerce Association of Minneapolis, which is promoting "Made in Minneapolis" week, expects to have at least 500 exhibits of Minneapolis made goods in the business district. General Manager Pack, of the Minneapolis General Electric Co., has donated all the power needed for the operation of exhibits during the week, and in addition, all the lighting necessary. Arrangements have already been made for the electric operation of 100 working models of manufacturing processes on the streets and in the windows of the retail stores.

The company will have in its window an electric freight train in operation. On this train will be loaded various products of Minneapolis manufacturers.

Salaries of School Engineers in Chicago

The daily papers of Chicago recently published figures relating to the salaries paid to the public-school engineers in that city. The figures as given were exceedingly interesting and if correct would have indicated that the public-school engineers of Chicago were among the highest paid men to be found in the profession.

Investigation showed the papers had not told the whole story, however, for while the gross salaries received by some of the engineers are large, they must pay all their assistants out of the money they receive from the city.

The most spectacular case cited was that of the chief engineer of the Bowen High School, whose gross salary was reported as $9065.04 per year. His salary actually is $8465.04, out of which he must pay for the assistance required to take care of the heating plant, do all the sweeping and cleaning required throughout the building and care for the walks and the lawns which belong to the school.

The building contains some hundred rooms which, with the corridors, have 167,000 sq.ft. of floor surface to sweep and clean. The cleaning must be done in a specified manner and at specified intervals. Adjoining the building is a space of one city block used as a botanical garden, the care for which must be provided by the engineer. Thus it will be seen that the engineer is really a superintendent for the entire building and grounds and as such is responsible for maintaining them in good condition.

A fair estimate of the help required at the Bowen High School follows:

	Months per Year	Wages per Month	Wages per Year
One assistant engineer	12	$100	$1200
One fireman	12	80	960
One coal passer	8	70	560
One coal passer	3	70	210
One man for cleaning	10	70	700
One woman for cleaning	10	40	400
Four women for cleaning	10	35	1400
One man for walks and lawns	12	90	1080
Total			$6510

Thus, on a fair estimate the engineer must pay out $6510 per year to his assistants. This, from $8465.04 leaves $1955.04 as his compensation, which is not exceedingly liberal, considering his responsibility and that during some months in the year he must be at the school very early each morning to see that it is suitably heated in time.

Even assuming that by dint of hard manual work on his own part and by exercising good judgment he is able to reduce the above estimated expenses 10 or 15 per cent., his income is not unduly high when compared with the pay received by men in similar positions with private concerns.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Constable Joe Ash will be promoted for good detective work when our police force is increased to two, says the *Cinders Siftings*. Joe writes us: "I ketched Slicebar an' Hoe aplayin' poker over in the Clinkerton b'iler room an' jugged 'em. Ain't that grate?"

MANAGER—There's a man loafing back of the heater.

CHIEF—That's Brick, my assistant.

MANAGER—You fire Brick!

It is decidedly adventitious for Aleck Smart to ask who put the vent in ventilator. Why not interrogate us as to the identity of the individual who put the nick in our last nickel and made us walk home?

Bud Sykes was a meddlesome joker
Who worked with a shovel and poker.
One day he got gay—
(The chief was away)
He took up the engine and broke 'er.

Waxing eloquent, a contemporary remarks that "a microscopic eye is needed in the perpetual chase after the recalcitrant heat unit." We are immune from the "pink-eye" and the use of the capital "I," but this microscopic type is an eyebolt that holds us mentally suspended by our galluses over unknown terrors.

BUTT STRAP—Old man Rivet looks het up.

STAY BOLT—Ya-as: Rivet went into the boiler shop on business and was thrown into the fire. Made him hot under the collar. Then he got hit on the head several times and found himself in a hole. Rivet has cooled off quite some, but he's upset about it.

After two years' work, says the *Universal Engineer*, the Strawberry Tunnel, in Utah, has been pierced, and the paper further remarks that the valley to be irrigated is especially interesting because it was settled by Bingham Young. As friend Brig. was some "engineer" in his prime, it would be further interesting to know who settled Brigham.

Highbrow politicians are dipping into our engineering preserves for their metaphors. T. R. has been called a "switch-engine," and La Follette a "donkey-engine." Our water-power possibilities ought to appeal to the Prohibitionists.

N. A. S. E. Anniversary Banquet

The subordinate associations in Chicago of the National Association of Stationary Engineers held a stag banquet at the La Salle Hotel on Saturday evening, Oct. 26, to commemorate the thirtieth anniversary of the organization's birth and to honor the newly elected national president, John F. McGrath, who is a member of Illinois No. 28 of Chicago. Over 300 members and guests sat down to the feast. After the banquet, National Secretary Fred W. Raven introduced John W. Lane, editor of the *National Engineer*, as the toastmaster. James C. ("Dad") Beckerleg, the first national treasurer and the second national president of the organization, spoke reminiscently of the small beginnings of the association in New York City just 30 years ago. He said that its splendid growth and progress made proved that while men may come and men may go the association will live on and prosper, because it is founded on sound principles and is a force for good.

Col. Henry A. Allen, chief engineer of the Chicago water-works system, gave some interesting facts and figures concerning the development and present size of the system and in concluding extended an invitation to all present to visit the various pumping stations and offer any helpful criticisms of them.

Prof. William J. Bogan, of the Lane Technical School, spoke on modern educational problems. Although the first purpose of the public schools is to make good citizens, fully as important is the development of judgment and initiative. In industrial and agricultural America there is need today of the development of systems of education better suited to the requirements of the future business life of the children being educated. He believed that before great progress can be accomplished in this direction we must educate our educators.

Charles W. Naylor, chief engineer of the Marshall Field & Co. properties, spoke briefly.

James H. Harris paid a graceful tribute to the character and worth of the guest of honor, John F. McGrath.

Samuel B. Forse, national treasurer, spoke of his work of caring for the funds of the association and contrasted the large sums handled at the present time with the small amounts of which the first treasurer had custody.

The toastmaster then introduced National President McGrath, who bears the distinction of being a man who did not seek the honor bestowed upon him, but rather endeavored to avoid it. President McGrath spoke of the recent trips he has made in the interests of the association and stated that he was well pleased with the healthy, active and enthusiastic condition in which he found the numerous subordinate associations throughout the country.

He stated that he would do his utmost in furthering its work and policies and fervently hoped that his successor would find the organization in as good or even better condition when he came into office a year hence.

Interspersed with the speeches were numerous musical numbers, generously provided by the Ted Snyder Music Co. and Messrs. Fred S. Hickey, of the Dearborn Drug & Chemical Works, and Sam Rosenthal, of the Fred Busse Coal Co.

GUESTS AT THE BANQUET CELEBRATING THE THIRTIETH ANNIVERSARY OF THE N. A. S. E.

Through an error in making up the list of exhibitors at the recent national convention of the N. A. S. E. in Kansas City, the names of the C. Lee Cook Manufacturing Co., Louisville, Ky., and the Metallic Packing & Manufacturing Co., Elyria, Ohio, were omitted from the report in the Sept. 24 issue. We regret that this error should have occurred and take pleasure in making the correction.

Hartford Wins in Pabst Suit

The United States Circuit Court of Appeals at Chicago has decided in favor of the Hartford Steam Boiler Inspection & Insurance Co. in a suit brought against it about a year ago by the Pabst Brewing Co., of Milwaukee.

Several important questions were involved, chief among which was the brewing company's contention that the failure of each of the three boilers was a separate explosion, and that the insurance company was liable for the limit of the policy, $50,000, for each explosion. The brewery company also claimed that the insurance company's inspection was imperfect and it was therefore liable for the entire loss. The Circuit Court of Appeals reverses the decision of the trial court and finds that there was but one explosion; therefore the Hartford company's liability is limited to $50,000 as provided in the policy.

The evidence adduced by the Pabst company that as portions of boilers Nos. 5 and 6 did not explode and therefore proved the condition of boilers Nos. 1, 2 and 3, that did explode, was declared inadmissible. The court also found that if Feliber, Pabst's engineer, knew that the boilers were not in the same condition as at the insurance company's last inspection, Pabst should not have relied upon the Hartford inspector's reports, and that if cracks in boilers Nos. 1, 2 and 3 should have been discovered by Feliber, the jury should have found for the insurance company.

The insurance company paid the Pabst Brewing Co. the $50,000 it claimed it

owed under the policy and the brewery company instituted suit for the balance it claimed was due it.

Details of this explosion were given in POWER of Nov. 9, 1909, page 771; Nov. 23, 1909, page 875, and Feb. 8, 1910, page 285.

Webb City (Mo.) Mine Boiler Explosion

On Oct. 18, a boiler explosion at the Joe Lewis Mine, Webb City, Mo., at 10 a.m., killed a hoister man and a timberman. The boiler circled hundreds of feet above the ground, and fell through the roof of the Meinhart greenhouse.

Joseph Lewis, owner of the mine, who was standing by the smoke-stack, was hurled 30 ft., but received only a few bruises. Two miners, who were working 80 ft. down in the shaft, were taken from the mine by workers from the Robertson mine, adjacent to the Lewis shaft. Neither of the men was aware of the accident until they were taken from the ground and told what had occurred.

It is believed that an excessive pressure of steam caused the boiler to burst, but as there was no one in the boiler room at the time of the explosion the cause is not definitely known.

NEW PUBLICATIONS

THE PRINCIPLES OF PARALLEL PROJECTING-LINE DRAWING. By Alphonse A. Adler, B. S., M. E. D. Van Nostrand Co., New York City. Cloth; 66 pages; 6x9 in.; 48 illustrations. Price, $1.

While this is intended as Part I of the author's "Theory of Engineering Drawing" it possesses the merit of a work complete in itself on the construction of parallel projecting line drawings in oblique, orthographic and axonometric projection. The illustrations and examples are selected with good judgment for enlisting and holding the attention of the student, and the subjects are treated so clearly as to enable the reader to apply the principles for himself in drawing familiar objects, and to discover the road to more advanced knowledge of the subject.

The work, therefore, is a good one for guidance as a class textbook or for self-instruction in the fundamental principles of projection.

TABLES AND OTHER DATA FOR ENGINEERS AND BUSINESS MEN. Compiled by Chas. E. Ferris, B. S., professor of mechanical engineering, University of Tennessee. University Press, Knoxville, Tenn. Morocco; 170 pages, 2¾x5¾ in. Price, 50c.

The seventeenth edition of this little work makes its appearance in the form of a vest pocket-book containing tables of circles, weights of materials, trigonometrical functions, factors of evaporation and other tables and data which are in constant demand by a busy engineer. Power engineers will be especially interested in tables of sizes of boiler tubes, iron pipe, flow of steam through pipes, calorific value of fuels, forced-draft capacity table for blowers, horsepower of waterwheels, wiring tables and a table of equivalent values of electrical and mechanical units.

It is to be noted, however, that the steam tables and table of equivalent evaporation from and at 212 deg. F., are based on the old figures, 965.2 for latent heat of evaporation at atmospheric pressure in place of 970.4 of the Marks and Davis steam tables, which have been more generally adopted by recent authors.

The Coking of Coal at Low Temperatures

Bulletin No. 60 on "The Coking of Coal at Low Temperatures," by F. W. Parr and H. L. Olin, is now ready for distribution, and may be obtained from the University of Illinois, Engineering Experiment Station, Urbana, Ill., at a nominal charge of 25c. The investigations discussed in the bulletin had two general purposes in view: First to discover some fundamental facts pertaining to the properties and characteristics of bituminous coals, and second to determine the feasibility of modifying the composition of raw coal, in order that a different type of fuel might be produced, or possibly an alteration accomplished of the entire fuel contents into forms better suited to present-day requirements. The results obtained, as stated in the summary and conclusion were as follows:

Coals of the Illinois type can be coked at a temperature of approximately 750 to 850 deg. F.

The gaseous products consist chiefly of illuminants of high candlepower, and represent, together with the condensible material, the chief elements involved in the formation of smoke in the ordinary combustion of raw coal. The nitrogen of the coal is liberated as NH_3, at these temperatures, in amounts representing approximately 20 per cent. of the total nitrogen present.

The condensible distillate consists largely of oils with the minimum amount of tar and free carbon. The oils represent positive values for fuel, for carburetting water gas, or for other specific uses on account of their chemical characteristics as unsaturated compounds.

The coke residue has special characteristics which seem to make it of value as a concentrated fuel, capable of combustion without the formation of smoke, suitable for storing without the possibility of spontaneous combustion, and presumably adapted to the manufacture of gas for use in the suction-gas producer.

Certain facts seem to have been developed concerning the principles involved in the formation of coke which may open a way to the production of a kind of coke of such texture and strength as to make it acceptable for uses that are not now possible with coke made from similar coal, but formed under ordinary conditions such as are found in the ordinary gas-house retort practice, or that of the byproduct coke oven.

An appendix gives a resumé of previous studies and research work in the low-temperature distillation of coal.

SOCIETY NOTE

"Measuring Efficiency in Manufacturing" will be the subject of the meeting of the New York members of the American Society of Mechanical Engineers, Nov. 12, 1912, at the Engineering Societies Building, 29 West Thirty-ninth St., New York City. The discussion will be opened with a paper by Edward B. Passano, of Baltimore, Md., who measures efficiency in terms of profit. His position is that all profit should be treated as an item of expense. All loss, through inefficiency, is a potential or a positive profit to be made in enterprise. A reduction in the expense of a business increases the actual profit and the efficiency. This method keeps constantly before the management the actual value of each unit of production to the organization, and the loss through inefficiency which is a potential or possible profit. When a new method is being tried out, such a system will show the benefits or losses which are expressed in actual profit or loss as the work progresses. The discussion will be open to all interested whether members or not.

PERSONAL

Edward H. Lane, an occasional contributor to our columns, and chief engineer of the Commerce Power Co., of Kansas City, Mo., has undertaken the direction of the evening course in stationary engineering at the Y. M. C. A. night school in that city.

Charles H. Bigelow has accepted the position of chief mechanical engineer of the Millville Manufacturing Co., Millville, N. J. Mr. Bigelow recently returned from St. Catharines, Ont., where he had charge of the construction of a factory which will handle the Canadian business of the Yale & Towne Mfg. Co., Stamford, Conn.

At the regular monthly meeting of the board of directors of the Joseph Dixon Crucible Co., held Oct. 21, the following changes in officers and board of directors were made on account of the death of Vice-president William H. Corbin: George E. Long, former treasurer, was elected vice-president to succeed Mr. Corbin; J. H. Schermerhorn, former assistant secretary and assistant treasurer, was elected to membership in the board of directors and treasurer; Albert Norris, assistant secretary and assistant treasurer.

Moments with the Ad. Editor

"A man who never does any more than he is paid for;

Never gets paid for any more than he does."

If you haven't heard this before, it will be well for you to repeat it until you know it, and remember it.

There's a moral in these words.

Imagine a man, if you can, with a pair of balancing scales trying to make the services he is rendering his employer (on one side of the scales) balance the dollars he is receiving from his employer (on the other side).

Such a man will have to do this balancing "stunt" but once, for it's a "cinch," with him, that the dollar side of the scales will never get any heavier.

Money is the most valuable tangible thing in the world.

And money is never exchanged for *anything* except when that thing is considered by the owner of the money to be of more value to him than the *amount* of money he exchanges for it.

This law of exchange applies to other things, as well.

What you exchange, whether it's money or not, is *never* exchanged for anything which you consider of *less* value to *you* than what you offer the other fellow.

Now, your services as an engineer are for sale. The plant owner wants to buy your services. He gives you a job. He gets your services. You get his money in return.

Both of you get the better of the exchange, or it would not have been made.

You are benefited by the exchange because you get money in return—the most valuable tangible thing we know.

Your employer is likewise benefited by the exchange because he, through your services, gets what you get—money.

It's what we all are striving for. We and you will always want more money. It's human nature. And the only way for you to get more money is to make *more valuable* what you give in exchange for the money you get.

But trying to keep your side of the scale from being hoisted in the air by the dollars on the other side means that you will never get paid for any more than you do, which in any event, can be but little.

Render your employer a *maximum* service, always. Then you increase the *value* of your services. Then, the *value* of the exchange, and consequently a bigger return.

Of course, there are many ways in which you can and are increasing the value of your services as an engineer.

For a long time, we have been telling you how much real good you can get out of Power Advertising.

The advertisers in this paper have a service to render or they would not be in the paper.

They are offering you something that will mean a money saving investment in your plant.

Don't be skeptical. Don't be prejudiced. Don't allow yourself to feel that you are doing all you are being paid for.

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The selling section of Power offers one way. Get all you can out of it now.

POWER

Vol. 36 NEW YORK, NOVEMBER 12, 1912 No. 20

Co-operative Management(?)

HOW THE EXPENSE BURDEN OF CENTRAL STATION POWER IS DIVIDED.

(See page 700)

Cost of Power in Central Stations

By Henry D. Jackson

Figures from two specific cases to show the initial cost per kilowatt and the cost of producing a kilowatt-hour.

The injustice of the present system of discrimination makes it possible to sell below cost.

The central-station agents claim that they can supply power cheaper than it can be produced by the isolated plant, for the following reasons:

Because of the excellence of the apparatus installed in the central station.

Because of the high load factor in the central station.

Because the power load is all "velvet," since the interest, maintenance and depreciation on the plant are taken care of by the lighting load.

Because their standby losses are small compared to those of the isolated plant.

Because the fixed charges on the installation are far less than in the case of an isolated plant.

Because their stations are so enormous, as compared to those of the isolated plant, they can produce power very much cheaper, as the larger the plant, the more economical is its production of power.

Investigating to see if conditions warrant these assumptions is interesting. No one will question that the isolated plant as a rule does not contain as expensive apparatus as that installed in the central station, mainly because the service to be rendered is not so exacting, and the economy of coal is not of such great importance; hence gaining it by very high-priced apparatus is not attempted.

In an isolated plant the load factor is far better than that of the central station, even if 24-hr. service is taken into account. In two plants, which will be discussed later, the load factor is 17.2 per cent. An isolated plant with a load factor as low as this is hardly conceivable.

If conditions could be so arranged that the power load came at hours that did not lap over on the lighting load, the power load might be considered "velvet," but unfortunately this is not true, as shown in the accompanying curves, which are fairly representative of a combined summer and winter load. The power load will be noticed to overlap the lighting load between 2 p.m. and 7 p.m., and to occur during the peak of the lighting load. During this peak the central station must have capacity to supply not only the lighting load, but the maximum power load. The power load thus requires a large additional capacity in the station.

The claim on standby losses can be easily refuted. From the records of the gas and lighting commissioners on two stations, one of approximately 75,000 kw. capacity and the other approximately 6000, the following facts regarding coal are obtained: The coal per kilowatt-hour at the switchboard of the large plant is 2.6 lb. and per kilowatt-hour sold is 3.72 lb. In the smaller plant the coal is 3.2 and 4.4 lb. respectively. These figures show that a large part of the power generated is not bringing in a revenue and could readily be charged as a standby loss. Neglecting this, however, the units installed in these plants are highly efficient, producing in the larger plant a kilowatt-hour at the switchboard for approximately 18 lb. of steam or 1.8 lb. of coal per kilowatt-hour. Therefore, it is plain that the standby losses in coal represent

$$\frac{2.6 - 1.8}{1.8} = 44\tfrac{1}{2} \text{ per cent.}$$

In the smaller plant the standby losses in coal alone are approximately 50 per cent. The reason is that the central station has to keep boilers under steam at all times for the total capacity which may be called for at the peak load, and also to act if any of the boilers break down. The engines, or at least some of them, have to be kept warmed up ready to be put into service at a moment's notice. No such conditions exist in the isolated plant. No more boilers or engines are kept under steam than the service requires, and the only standby losses are those during the noon hour and when the plant is shut down at night, and under banked fires only a very small part of the coal used during the day is needed.

The claim that the installation cost is less than for an isolated plant and, therefore, that the fixed charges are lower is hard to believe. In the larger plant the cost per kilowatt of steam and electrical apparatus in the station is $114. Including plant and lines, but not transformers, lamps, meters and other articles, the cost is $248 per kilowatt; including real estate the total cost in the large plant is $323 per kilowatt. In the smaller plant the costs in the same order are $61, $285 and $308. Evidently the cost per kilowatt against which interest, maintenance and depreciation have to be charged, is far greater in the central station than in an isolated plant.

The claim that the larger the plant, the lower the cost of power, is true only so far as operating cost of the station itself is concerned, and the gain is not very large. When the larger plant had approximately one-half its present capacity, the station operating expense was 0.828c. per kw.-hr. generated; its present cost is 0.737c. Per kw.-hr. sold, the cost formerly was 1.157c., and is now 1.01c. The general expense which covers the distributing and office expense and taxes was 1.72c., and is now 1.716c. per kw.-hr. generated; and 2.39c. and 2.44c. per kw.-hr. sold, or, in other words, with a plant of double the capacity, the total operating expense per kilowatt-hour sold has only decreased from 3.55c. to 3.45c.

In the smaller plant the conditions are practically the same, except that the generating cost is 1.48c. and the general expense 2.05c., showing a total cost of 3.53c. as against 3.45c. in the larger plant. Thus a plant over ten times the size of the smaller one is not much more economical, and at the time it was over six times as large, its costs were greater. In this particular case the larger plant has the great advantage of obtaining its coal by water at a low rate.

The figures quoted are simply those for operating charges, including those of station, distribution, management and taxes. Allowing 6 per cent. for maintenance and depreciation on plant and line gives an additional cost of 1.27c. per kw.-hr. and 6 per cent. for interest charges on plant, lines and real estate adds 1.8c., making the total costs 6.52c. per kw.-hr. The average receipts in this plant were 5.95c. per kw.-hr., showing that the maintenance and depreciation charges are even less than 6 per cent. Of course, the interest charges represent the dividends paid to stockholders, but the other charges must be taken into account to keep the plant in proper condition to operate.

Interesting to notice is the relation between operating expenses at the station and the other charges against central-station power, as the isolated-plant expense in these items is so small. With the operating charges as a basis, the general expenses are:

	Per Cent.
Operation	100
Office	101
Distribution	73
Taxes	68.2
Maintenance and depreciation at 6 per cent.	125.7
Interest on investment	178.6

How, then, is it possible to give rates as low as 1 to 1.5c. per kw.-hr.?

If power is sold to some of the consumers at a price which will a little more than cover the operating charges in the station, the others must pay the interest, maintenance, depreciation, taxes and general expense of the plant, and that is what actually happens.

Referring to Fig. 1, the lighting load reaches the maximum point 16, which the power load brings up to 26.5. The power load, therefore, should be required

to pay the fixed charges on the increased capacity of 10.5. As a matter of fact with the prices as arranged, the lighting load pays the fixed charges upon the total capacity installed.

A further study of this curve shows that during 24 hr. a continuous capacity is required in the station of approximately 2.3 which is composed of a small part of the power load and the lighting load. This represents the very best condition of load that a power plant could ask for, and would require in the station only the capacity necessary for its generation which would be in continuous operation. Against this load the minimum charge ought to be made, for it requires the minimum of apparatus and the minimum cost for operation. The charges against it are made up partly by small users whose bills do not average over $5 a month, but who pay from 12c. to 15c. per kw.-hr., depending on the location of the plant. These people are small users, but continuous in their use of light, and as such should pay the minimum rather than the maximum rate. During this period a small power load is supplied by the same apparatus, yet these power users are given very close to the minimum figures.

That part of the load between lines *AA* and *BB* is also desirable, as it continues over a long period and is reasonably steady. The power load is large, and being practically continuous during the period of operation, its cost could fairly be assumed to be small except for the fact that it runs into the peak load at night, so that the fixed charges of the extra apparatus required should be charged against it. Since the lighting load here exists only in a small part, it is reasonable to suppose that it would pay only a small part of the cost, as this load is continuous, or practically so, and would require a minimum amount of apparatus. Here again, however, since the use of it is small and by a very large number of users, the maximum rate is applied.

The load which is undesirable and costly is represented by the peaks above the line *BB*. These peaks require the boilers being kept under steam and engines ready for operation to produce it; and as they last but a short time, the cost both for the apparatus and operation is excessive. The charges against this apparatus should be proportioned against those users who make it necessary—the power users during this period, and the light users. The latter are those who utilize light for only a short time during the day, such as the big department stores, small shops and offices, which cut out their light at or about 6 o'clock. This peak load gradually reduces as these loads come on. The peak, however, is somewhat held up by the gradual increase of light in the residential section, as the people from the offices and stores reach their homes. This lighting load at the residences is comparatively small compared to that of the peak, and lasts fairly well into the night, and after this comes a considerable lighting load from the all-night users, such as stores and residences. The peak then is mainly from the large users of the stores, the last hour or two of the power, and the comparatively small users of the offices and the small shops. As this peak is of short duration and requires excessive apparatus both in the station and line, those causing it should pay a high price; but do they?

The small offices and small stores whose maximum demand is less than a certain minimum, pay the full price of

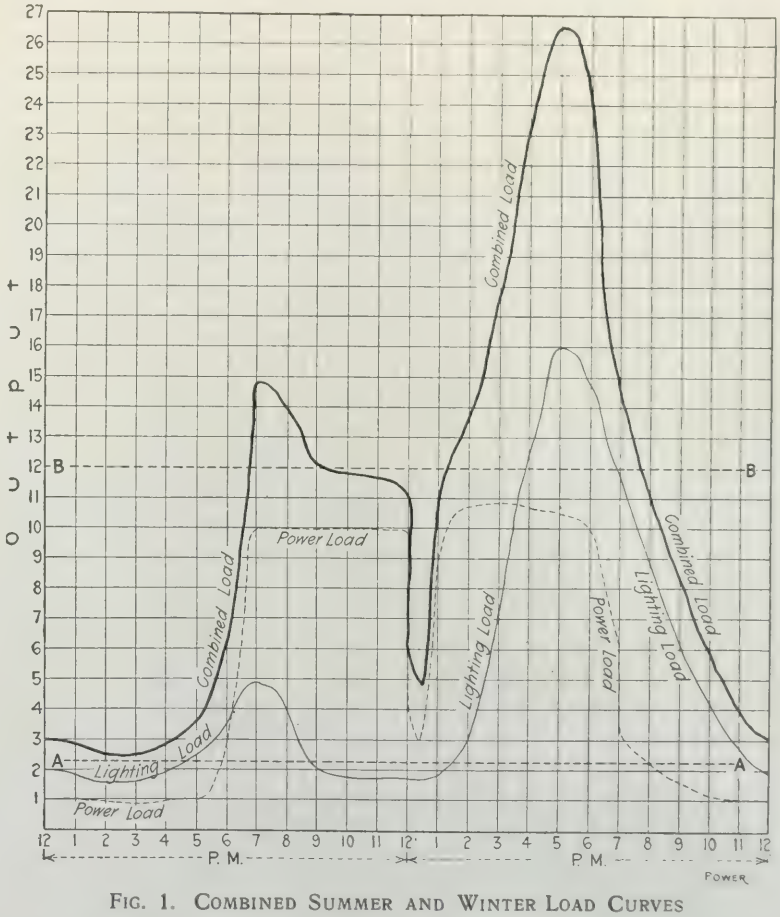


FIG. 1. COMBINED SUMMER AND WINTER LOAD CURVES

12 to 15c. per kw.-hr. The large users, such as those requiring power, and the large department stores, pay a very much reduced price in accordance with the total amount of power used, not to any great extent dependent upon the hours during which this power is used. Since the same apparatus is used for supplying this power to all users, that the small ones should pay a much larger price than the large ones seems unjust, and still more unjust to the residence users who use power for several hours, while the large department stores pay a low price in spite of the fact that they burn but three or four hours. Also, since the power load during this period is a large part of the peak, it seems unreasonable that those occasioning it should pay exceedingly low rates when the other users covering long periods should pay the maximum price.

Another point illustrating how the power load can be sold at low price is the fixed rate per month for the meter or what might be termed the readiness-to-serve charge, that is, a minimum charge of $1 a month whether power is used or not. Doubtless in a large city about 30,000 so use electricity, and during five months of the year pay $1 a month for meter rent, during which time practically no power is used, or they pay a total of $150,000 during this time, just for the privilege of having light if they happen to want it, and during the rest of the year they may pay anywhere from $5 to $18 a month at a price of 12c. per kw.-hr. Thus this number of users might readily pay the entire maintenance and depreciation and part of the general expense of the plant.

While riding on a car, the author heard a conversation between two gentlemen—one complaining regarding the cost of light at his house, the other evidently an officer in one of the large power-producing companies. The latter told the complainant that his company did not care for the small consumers, as they cost far more to take care of than they returned,

and that the profit of the company was made not from the residential districts or small consumers, but from the sales to the big department stores and users of large amounts of power. An examination of the conditions would refute this. Fig. 2 shows the conditions which would exist with various parts of the total power sold being supplied to those who pay 12c. per kilowatt-hour. The average rate received by this large company during the year for all its power sold was 5.95c. Assuming that 10 per cent. of the users pay 12c., the remainder would have to pay an average of 5.3c. If 40 per cent.

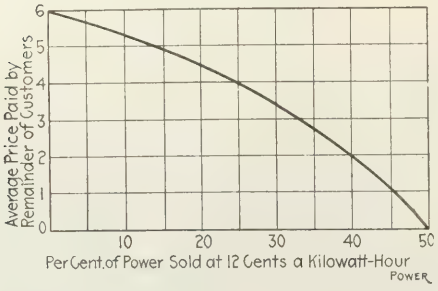


FIG. 2. DIVISION OF POWER RATES

of the power sold was supplied to customers paying 12c., the remaining 60 per cent. would have to pay an average under 2c., showing conclusively that the small consumers or those paying 12c. per kilowatt-hour were a very desirable load.

A public-service corporation is supposed to be operated for the benefit of the general public. The lighting companies, however, seem to operate for the benefit of a few, as the following figures, worked out from published rates, would indicate. A small user, "A," having a maximum use of 200 watts 24 hr. a day, 30 days in the month, would pay $17.28 for his power, although his load was continuous, and he would have to pay the full price of 12c. per kilowatt-hour.

Another user "B," having a kilowatt capacity of 5, which he operates for 10 hr. a day, 30 days in the month, would have a fixed charge remaining constant whether he uses power or not. He then pays an additional price for his power, making his total power cost, if he uses 1500 kw.-hr. per month, approximately $100, or per kilowatt-hour the price is 6.66c.

Still another consumer "C" has a kilowatt demand of 200, 30 days in the month, 4 hr. during the day. He, too, has a fixed charge, and pays varying rates for his power in accordance with the power used. On the basis of 24,000 kw.-hr., his total bill will be approximately $1165, or per kilowatt-hour, 4.85c.

Another consumer "D" has a demand of 800 kw., 30 days in the month, 4 hr. a day, or a total consumption of 96,000 kw.-hr. His bill would be approximately $36,000, or 3.75c. per kilowatt-hour. If this same customer uses an additional 100 kw. for 20 hr. per day, he reduces his price per kilowatt-hour to 2.71c.

Still another "E" has a maximum kilowatt demand of 1000 kw. for 2 hr., 25 days in the month, and 800 kw. 7 hr., 25 days in the month. He pays a fixed charge and varying rates for his power, bringing his cost per kilowatt-hour on the basis of 190,000 to 2.3c. If a user, call him "F," has 800 kw. for 22 hr. during 25 days, and 1000 kw. for 2 hr. during the 25 days, his cost for power on the basis of 490,000 kw.-hr. is 1.35c. Or he pays very slightly in excess of the station expense for his power, although he has to have 1000 kw. in the station and in the line capacity. There is reason for this latter customer getting a comparatively low price, because he has a 24-hr. load.

These charges against the consumers make an interesting study. The small user "A" has a 100 per cent. load factor, and, as such, should get a low rate. User "B" has a load factor of 41.6 per cent., and yet his price is far higher than user "D," who has a load factor of only 27 per cent., even when he has the additional 100 kw. continuously in use. Large user "E" has a load factor of 31.6 per cent., and yet he pays an even lower price; and "F," who has a load factor of 18.7 per cent., pays a still lower price.

Evidently then the rates are not adjusted either by a study of the load factor or the time of burning. They are adjusted:

First, to get the maximum price from the largest number of customers.

Second, to make the price so that they can sell power to the large users, meaning by this term those who use large amounts of power regardless of the time during which they use it.

Quite evidently it will cost an isolated plant far more to operate if the time of service is short, and, therefore, the rates of the central station are adapted to meet this condition. Evidently also, the longer the time the plant operates and the nearer its load to the capacity of the plant, the lower will be its cost for power, and thus the central station attempts to meet these conditions.

The figures here given are worked out from published rates. There is good reason to believe that these rates are elastic, and that far lower figures will be quoted if the isolated plant can produce it lower than these figures. To get the business the central station meets the figures and goes them one better. In every case, however, if the plant owner but knew it, a string is attached, this string consisting often of the fixed charge, and often a very large charge for any additional power which may be used over and above that contracted for, and these strings frequently work greatly to the disadvantage of the isolated plant.

An Old Locomotive and an Interesting Letter

BY PROF. F. E. SANBORN

In the South Kensington Museum, London, is the original locomotive known as "Puffing Billy." It was invented and constructed in 1813 by William Hedley at the Wylam Colliery. It was still at work there in 1862 and was received by the museum in 1865, after about 50 years of service.

The engine has two vertical, steam-jacketed cylinders, 9 in. in diameter by 36 in. stroke. The piston rod passes out the upper end of each cylinder and connects to a grasshopper beam. A long rod pivoted at about the middle of the beam transmits the motion downward to an overhung crank. The cranks, one on each side of the engine, are 90 deg. apart. The crankshaft transmits motion through spur gears to the two driving shafts, one forward of and the other behind the crankshaft. Each driving shaft has two driving wheels, 39 in. in diameter.

The valve is a short D-slide valve, worked by tappets.

The boiler is a wrought-iron cylinder with one end egg-shaped, and with an internal return flue, Trevithick type. The grate area is 6 sq.ft., the heating surface 77 sq.ft. The fireman and the engineer stood at opposite ends of the boiler.

When first run the noise and the smoke were considerable, so that legal opinion was consulted as to whether the engine was a "nuisance." Later the trouble was partly overcome by passing the exhaust steam into a quieting chamber before it was discharged into the chimney. The subject of the smoke nuisance is thus not new, and the recognition of a noise nuisance is not altogether modern.

The legal opinion sought was expressed in the following letter:

"Mr. Blackett,
Proprietor Coal Mine,
Wylam, Northumberland.

"It does not appear to me that there is any objection arising from the lease itself to Mr. Blackett's conveying his coal waggons by means of this steam engine. But I think that the use of such an engine may be deemed a nuisance to A if the smoke and noise occasioned thereby render his habitation unhealthy or uncomfortable. But this must entirely depend upon the quantity of smoke and noise so occasioned and the distance of the house of A from the waggon way."

Then apparently an afterthought occasioned a postscript.

"If the noise of this engine disturbs the cattle grazing in the lands adjacent to the waggon way so as to injure them with regard to their feeding, I think it may be considered as a nuisance."

(signed) "ROBT. HIGGINS WILLIAMSON,
"N. Castle, 12th Aug. (?) 1814."

Gantry Coal Handling Crane

By A. D. Williams

A crane having a guaranteed capacity of 100 tons per hour, hoisting coal from cars to a discharge chute 75 ft. above. It is operated by motors and controlled by one operator.

Peculiar conditions complicate the problem of handling coal at the Madison Ave. power house of the Toledo Ry. & Light Co., on the bank of the Maumee River. Any system of track hoppers and underground conveyors would have been extremely expensive, owing to the large amount of waterproofing required. As the water level is only 4 or 5 ft. below the surface of the ground it would have been troublesome to install and maintain an under-track pit of adequate capacity.

The special crane designed to meet the conditions, shown herewith, is a gantry truss, spanning three tracks with the upper chord on which the grab-bucket trolley travels cantilevered over a fourth track. The machine travels parallel with the boiler room on a runway 400 ft. long, and can take coal from cars on any of the four tracks, crushing and elevating it and delivering into a storage bin above the boilers.

The clam-shell grab bucket holds 2.5 cu.yd., or aproximately 3600 lb. of coal. Coal is raised about 45 ft. and is carried to the end of the bridge nearest to the building and dumped into a hopper having a screen bottom. The lumps pass over this and through the single-roll crusher to a screw conveyor delivering to the elevator boot, while the fine coal drops through the screen into a chute delivering to the boot. The elevator is of the continuous bucket type and lifts the coal about 75 ft. and discharges it into a spout delivering at an angle through hatches in the roof of the boiler room to the storage bin from which it feeds by gravity to the boilers.

The entire machine is controlled by an operator in a cab on the outboard leg of the gantry at a point where he can observe the operation of the bucket and machinery. This machine was guaranteed to have a capacity of 100 tons per hour and a test run showed a handling capacity of nearly 150 tons per hour. In handling coal at this rate the grab bucket must make a complete round trip between the car and the receiving hopper every 30 sec. on the average, and many of the trips are made at a higher speed. Bumping logs which prevent the bucket from swinging in below the hopper when hoisting and racking at the same time, contribute to rapid operation and may be seen immediately below the lip of the hopper over which the bucket passes.

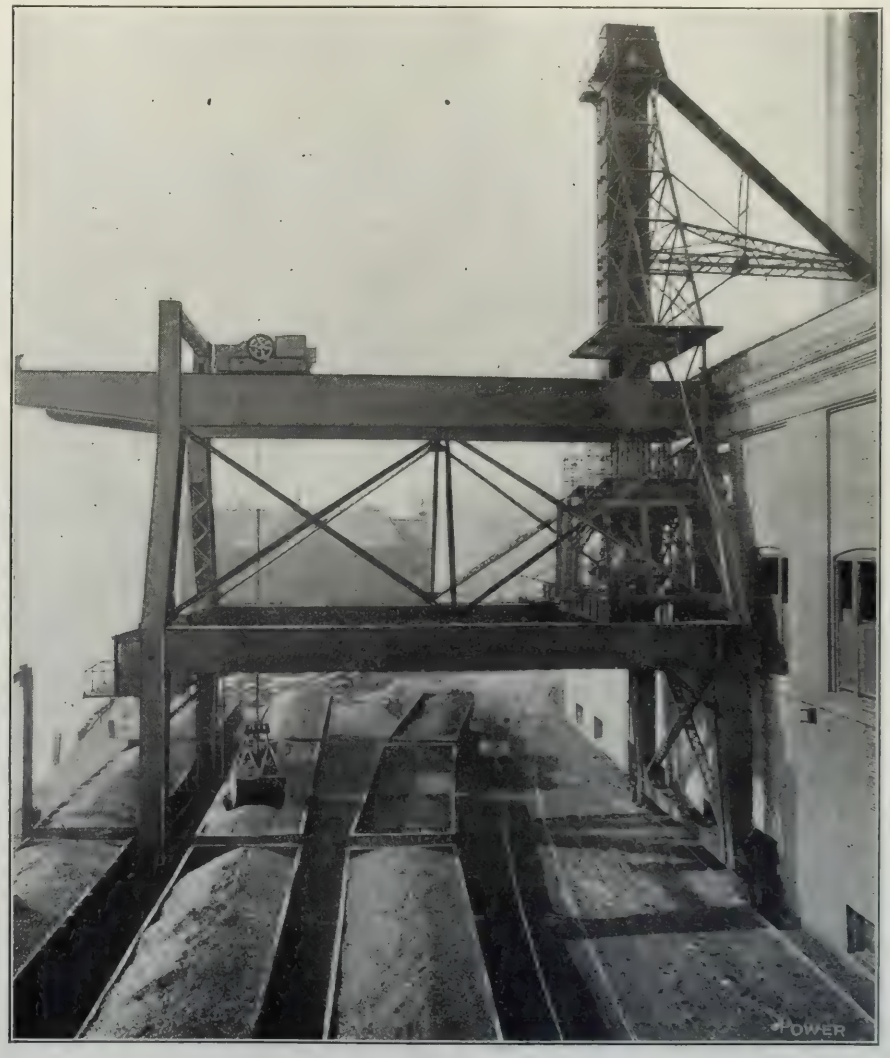
The crane is operated by 550-volt, direct-current motors. The main and auxiliary hoist motors are series wound and have dynamic brake control which facilitates rapid stops and reversal. The racking motor and the two bridge-travel motors are series wound with rheostatic control, and the crusher and elevator motors are compound wound with ordinary starting boxes. These latter run continuously when the machine is working. A summary of the power consumption of this machine is shown in the accompanying table.

AVERAGES FROM SEVEN READINGS

	Volts	Amp. to Start	Amperes Running, Up	Down
45 hp. Main hoist motor	548	114	56	31
30 hp. aux. hoist motor (bucket closed)	541	83	44	26
10 hp. Racking motor....	530		20 light	
Two 30 hp. Bridge motors (operated in parallel)....	543	79	36	
	530	75	35	
20 hp. Elevator motor...	551		13	

The difference in the readings taken are due to a single set of instruments being used upon each of the motors in turn. This might readily account for the difference shown in the case of the two bridge motors. One being tested running down the runway and the other while returning, the wind would help one motor and retard the other. As the crane has considerable exposed surface, 3 or 4 lb. wind pressure per square foot would make a very appreciable difference. The large voltage fluctuations, however, are due to the power being taken from a traction plant.

The total weight of the crane is 125



GANTRY COAL HANDLING CRANE

tons and the speed of bridge travel approximately 125 ft. per min. The crab or trolley weighs 29,500 lb., the grab bucket 5800 lb. and will carry 3600 lb. of bituminous coal. In the test mentioned above the trolley round trip was about 110 ft., and that of the hoist 90 ft. per round trip.

The hoist motor is geared so that the hoisting speed is 140 ft. per min. at 600 r.p.m. of the motor with 56 amp. at 550 volts current input. Very close to two seconds is taken to dump the bucket and about 7 sec. to fill it. During the balance of the time the trolley and hoist are both working, half of the time with a load of coal, the return trip with an empty bucket. Both the main and auxiliary hoist motors pull at the same time through different falls.

The foregoing information and data were supplied by the Toledo Bridge & Crane Co., which designed and built the crane.

Light and Power Plant Accounts

By E. R. Shields

Pointers for the accounting end of power plant management.

It is also shown how to enter various items in their proper place.

In accounts dealing with electric-light and power plants, special attention should be given to distinguishing between capital and revenue expenditures or expense. An electrical plant more than any other is subject to changes. Improvements and extensions are constantly under way, and it requires the unremitting effort of the auditor to classify the various disbursements.

Depreciation calls for a great deal of consideration. The machinery in most electrical plants operates 24 hr. daily, causing a high up-keep cost. The advent of new and improved electrical machinery is as much a cause for rapid depreciation, as wear and tear.

Depreciation should be based on the way repairs are maintained. This applies to the motors, dynamos, engines, boilers, etc. Wires and lamps are depreciated from 10 to 40 per cent. annually by some. Poles and crossarms depreciate about 10 per cent. annually. In each plant should be taken into consideration, age, maintenance and whether the machines are run almost constantly or not.

Meter No.	Month	Jan.	Feb.	Mar.	Apr.	May	June	July	Aug.	Sept.	Oct.	Nov.	Dec.
	Date Read	1/1	3/3	4/2									
100	Jno. Jones 25 W. 8th St.	4567	4610	4720									
	Total												

Power

FIG. 1. METER READING CARD

Some companies do not make a practice of installing private wiring and fixtures, which is done by an outside electrician. Others have a department which not only installs the wiring and fixtures but conducts a store and show room, retailing such appurtenances as electric irons, toasters, etc. In the latter event, the installation department and the store should be separated so that the profits therefrom can be ascertained with least effort.

Usually a deposit of $5 or more is required from a customer before a meter is installed. This is carried as a liability and called "customers' deposit account." The account is credited with the money paid in, and debited when refunded. If a customer fails to pay for current used, the deposit account is debited with the total deposit, and accounts receivable credited with the amount due, and a check drawn for the balance, thus crediting cash. The entry would be

Customers' deposit account.....	$5.00
To sundry	5.00
Accounts received	3.50
Cash	1.50

While an employee is engaged upon work classed as improvements, such as extending a transmission line, his salary should be capitalized. Small companies employ two or three linemen who perhaps spend half of their time making these extensions, and the other half making repairs. It should be the duty of the payroll clerk or the time-keeper to specify what proportion is "repairs."

Care should be taken to see that freight charges have been capitalized on machinery. The freight cartage and extra help required in installing boilers and engines frequently run up to large sums which should not be overlooked. Separate stock accounts should be maintained for meters, lamps, poles, crossarms and wiring. A storekeeper should be put in charge with instructions to deliver only upon a requisition signed by the official in charge and countersigned by the auditor or the

		January						February						March						April					
Meter No.	Name and Address	Due from Last Month	Current used	Amount	Mdse	Due this Month	Paid	Due from Last Month	Current used	Amount	Mdse	Due this Month	Paid	Due from Last Month	Current used	Amount	Mdse	Due this Month	Paid	Due from Last Month	Current used	Amount	Due this Month	Paid	
100	Jno Jones 25 W 8th St	3.20	241	482	2.50	1293	700	593																	
	Total																								
Columns		1	2	3	4	5	6	7																	

Power

FIG. 2. PAGE OF ACCOUNTS RECEIVABLE LEDGER

Meter No. 100 Date _____ 191__

BROWNSVILLE POWER CO.
BROWNSVILLE, O.

To Jno. Jones
25-W. 8th St

Reading This Month 4567
Reading Last Month 4326
Current Used 241 @02 $4.82

NOTICE:-
This bill must positively be presented when payment is made.

Meter No. 100
Date _____ 191

Jno Jones
25-W. 8th St.

Reading This Mo. 4567
Reading Last Mo. 4326
Used 241
Amount 482

To be Retained by Company.

Power

FIG. 3. MONTHLY STATEMENT

head bookkeeper. Articles when taken from stock and put into actual service should be transferred in the ledger as follows:

Meters in service..... $250.00
To meters in stock..... $250.00

Per requisitions Nos. 4, 7, 10, 15, 18, 25.

Stock accounts should also be kept for coal, oil and waste, supplies, etc., the account to be credited and expense to be debited by journalizing the requisitions. Coal will have to be charged to expense as notices are received from the engineer that it was used.

The report form Fig. 1 is used in meter reading. The difference between the reading of the preceding and the current month is then posted into the accounts receivable ledger, which is ruled as shown in Fig. 2.

Column 1 represents the balance carried forward from the preceding month, and the total of that column represents the total accounts receivable, and should agree with the controlling account in the general ledger. Thus the total of the first column in January represents the December accounts receivable.

In column 2 is the meter reading which has been carried from the reading book.

The total of column 3 is debited to accounts receivable in the general ledger, and credited to the current-sold account. This serves the purpose of a sales book as it represents the total current sold for the month.

The entry in column 4 represents the merchandise sold which the purchaser wanted charged in the regular account rendered monthly. The total should be credited to the merchandise account and debited to accounts receivable.

Column 6 should agree with the accounts-receivable column on the debit side of the cash book. The advantages of this book are obvious, and where from 10,000 to 20,000 accounts must be handled monthly it will be found a necessity.

Req. No. _____ Date _____ 191

Storekeeper:-

Deliver To _____

Quantity		Cost Price

For Office Use

Debit		Credit	
Acct.		Acct.	

Power

FIG. 4. REQUISITION BLANK

In making the monthly statements, a form similar to Fig. 3 is used. When this is presented at the office for payment the cashier removes the stub which he retains, the larger portion he stamps paid and returns to the customer as a receipt. The auditor by checking these stubs with the payments recorded in the accounts-receivable ledger can easily detect any irregularity or fraud.

The requisition to be used for service on the storekeeper can be ruled as in Fig. 4, the accounting department to retain a carbon copy. This is also used as a voucher for journal entries.

Amendments to National Inspecting and Testing Rules

At a general session of the Interstate Commerce Commission, held in Washington, Sept. 12, in the matter of the preparation, approval, and establishment of rules and instructions for the inspection and testing of locomotive boilers and their appurtenances, it was ordered that Rules 29 and 35, as approved by the commission, June 2, 1911, be amended to read as follows:

29. Siphon. Every gage shall have a siphon of ample capacity to prevent steam entering the gage. The pipe connection shall enter the boiler direct and shall be maintained steam tight between the boiler and the gage. The siphon pipe and its connections to the boiler must be cleaned each time the gage is tested.

35. Setting of Safety Valves. Safety valves shall be set to pop at pressures not exceeding 6 lb. above the working steam pressure. When setting safety valves two steam gages shall be used, one of which must be so located that it will be in full view of the person engaged in setting such valves; and if the pressure indicated by the gages varies more than 3 lb. they shall be removed from the boiler, tested, and corrected before the safety valves are set. Gages shall in all cases be tested immediately before the safety valves are set or any change made in the setting. When setting valves the water level in the boiler shall not be above the highest gage cock.

These amendments will be made effective on and after Jan. 1, 1913.

The Edison Illuminating Co., of Detroit, is erecting a large lighting plant on the river front in Sandwich, Ont., to supply current for both heating and lighting in Essex, Leamington and Amherstburg. These towns are at present supplied by small local plants.

Aristotle has said of the somnambulistic state: "There are individuals who walk about in their sleep seeing as clearly as those that are awake." Can't do this in a power plant. A sleep-walker in an engine room wakes up to find his job a dream.

Steam Consumption Computations

By J. A. Knesche

By the use of formulas employing empirical constants the losses due to condensation and to leakage may be approximated for any given type of engine, steam pressure, etc. This, added to the computed useful steam per horsepower-hour gives the total steam consumption.

In many manufacturing plants the cost of steam must be determined to find the complete production cost. Usually the power required for the different operations can be computed readily from test data. The steam required by different engines also can be readily determined if a surface condenser and testing apparatus are available when making a test. However, where several large engines operate under different conditions, but draw their steam from one boiler plant, determining the steam required per horsepower for each engine is exceedingly difficult.

The usual procedure is to determine the steam theoretically required from the indicator diagram and then allow for condensation and leakage. As these losses may range from 20 to 100 per cent. of the steam theoretically required estimating their proper values is often difficult.

The superiority of the following method over the one usually given in textbooks is shown in the computed steam consumption of several examples, the results being very close to what the engineer knows should be the approximate steam consumption for a given set of conditions. However, scientific accuracy cannot be claimed for this method, even though all the conditions bearing on a given set of indicator diagrams and the engine room. To be able to determine known, but for practical purposes scientific accuracy is not necessary in the engine room. To be able to determine the steam consumption to within 5 or 10 per cent. is all that can be claimed for the method herein outlined.

In Tables 1 to 4 are given the useful steam, exclusive of losses, required per indicated horsepower-hour for the engine, using dry saturated steam. Occasionally compound engines must be operated noncondensing, but as no reliable data were available for determining the steam consumption under these conditions, no table was compiled covering this case. However, the gain in steam economy by equipping a compound noncondensing engine with a first-class condenser usually will be approximately 25 per cent. Therefore, if the figure representing the useful steam for compound noncondensing engines be taken from Table 4, and if the steam losses are determined by the methods given, the sum of these divided by 0.75 will give a value very close to the true steam consumption of the compound noncondensing engine under consideration.

The greater part of the steam loss within the cylinder is due to condensation and the smaller part to leakage past the piston and valves. The condensation losses S_c are determined from the formula

$$S_c = \frac{K}{1 \quad P} \qquad (1)$$

where

S_c = Steam losses through condensation;

P = Piston speed in feet per second;

K = Coefficient as given in Table 5.

when the ratio of stroke to diameter, $\left(\frac{s}{d}\right)$ is approximately 2.

The smaller figures are to be applied to engines that are new or in very good condition.

When $\frac{s}{d}$ differs considerably from 2, the values in Table 5 are to be multiplied by the coefficients which are given in Table 6.

If the admission steam is superheated sufficiently, cylinder condensation may be entirely avoided. With cutoff in the high-pressure cylinder ranging from 40 to 25 per cent. a superheat of from 175 to 250 deg. F. is sufficient to prevent condensa-

TABLE I. USEFUL STEAM PER I. H.P. HR., S_u IN POUNDS WITH SINGLE CYLINDER NON CONDENSING, ENGINE STEAM THROTTLED*

Avg. abs. admission pressure	Cutoff in per cent of full stroke.								
	70	60	50	40	33.3	30	25	20	15
30									
35	51.50	51.00							
40	46.00	44.50	44.50						
45	42.50	40.50	39.50	39.00					
50	39.50	37.75	35.75	35.00	35.00				
55	37.85	35.55	33.50	32.25	32.00				
60	36.50	34.00	31.80	30.40	29.95	29.80			
65	35.25	33.00	30.75	28.80	28.20	27.75			
70	34.25	32.00	29.80	27.80	26.85	26.30	25.95		
75	33.50	31.00	29.00	27.00	25.85	25.20	24.75		
80	32.65	30.40	28.20	26.20	25.00	24.30	23.75		
85	32.00	29.80	27.55	25.50	24.45	23.52	23.00	22.10	
90	31.50	29.35	27.15	25.00	23.85	23.05	22.50	21.45	
95	31.00	28.95	26.60	24.60	23.35	22.75	22.00	20.90	
100	30.60	28.50	26.30	24.25	22.90	22.25	21.50	20.50	19.60
105	30.30	28.05	26.00	24.00	22.50	21.95	21.00	20.10	19.45
110	30.00	27.80	25.75	23.65	22.25	21.55	20.75	19.90	19.10
115	29.75	27.50	25.50	23.40	22.00	21.30	20.50	19.50	18.96
120	29.50	27.30	25.25	23.05	21.65	21.10	20.25	19.25	18.52
125	29.15	27.05	24.95	22.80	21.50	20.96	20.00	19.00	18.40
130	28.85	26.80	24.75	22.60	21.35	20.75	19.75	18.85	18.25
135	28.60	26.55	24.50	22.45	21.15	20.50	19.50	18.60	18.00
140	28.45	26.30	24.30	22.30	21.00	20.30	19.30	18.50	17.82
145	28.22	26.15	24.15	22.25	20.80	20.15	19.15	18.25	17.75
150	28.05	26.00	24.05	22.15	20.60	20.00	19.00	17.80	17.55

* With new engines S_u may be taken approximately 1.75 lb. less.

TABLE 2. USEFUL STEAM PER I. HP. HR. S_u IN POUNDS WITH SINGLE CYLINDER, NON-CONDENSING ENGINES, AUTOMATIC CUTOFF.*

Avg. abs admission pressure.	Cutoff in per cent of full stroke.										
	70	60	50	40	33.3	30	25	20	15	12.5	10
35	50.50										
40	44.50	40.50	39.50	38.30	39.50						
45	40.50	36.80	35.50	34.00	33.50	34.00	35.50				
50	38.00	34.80	32.50	30.80	30.25	30.50	31.60				
55	36.00	33.00	30.75	29.00	27.90	27.80	28.55	30.50			
60	34.50	31.75	29.50	27.50	26.40	25.90	26.15	26.50	28.00		
65	33.45	30.75	28.40	26.25	25.00	24.50	24.45	24.40	25.50	27.50	
70	32.50	29.90	27.50	25.20	24.00	23.45	23.10	22.75	23.55	25.00	27.25
75	31.55	29.20	26.75	24.45	23.15	22.60	22.25	21.60	22.10	23.20	25.00
80	31.00	28.50	26.00	23.75	22.50	22.00	21.50	20.80	21.00	21.95	23.00
85	30.50	28.00	25.45	23.20	22.00	21.45	21.00	20.00	20.00	20.80	21.50
90	30.00	27.50	25.00	22.75	21.50	20.95	20.50	19.50	19.50	19.90	20.50
95	29.50	27.00	24.60	22.40	21.10	20.55	20.00	19.00	19.00	19.10	19.50
100	29.10	26.50	24.20	22.00	20.85	20.10	19.50	18.60	18.45	18.50	19.00
105	28.80	26.10	23.90	21.65	20.45	19.80	19.00	18.40	17.85	18.00	18.50
110	28.50	25.75	23.65	21.45	20.10	19.50	18.80	18.15	17.45	17.50	18.00
115	28.25	25.55	23.40	21.20	19.85	19.20	18.55	17.80	17.00	17.10	17.55
120	28.00	25.40	23.15	21.00	19.55	19.00	18.35	17.50	16.60	16.80	17.10
125	27.75	25.25	22.95	20.75	19.40	18.80	18.10	17.20	16.40	16.50	16.65
130	27.50	25.10	22.75	20.50	19.25	18.60	17.90	17.00	16.25	16.25	16.35
135	27.35	24.90	22.55	20.30	19.05	18.45	17.60	16.80	16.10	16.00	16.00
140	27.10	24.65	22.40	20.15	18.85	18.30	17.35	16.55	15.95	15.75	15.55
145	26.90	24.45	22.30	20.00	18.70	18.15	17.10	16.45	15.70	15.55	15.25
150	26.75	24.25	22.15	19.88	18.55	18.00	16.95	16.35	15.50	15.40	15.00

* With new engines the above values may be taken approximately 1.5 lb. less.

tion. But even in such a case, S_t must not be taken as zero, because superheated steam, compared with saturated steam, does less work in the engine cylinder on account of the more rapid fall of its expansion curve and also, because heat is required to superheat the steam. If S_t is determined from the formula

$$S_t = \frac{33{,}000}{1 + 1.4P} \times \frac{1.4}{i.hp.} + \frac{0.62}{P} \quad (2)$$

where

$i.hp.$ = Indicated horsepower;

P = Piston speed in feet per second.

For compound engines the leakage loss

TABLE 4. USEFUL STEAM PER I.H.P. HR., S_u, IN POUNDS WITH SINGLE-FLUID CONDENSING ENGINES AUTOMATIC CUTOFF

Avg. abs. admission pressure	Cutoff in per cent of full stroke										
	50	45	40	35	30	25	20	15	12.5	10	7	5
40	24.10											
45	23.30	22.50	22.20	21.20	19.40	18.60	17.80	16.95	16.95	16.60		
50	23.25	21.10	19.85	19.40	18.85	17.45	16.70	16.50	16.30	16.15		
55	23.00	20.80	19.50	18.75	18.05	17.10	16.45	16.10	16.00	15.75	16.10	
60	22.74	20.40	19.20	18.50	17.80	16.90	16.15	15.75	15.75	15.45	15.65	
65	22.50	20.05	19.10	18.25	17.55	16.70	15.90	15.50	15.50	15.15	15.45	
70	22.10	20.10	18.90	18.05	17.40	16.50	15.70	15.30	15.35	14.95	15.20	
75	21.95	19.97	18.70	17.80	17.25	16.35	15.50	15.10	15.20	14.74	15.05	
80	21.80	19.81	18.50	17.72	16.95	16.25	15.30	14.90	14.90	14.50	14.85	
85	21.70	19.70	18.30	17.60	16.85	16.07	15.05	14.80	14.75	14.40	14.70	
90	21.60	19.60	18.10	17.50	16.75	15.95	14.95	14.70	14.60	14.30	14.25	
95	21.50	19.50	18.00	17.40	16.65	15.75	14.85	14.67	14.50	14.20	14.05	
100	21.40	19.40	17.90	17.30	16.55	15.65	14.75	14.60	14.40	14.10	13.90	
105	21.30	19.30	17.80	17.27	16.45	15.55	14.67	14.53	14.30	14.000	13.80	
110	21.20	19.20	17.76	17.15	16.40	15.47	14.60	14.47	14.20	13.95	13.75	
115	21.10	19.10	17.67	17.07	16.35	15.37	14.55	14.40	14.10	13.90	13.70	
120	21.00	19.00	17.60	17.00	16.30	15.30	14.50	14.35	14.00	13.85	13.65	
125	20.90	18.95	17.50	16.95	16.25	15.23	14.45	14.30	13.95	13.80	13.60	
130	20.80	18.90	17.50	16.90	16.20	15.15	14.40	14.25	13.90	13.75	13.55	
135	20.74	18.85	17.45	16.85	16.15	15.08	14.37	14.20	13.85	13.70	13.50	
140	20.70	18.80	17.40	16.80	16.10	15.00	14.35	14.15	13.77	13.65	13.45	
145	20.65	18.75	17.35	16.75	16.05	14.93	14.30	14.10	13.75	13.60	13.40	
150	20.60	18.70	17.30	16.70	16.00	14.92	14.27	14.07	13.70	13.55	13.35	

TABLE 4. USEFUL STEAM PER I.H.P. HR., S_u, IN POUNDS WITH COMPOUND CONDENSING ENGINES *

Avg. abs. admission pressure	Cutoff in per cent of full stroke (reduced to low-pressure cyl.)							
	25	20	15	12.5	10	7	5	4
40								
45	17.30	16.45	15.25	14.80	14.50	14.75	15.00	
50	17.25	16.10	15.00	14.50	14.10	14.25	14.50	
55	16.95	15.85	14.80	14.15	13.95	13.75	14.00	
60	16.70	15.55	14.60	13.85	13.25	13.40	13.50	13.80
65	16.50	15.35	14.40	13.65	13.10	13.05	13.00	13.00
70	16.35	15.15	14.20	14.30	12.95	12.70	12.60	12.75
75	16.25	15.00	14.00	13.35	12.85	12.50	12.25	12.50
80	16.15	14.90	13.85	13.25	12.75	12.30	12.00	12.25
85	16.07	14.80	13.70	13.15	12.65	12.10	11.80	12.00
90	15.95	14.70	13.55	13.05	12.55	11.95	11.60	11.75
95	15.90	14.60	13.45	12.95	12.47	11.85	11.45	11.50
100	15.85	15.60	13.35	12.85	12.35	11.75	11.30	11.35
105	15.82	14.50	13.25	12.80	12.25	11.65	11.20	11.20
110	15.74	14.50	13.15	12.70	12.15	11.54	11.10	11.10
115	15.76	14.45	13.05	12.60	12.06	11.45	11.03	11.00
120	15.73	14.40	12.95	12.50	11.95	11.35	10.94	10.90
125	15.70	14.35	13.00	12.45	11.85	11.25	10.83	10.80
130	15.67	14.30	12.87	12.42	11.76	11.15	10.75	10.70
135	15.64	14.25	12.96	12.39	11.68	11.08	10.67	10.60
140	15.61	14.20	12.99	12.36	11.55	10.98	10.56	10.50
145	15.58	14.17	12.91	12.33	11.46	10.86	10.53	10.40
150	15.55	14.14	12.90	12.32	11.40	10.80	10.47	10.30

* The above values are for engines in good condition and well jacketed, cutoff and without preheating of the receiver.

With new engines the above values may be taken from 1 to 1.5 lb. less to the smaller cutoffs.

TABLE 5.

Engine Type	Coefficient
Throttling, single-cylinder, non-condensing	.27 to .31
Automatic cutoff, single cylinder, non-condensing	.20 " .26
Automatic cutoff, single cylinder, condensing	.21 " .26
Compound, condensing	.19 " .27
Triple expansion, condensing	.16 " .19

$S_t = \frac{K}{1 + P}$ when superheated steam is used, then K will be from ¼ to ½ the value for saturated steam, as given in Table 5.

For single-cylinder engines the leakage past the piston, S_l, may be determined according to the formula

$$S_l = \frac{33{,}000}{1 + 1.4P} \times \frac{1.4}{i.hp.} + \frac{0.62}{P} \quad (3)$$

where

$i.hp.$ = Indicated horsepower;

P = Piston speed in feet per second.

For compound engines the leakage loss is 80 per cent. and for triple-expansion engines 64 per cent. of the value given by this formula. With engines in very good condition S_l may be only one-half the foregoing values while, with pistons in visibly leaky condition, the leakage loss may be twice this or even more.

The condensation losses in the steam lines plus any water carried over with the steam when the boilers prime may be taken from 4 to 10 per cent., depending upon the size and length of the steam line, its covering and the frequency with which the boilers prime.

The following examples will make clear the application of the preceding discussion, tables and formulas:

Example 1.—A 24×48-in. vertical, single-cylinder, piston-valve, throttling engine, the diagrams from which are shown in Fig. 1.

Top End.

Useful steam per indicated horsepower-hour from Table 1, S_u = 34.25 lb.

Average admission pressure = 67.8 lb. absolute.

Cutoff = 73.45 per cent.

Ratio of stroke to diameter $\left(\frac{s}{d}\right)$ = 1.43.

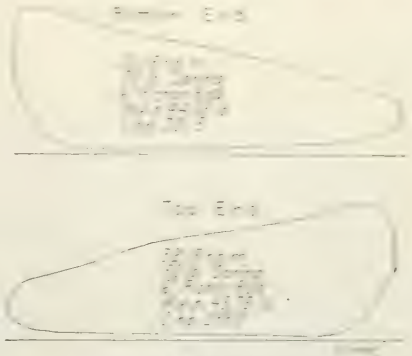


Fig. 1

Piston speed (P) = 5.36 ft. per sec.

Then

$$1 + \sqrt{P} = 1 + \sqrt{5.36} = 3.315$$

From Tables 5 and 6, K = 27.93 × 0.91 = 25.4 and from equation (1)

$$S_t = \frac{25.4}{3.315} = 11.05$$

TABLE 6.

If $\frac{s}{d}$ is approximately	Then K in Table 5 is to be multiplied by
1	0.92
1.25	0.95
1.50	0.97
1.75	1.00
2.00	1.02
2.50	1.05
3.00	1.07
4.00	1.09
5.00	1.11

The leakage losses S_l from equation (2) are

$$\frac{33{,}000}{1 + 284 \times 5.36} \times \frac{1.4}{34.25} + \frac{0.62}{5.36} = 1.57 \text{ lb.}$$

the horsepower being computed from the indicator diagram, Fig. 1. Therefore,

S = 34.25 + 11 + 1.57 = 46.82 lb.

The steam-line losses are taken at 4 per cent.; hence the total steam consumption is 46.82 × 1.04 = 48.7 lb. per hp.hr.

Bottom End.

Useful steam per indicated horsepower-hour from Table 1, S_u = 33.5 lb.

Average admission pressure = 57.8 lb. absolute.

Cutoff = 70.5 per cent.

Condensation losses S_c same as for top end or 11 lb.

Leakage losses S_l =

$$\frac{33{,}000}{1 + 284 \times 5.36} \times \frac{1.4}{34.25} + \frac{0.62}{5.36} = 1.55 \text{ lb.}$$

Therefore

$S = 33.5 + 11 + 1.55 = 46.05$ *lb.*

The steam-line losses are taken at 4 per cent.; hence the total steam consumption is

$46.05 \times 1.04 = 47.9$ *lb. per hp.-hr.*

Therefore the average steam consumption for the engine is 48.3 lb. per indicated horsepower-hour.

Example 2—A 12 and 20 by 18-in. compound flywheel boiler-feed pump, noncondensing with Meyer cutoff gear, the diagrams from which are shown in Fig. 2.

High-pressure Head End and Low-pressure Crank End*

Useful steam per indicated horsepower-hour from Table 4, $Su = 13.6$ lb.

Average admission pressure 124.7 lb. absolute.

Cutoff = 47.5 per cent.

Effective area of high-pressure piston = 109.95 sq.in.

Effective area of low-pressure piston = 309.25 sq.in.

Cutoff in high-pressure cylinder reduced to low-pressure cylinder =

$$\frac{47.5 \times 109.95}{309.25} \text{ or } 16.89 \text{ per cent.}$$

Condensation losses Sc where $\frac{s}{d} = 1.5$:

From Tables 5 and 6, $K = 19.95 \times 0.91 = 18.15$

$P = 1.6$ and $\sqrt{P} = 1.26$

Therefore,

$$Sc = \frac{K}{\sqrt{P}} = \frac{18.15}{1.26} = 14.4 \text{ lb.}$$

The leakage losses Sl =

$$0.8 \times \left[\frac{35.14}{\sqrt{(8.989 + 8.33) \times 1.6}} + \frac{3.62}{1.6}\right] \times 0.5 = 3.57 \text{ lb.}$$

Then

$S = 13.6 + 14.4 + 3.57 = 31.57$ *lb.*

Taking the steam-line losses at 3 per cent., the total steam consumption is

$31.57 \times 1.03 = 32.5$ *lb. per hp.-hr.*

High-pressure Crank End and Low-pressure Head End

Average admission pressure 119.7 lb. per sq.in. abs.

Cutoff = 54.2 per cent.

Effective area of high-pressure piston = 108.19 sq.in.

Effective area of low-pressure piston = 311.01 sq.in.

Cutoff in high-pressure cylinder reduced to low-pressure cylinder =

$$\frac{54.2 \times 108.19}{311.01} = 18.9 \text{ per cent.}$$

Su from Table 4 = 14.2 lb.

Condensation losses same as for head end or 14.4 lb.

Leakage losses, Sl =

$$0.8 \times \left(\frac{35.14}{\sqrt{(8.84 + 11.49) \times 1.6}} + \frac{3.62}{1.6}\right) \times 0.5 = 3.37 \text{ lb.}$$

Therefore,

$S = 14.2 + 14.4 + 3.37 = 31.97$ *lb.*

The steam-line losses are taken at 3 per cent.; hence the total steam consumption is

$31.97 \times 1.03 = 32.93$ *lb.*

The average steam consumption for the entire pump, condensing is

$$\frac{32.5 + 32.93}{2} = 32.72 \text{ lb.}$$

Noncondensing, or as operated, it is

$$\frac{32.72}{0.75} = 43.66 \text{ lb. per i.hp.-hr.}$$

As this pump was practically new when the diagrams were taken, the leakage loss is assumed as 50 per cent. of the usual figure.

Example 3—A 22 and 36 by 42-in. compound Corliss engine, noncondensing.

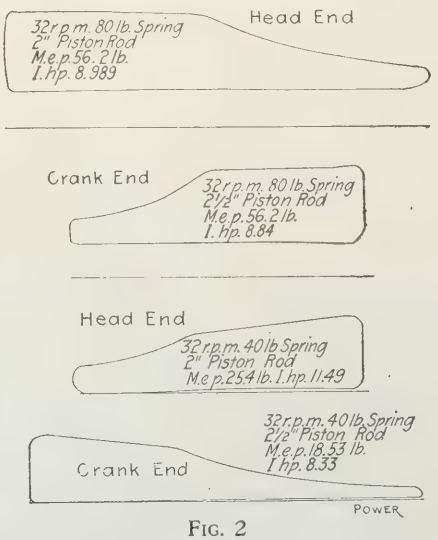


Fig. 2

High-pressure Head End and Low-pressure Crank End

Average admission pressure = 134.7 lb. absolute.

Cutoff = 17.5 per cent.

Effective area of high-pressure piston = 380.13 sq.in.

Effective area of low-pressure piston = 1004.51 sq.in.

Cutoff in high-pressure reduced to low-pressure cylinder =

$$\frac{17.5 \times 380.13}{1004.51} = 6.6 \text{ per cent.}$$

Su from Table 4 = 11.05 lb.

Condensation losses:

$$\frac{s}{d} = 2 \text{ (approximately)}$$

$$K = Sc\sqrt{P} = 19.95$$

$$P = \frac{2 \times 3.5 \times 112}{60} = 13.07$$

and

$$\sqrt{P} = \sqrt{13.07} \text{ or } 3.618$$

hence,

$$Sc = \frac{19.95}{3.618} = 5.52 \text{ lb.}$$

Leakage losses, Sl =

$$\left(0.8 \times \frac{35.14}{\sqrt{(180.89 + 93.28) \times 13.07}} + \frac{3.62}{13.07}\right) = 0.69 \text{ lb.}$$

Therefore,

$S = 11.05 + 5.52 + 0.69 = 17.26$ *lb.*

With the steam-line losses taken at 4 per cent., the total steam consumption is $17.26 \times 1.04 = 17.95$ *lb. per i.hp.-hr.*

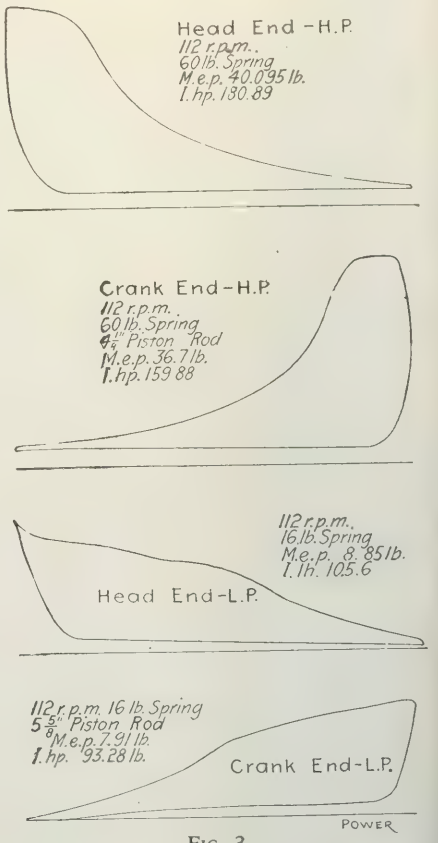


Fig. 3

High-pressure Crank End and Low-pressure Head End

Average admission pressure = 120 lb. absolute.

Cutoff = 16.4 per cent.

Effective area of high-pressure piston = 366.77 sq.in.

Effective area of low-pressure piston = 1004.51 sq.in.

Cutoff in high-pressure cylinder reduced to low-pressure cylinder =

$$\frac{16.4 \times 366.77}{1004.51} = 6 \text{ per cent.}$$

Su from Table 4 = 11.14 lb.

Condensation losses same as for head end or 5.52 lb.

Leakage losses Sl =

$$0.8 \times \left(\frac{35.14}{\sqrt{(159.88 + 105.2) \times 13.07}} + \frac{3.62}{13.07}\right) = 0.70 \text{ lb.}$$

*The opposite ends of the high- and low-pressure cylinders are added because immediately after the steam exhausts from the head end of the high-pressure cylinder, steam is admitted into the crank end of the low-pressure cylinder.

$S = 11.14 + 5.52 + 0.7 = 17.36$ *lb.*

With the steam-line losses taken at 4 per cent., the total steam consumption is $17.36 \times 1.04 = 18.05$ *lb. per i.hp.-hr.*

The average steam consumption for the entire engine condensing is 18 lb., while noncondensing, or as operated, it is

$$\frac{18}{0.75} = 24 \text{ lb. per i.hp.-hr.}$$

Sterling Model D Sight Feed Oil Pump

This pump provides a variable crank speed, and each pumping unit has two plungers. The primary plunger draws the oil from the reservoir, forcing it through the sight-feed whence it is

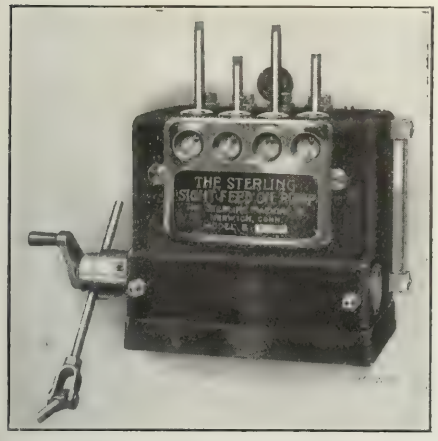


FIG. 1. SIGHT FEED PUMP

pumped to the point of discharge by the secondary or high-pressure plunger. Each plunger and sight-feed unit is complete and independent of the other, and may be removed from the reservoir without draining off the oil, or stopping the other pumping units.

The stuffing-boxes of the high-pressure plungers are back of a removable plate, and, although concealed and protected from dust and dirt, are accessible and easily packed.

By manipulating a knurled nut, the primary or circulating pump can be set to deliver a measured amount of oil at each stroke. The driving mechanism is steel, hardened where practicable and inclosed. The pump is shown in Fig. 1.

Fig. 2 is a section of the oil pump. In operation the oil in the reservoir *A* is drawn in by the circulating pump for the sight-feeds. The tube *B* is fastened to the adjusting screw *C* and is stationary during the operation of the pump. The cam *D*, fastened on the shaft *E*, causes the yoke *F* to reciprocate vertically as the shaft rotates, thus covering and uncovering the port *G* which allows oil to flow into the chamber *H*. With every upward stroke oil is forced past the small check *I*, through the tube *B* and ports *J*, *K* and *L*, to the disk nozzle *M*, from whence it drops into the chamber *N* back of the sight-feed glass *O*, which enables the drop to be seen.

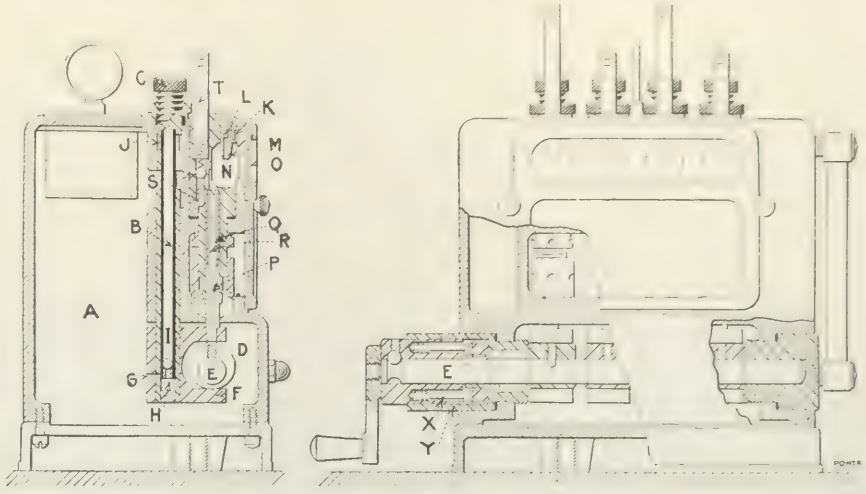


FIG. 2. SECTIONS THROUGH OIL PUMP

Oil is then drawn into the force pump of the system. The main plunger *P* has a small plunger or valve *Q*, having a limited motion within the main plunger. As the main plunger goes down, this small plunger *Q* keeps the port closed until near the bottom of the stroke, when it is opened suddenly and draws in oil regardless of its viscosity. The plunger then starts upward, and as the oil is in the bottom of the chamber *R*, any air which may be in the chamber is forced out. The valve or small plunger *Q* then seats, and the main plunger forces the oil through the double checks *SS* to the standpipe *T*, and into the line, through the check valve which prevents any steam backing into the lubricator.

A driving sleeve carries four pawls *X*, which engage with the ratchet *Y*, keyed to the shaft *E*. Any angle drive may be obtained and any amount of angular movement from 7 to 360 deg. The pump is also fitted with a crank whereby the pump may be flooded by hand without disarranging the adjustment of the nozzle cam.

The actuating cam of each feed may be set at an angle independent from the other cams, or any number of feeds may be operated at the same time by the same cam, such cam being located on the main shaft of the lubricator. Each pumping unit, also including the secondary or forcing pumps, may be unscrewed and raised from the reservoir without interfering with the work of the other units. These features make the pumps especially desirable for timing the lubrication of gas engines, in which case the pump may be operated by either a sprocket mounted on the main shaft, chain driven, or by bevel-gears. This pump is made by the Sterling Machine Co., Norwich, Conn.

Fireman's Favorite Tube Scraper

The "Fireman's Favorite," tube scraper is an improvement over the old style known as the "Engineer's Favorite," in that the scraping arms are steel; those in the old style were malleable iron. They are formed in the shape of a scoop, as illustrated in Fig. 1. The main features of the old scraper are retained.

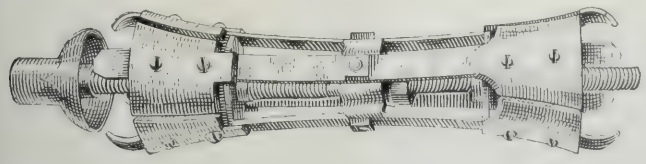


FIG. 1. TUBE SCRAPER

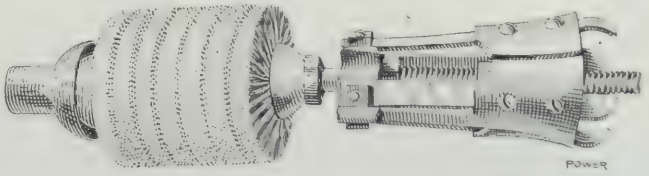


FIG. 2. BRUSH AND SCRAPER

The scraper is also made with a brush, as in Fig. 2. A. W. Chesterton Co., 64 India St., Boston, Mass., manufactures the scrapers.

Bessemer steel was first made in this country in September, 1864, by William F. Durfee, at an experimental plant at Wyandotte, Mich., and openhearth steel in 1864 by the New Jersey Steel & Iron Co., at Trenton.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

The Magneto and the Spark Coil

By Joseph B. Baker

The electric spark for igniting the mixture in an internal-combustion engine may be produced by the magneto (short for magneto-electric generator) or by the spark coil or induction coil, both operating by dynamo-electric induction.

Dynamo-electric Induction

To understand how the magneto and spark coil work, it is necessary first to understand how dynamo-electric currents are generated. A simple conductor or wire, *AB* (Fig. 1), through which a current is flowing in the direction indicated by the large arrow, is surrounded by a magnetic field which at any point has the direction, relative to the direction of the current, indicated by the small arrows. This relation between a current and its surrounding field is analogous to the advance and rotation of a corkscrew. The field is strongest near the wire and grows weaker with the distance from the wire, as indicated in Fig. 2. The field of a current varying in strength, growing alternately stronger and weaker, is surrounded by a bundle of lines of force which alternately expand out from the wire and collapse in upon it, the strength of the current corresponding at any instant to that of the magnetic flux surrounding the wire and varying with the rate of change of the flux.

Now let the wire C_1, Fig. 3, be brought within the field of a fixed wire C carrying a current. The outermost of these lines will begin to cut C_1 and will induce an electromotive force in C_1, causing a current to flow through the latter (if a closed circuit). This induced current in C_1 will flow as long as the number of lines threading through the circuit of C_1 is being added to, as by the continued encroachment upon the field of C; it will cease when this foreign field stops growing in C_1's territory. If the foreign field begins to diminish an induced current will flow in C_1 in the reverse direction.

The foreign field may be that of a magnet. Thus the coil of wire in Fig. 4 will have a current induced in it by any relative movement of the coil and the near-by magnet.

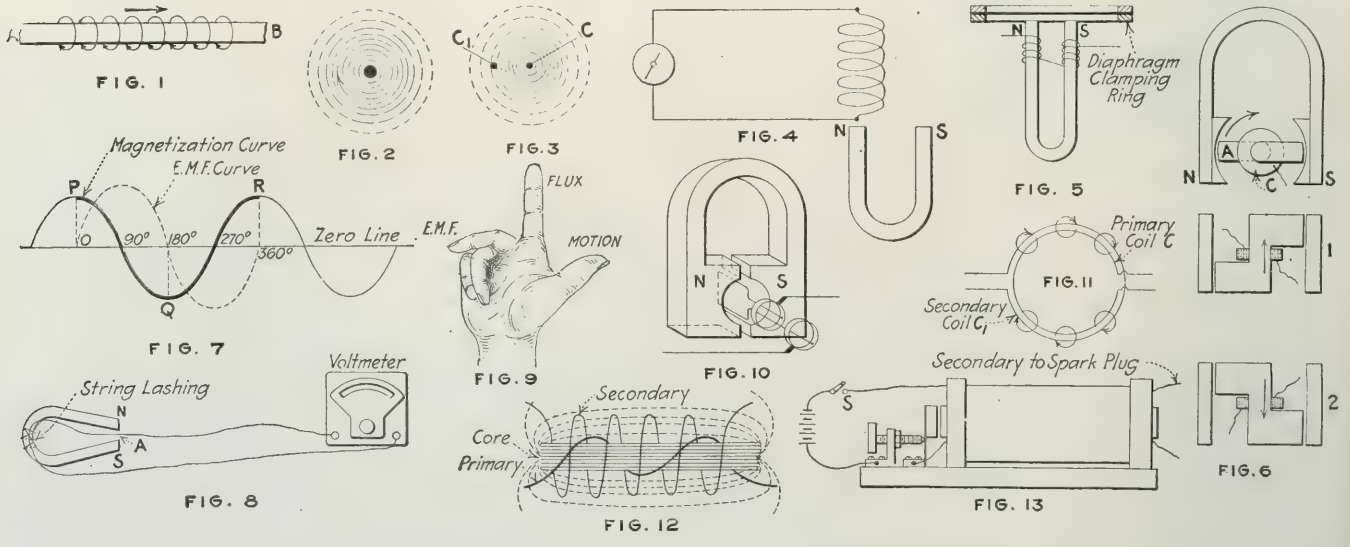
Note that it is the variation in position or strength of the field that produces these effects. If in Fig. 3 both C and C_1 are stationary a field built up around C by connecting a battery to this wire will, as it expands outward from C, encroach on C_1's territory and will therefore induce a current in C_1. If this inducing current in C is interrupted, the collapse of the field will induce a reverse current in C_1.

These experiments illustrate dynamo-electric induction as the inducing of currents in a circuit by any change in the strength or direction of the flux threading through the circuit, the effect being proportional to the rate of change in the flux.

Magneto-electric induction, the basis of the magneto, is the generation of an electromotive force in a conductor by the relative movement between a field and the conductor, so that the flux sweeps through the conductor, or *vice versa*. In electromagnetic induction, which is the basis of the spark coil, the electromotive force is generated by the varying flux which is in turn produced by varying the current in a fixed conductor mounted in inductive relation to another fixed conductor.

The Inductor Magneto

Connect an ordinary telephone receiver to a low-reading voltmeter by wires several feet long, so that the telephone



Principles of the Magneto and the Spark Coil Illustrated

Magneto-electric induction, the basis of the magneto, is the generation of an electromotive force in a conductor by the relative movement between a field and the conductor, so that the flux sweeps through the conductor, or *vice versa*. In electromagnetic induction, which is the basis of the spark coil, the electromotive force is generated by the varying flux which is in turn produced by varying the current in a fixed conductor mounted in inductive relation to another fixed conductor.

The Inductor Magneto

Connect an ordinary telephone receiver to a low-reading voltmeter by wires several feet long, so that the telephone receiver consists of a steel magnet with its poles opposite a flexible iron disk or diaphragm and inclosed by coils of insulated wire (Fig. 5). Lines of force (not shown) extend from pole to pole of the magnet, taking a certain arrangement determined by the close proximity of the iron disk. The amount of the total flux that threads completely through the two coils and goes into the disk, as compared with the amount of the flux that passes directly across from pole to pole, will depend on the distance of the disk from the poles and will be changed by any to-and-fro movement of the disk. While watching the voltmeter place the thumb on the telephone disk and press it suddenly in. The movement of the disk deflects the

pointer momentarily; and on suddenly releasing the disk the pointer moves again but in the opposite direction. By sharply tapping with the end of a lead pencil, so as to move the disk more quickly, larger deflections are caused. In moving the iron disk in the field of a permanent magnet, thereby causing the latter's flux to cut the turns of the coils, is illustrated one method of magneto-electric generation of current.

Now perform this experiment in another way, that is, with the apparatus shown in Fig. 6, which constitutes the essentials of an inductor magneto. A Z-shaped inductor armature *A* is mounted on a shaft and rotated in the interpolar space of the magnet *NS*. Surrounding the middle part of the inductor armature is a stationary coil *C*. It is clear that the magnet's field will pass through the armature (and, therefore, through the middle of the coil) in one direction in the plan view 1 and in the opposite direction in the plan view 2, showing the inductor armature 180 deg. later. Therefore, when the armature is rotated rapidly an alternating current is generated in the coil by the forcing of a rapidly changing flux through the coil. These changes in strength and direction are indicated in the full-line curve, Fig. 7, of which the portion *PQR* corresponds to one complete revolution of the inductor armature. The height of the initial point *P* above the zero line indicates the flux when the armature is in the position of view 1, Fig. 6, allowing the maximum number of lines of force to thread through the coil in the direction indicated. As the armature is rotated, the widening of the gaps between it and the polepieces chokes back the flux traversing the moving iron more and more until this flux falls to zero in the position at right angles to that of view 1 (the 90-deg. point on the curve). As the rotation continues the field begins to drive a flux through the armature in the reverse direction; this flux attaining a maximum value in the 180-deg. position represented by view 2, Fig. 6, and corresponding to the point *Q* on the curve.

By this alternation of the armature flux, an alternating electromotive force is generated, having a number of double reversal cycles per minute equal to the number of revolutions per minute of the armature. The maximum values of the induced electromotive force in both directions correspond to the positions of the inductor armature where the rate of change of the flux traversing the coil is at its maximum; that is, at the point where the full-line curve crosses the zero line in the reversal of the flux. On the other hand, the electromotive force will pass through its zero value (in reversing) at the point where the rate of change of the flux through the armature is zero, that is, at the points where this curve is at its maximum. Therefore, the dotted line curve is shifted along 90 deg. from the magnetization curve and represents the alternating electromotive force. Thus the alternating electromotive force follows the alternating flux at a "phase displacement" of one-quarter period.

The Wound-armature Magneto

Let Fig. 8 represent an experiment based on the concept of Fig. 3; that is, the cutting of a flux by a moving conductor. If the wire is tapped sharply at the point *A*, between the poles of the magnet, the pointer of the voltmeter will deflect momentarily—the direction and amount of the deflection depending upon the direction and rapidity of the movement of the wire. As in the experiment with the telephone receiver and voltmeter the electromotive force induced depends upon the rate of change in the number of lines of force inclosed by the circuit of the wire. The directions of the magnetic field, of the electromotive force induced and of the motion of the conductor are all at right angles to each other, and their directions may be represented by the thumb and first two fingers of the right hand, as in Fig. 9.

The experiment just described is utilized in the construction shown in Fig. 10. The wire-frame armature, revolving on a shaft (not shown) coaxial with the interpolar space of the magnet *NS* will inclose a varying number of lines of force; as this armature revolves the plane will be inclined more and more to the direction of the flux, fewer and fewer lines of force being threaded through it. At the end of one-quarter of a revolution from the vertical position shown no lines of force will be threading through the armature; as the revolution proceeds the flux through the armature will increase, reaching a maximum when the armature has reached the vertical position, and so on. An alternating electromotive force will therefore be delivered at the two collector rings in a cycle (similar to that represented in Fig. 7).

In the construction of actual wound-armature magnetos the efficiency of the armature is greatly increased by a shuttle-shaped iron core. The alternating currents generated as this armature is revolved will be very much stronger than in an armature without iron on account of the low-reluctance iron path. In ignition magnetos built on this principle, whether of the so called "low-tension" or "high-tension" type, several field magnets are employed, and the armature winding consists of a large number of turns. The voltage is therefore proportional to the strength of field, the speed of the armature and the number of turns in the armature. In the low-tension magneto the armature winding is of few turns of heavy wire, generating a large current at low voltage. The high-tension magneto may be a "primary armature" machine having a long, fine-wire armature winding, or it may be a "compound armature" machine having a long, fine-wire winding and also a low resistance secondary winding and a condenser.

Certain nonrotating magnetos have been designed to generate electromotive force for a single igniting spark instead of a continuously alternating electromotive force. In the oscillating type of machine a single properly timed impulse is given by a short oscillation through the maximum electromotive force portion of a revolution only. Several types of "pull-off armature" magneto are also made in which the single spark is given by suddenly jerking an armature off the polepieces of a specially designed magnet. Two-spark magnetos with a double secondary winding are also made.

In the usual ignition magneto, instead of the collector rings shown in Fig. 10, one terminal of the armature winding is grounded on the core and the other is brought out to an insulated stud. In the design of bipolar machines for generating a direct current, the two terminals are brought out to a split-ring commutator, which at every half revolution reverses the electromotive force delivered to the external circuit; this is essentially a dynamo, having an electromagnetic field. Dynamos built on this principle and used for ignition on automobiles usually have continuous armature windings consisting of a number of coils connected in series and with taps brought to a corresponding number of commutator segments. The electrical output of these machines is high compared with that of a magneto machine of the same size.

The Spark Coil

If a current is made to flow through the one-turn coil *C*, Fig. 11, the accompanying field will loop through C_1 also, as indicated. Electromotive force surges through the secondary for every change of current strength in the primary, the induced electromotive force being in one direction for an increasing primary current and in the other for a decreasing primary current; its strength depends on the *rate of change* of the strength of the primary current. In practice, the induction coil (Fig. 12) has a primary of a few turns of heavy wire, a secondary of many turns of fine wire, and a core of soft iron wire; the flux (indicated by the dotted line) being greatly increased by the iron core. When the primary is connected to a source of varying current of low voltage, an alternating current of high voltage is generated in the secondary by the alternate expansion and collapse of the flux, in accordance with the ratio of the number of turns of the two coils. For ignition work the coil is designed to transform direct-current energy at 4 to 8 volts into alternating-current energy at the necessary high elec-

tromotive force to give a jump spark in the engine cylinder under working conditions.

The variation in the primary current necessary to give a continuous series of sparks at the terminals of the secondary is usually given by a trembler interrupter mounted opposite one end of the soft iron core, the connections being as shown in Fig. 13. The contact screw is adjusted to touch the hammer so that on closing the switch the battery sends current through the contact to the primary. The resulting magnetization of the core attracts the hammer, causing its spring to open the contact; but this interruption of the current demagnetizes the core, allowing the hammer to spring back, this make-and-break action is repeated as long as the battery circuit is closed, giving a series of discharges at the spark plug.

A condenser which forms part of the spark coil serves to render the make-and-break more abrupt, and also to protect the platinum trembler contacts from the burning effect produced by the contact spark or the arc occurring at every break.

Mention should be made of a special utilization of the spark coil to give a single good igniting spark in the engine cylinder at the proper instant, instead of a series of sparks, by a mechanical interrupter operated by a cam driven from the engine instead of the continuously operating trembler interrupter.

Five New Mazda Lamps

Formerly the only round-bulb Edison Mazda lamps available for 100- to 130-volt circuits were the 15-, 25-, 40-, 60-, 400- and 500-watt sizes. On Oct. 1, the General Electric Co. added two new intermediate sizes, namely, the 100-watt and 150-watt sizes. Also for use on 200- to 260-volt circuits they have added 25- and 100-watt sizes to the existing sizes of 40 and 60 watts. This gives the user nearly the same wide range of sizes now available in the standard or pear-shaped bulbs.

Another new lamp, put on the market at the same time, is the "concentrated light source" Edison Mazda lamp. This lamp has a concentrated filament and will be used on 100-130-volt circuits wherever a concentrated light source is desired. To obtain the necessary concentrated light source the wire filament is formed into a helix of small diameter somewhat resembling a closely coiled spring; this is then mounted in a space approximately ½ in. each way, thus giving a greatly concentrated light source. The lamp operates at the high efficiency of 1.23 watts per mean horizontal candlepower, and is designed to give a life of approximately 250 hr. at its rated efficiency.

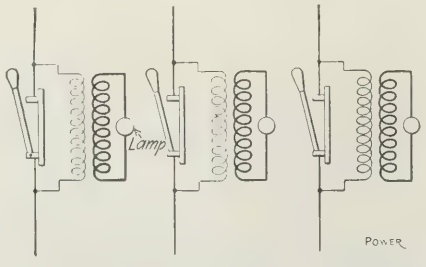
CORRESPONDENCE

Inspecting and Testing Electrical Apparatus

In Mr. Cook's article, in the Aug. 13 issue, methods are given for testing out phases on alternating-current generators, or other apparatus. For this purpose lamps are shown connected in series with each leg of the circuit for low-voltage work and transformers connected across the respective circuits for high-voltage work; in the latter case the lamps are connected between the two sets of transformers in the secondary circuits.

The connection given for low-voltage work, using lamps, is correct; that given for the transformers is uncertain, as the transformers may not all be wound and connected to their respective terminals alike; that is, one may be crossed as compared to some other, which condition would cause a wrong indication at the lamps. Another bad feature is the large number of connections tending to make the method confusing. Also all opposing transformers must be alike; in practice it is not always possible to obtain the required number of transformers, for such a test, which fulfill this requirement.

A method which is much more simple and which cannot be affected in its results, by unlike internal connections of the transformers, and in which, transformers with unlike ratios can be used if necessary, is herewith described.



TRANSFORMERS IN SERIES WITH EACH LEG OF CIRCUIT

One potential transformer is required for each leg of the circuit, the primary wound for the normal working voltage and all the transformers to have the same ratio if possible. Each transformer has a lamp (or lamps in series) in its secondary circuit to correspond to the normal voltage of that circuit. With the proper number of transformers prepared as outlined, connect the primary coil of one in series with each leg of the circuit as shown; this rule to be followed regardless of the number of wires in the circuit. Synchronism will be indicated when all the lamps are dark.

If necessary to use transformers with unlike ratios the load on the secondary circuits must be so adjusted that the primary current will be the same in each.

There are certain conditions under which this method may be varied but for ordinary conditions it is better to follow the rule as stated and use similar transformers if available.

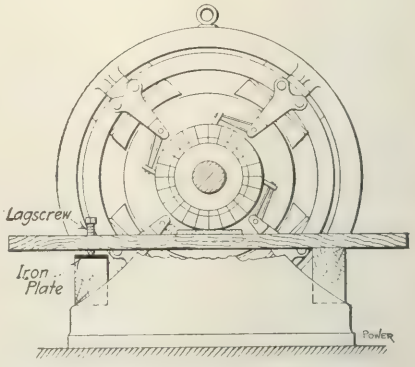
C. W. BELL.

Scranton, Penn.

Commutator Grinding Device

The illustration shows an apparatus for truing commutators, which has been found very convenient in plants whose equipment does not include a commutator grinder.

The apparatus is composed of a sandstone slab about 2½x6 in., cut from an old grindstone, a 4x4-in. timber which has a mortised slot to receive this slab, two blocks placed on the frame of the machine upon which the timber rests, a lagscrew and an iron plate. The blocks are of such height that the sandstone slab just clears the commutator when in position. The lagscrew is screwed into one end of the timber and its point rests on the iron plate, as shown.



GRINDING DEVICE IN POSITION

In use the brushes, or at least the bottom set, are removed and the machine is brought up to speed. The lagscrew is then screwed in until the sandstone touches the commutator. A man at each end of the timber pushes it back and forth lengthwise with the commutator and also in the direction of its own length; the latter movement is to keep the face of the stone worn evenly. The lagscrew is tightened and the movement continued until the commutator is ground round and smooth.

The timber and blocks must be smooth to prevent the timber from rocking. It must also be stiff enough to prevent springing or the commutator will not be round. Collector rings can be trued with this apparatus equally well. It is especially adapted to commutators with under-cut mica even when it is convenient to take the armature to a lathe, for attempts to turn this type of commutator on a lathe are not always successful.

JAMES SPIVENS.

Albuquerque, N. M.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

A New Gas Engine Design

To avoid the difficulties in making complicated cylinder castings which shall be free from cracking and internal strains, and to eliminate troubles due to faulty valves and valve-gears, many designers have directed their attention to simplicity

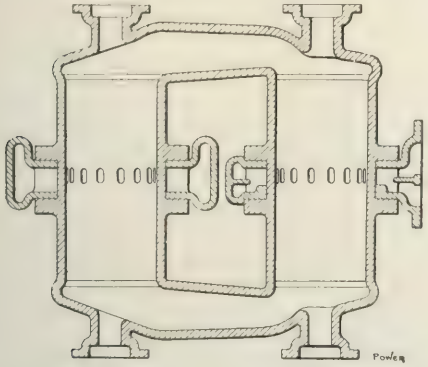


FIG. 1. SECTION THROUGH CYLINDERS

in design. One of the most successful attempts in this respect is the Oechelhaeuser engine, in which a simple cylinder is employed, open at both ends, with two pistons coupled to the same crankshaft by a piston rod and side rods. Not

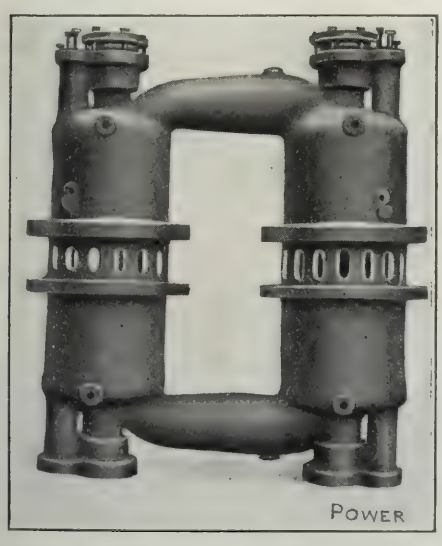


FIG. 2. CYLINDERS READY TO BE MOUNTED

only is the cylinder casting thus reduced to its most elementary form, but also, by making the pistons cover and uncover ports in the cylinder walls, valves are done away with altogether. Unfortunately, there are objections to this design which have prevented its general adoption.

The principle, however, was too good to be abandoned, and at the meeting of the Iron and Steel Institute of England in October a new method of carrying it into practice was described by A. E. L. Chorlton. The design which he has developed possesses features of remarkable ingenuity, and represents a distinct departure from current practice.

Referring to Fig. 1, it will be seen that two cylinders are employed, or a multiple of two, each pair being connected at the top and bottom by a trunk. Each pair of cylinders is cast in halves, which are accurately fitted together in a plane passing through the ports, and the castings are free from awkward corners, pockets, etc., which would interfere with uniform expansion at high temperatures. Long pistons are employed, with piston rods

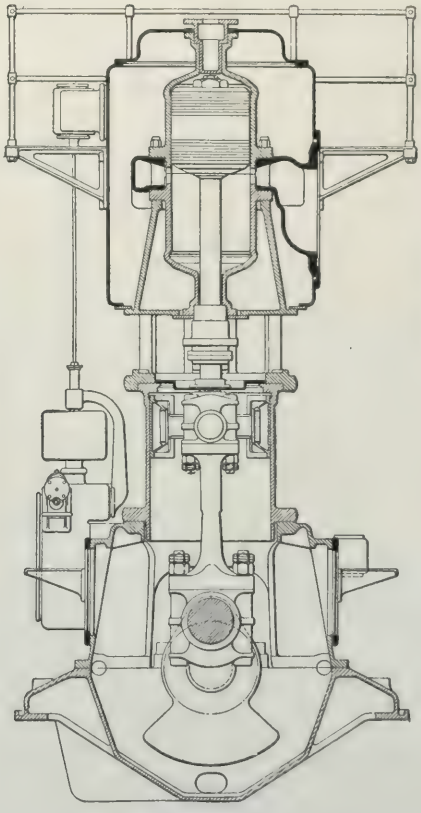


FIG. 3. END SECTION OF ENGINE

passing through the stuffing-boxes at the lower ends; the boxes at the upper ends are blocked up, as shown in the section, Fig. 3. Around the middle portion of each cylinder is a circular box, bolted to flanges and serving as inlet and exhaust chambers. The combined cylinders rest by their flanges on stools in a large tank, which forms a water jacket.

The engine works on the two-stroke-cycle, and the action is as follows: The left-hand crank, connected with the piston controlling the exhaust ports, is set slightly in advance of the other, so that the exhaust ports are uncovered before the inlet ports, releasing the burnt gases. The inlet ports are then uncovered, and a

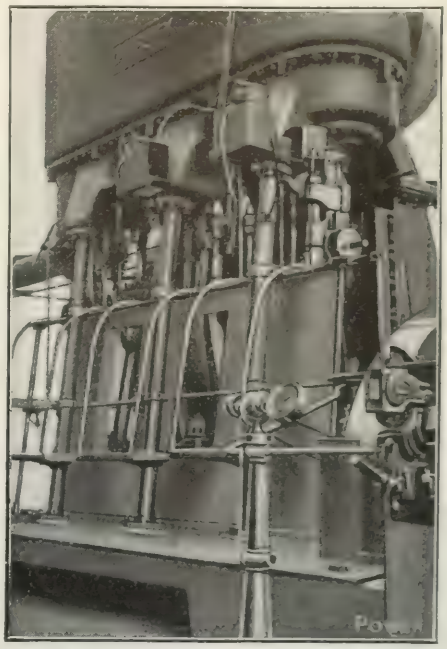


FIG. 4. ASSEMBLED ENGINE OF THE "DUPLEX TYPE"

scavenging charge of pure air is blown through the cylinders by a pump, followed by the fresh charge of gas and air as the exhaust ports close. The inlet ports then close, and compression takes place, followed by ignition and expansion in the usual way. This cycle occurs in each end of the cylinders during every revolution, so that the maximum possible output is obtained from a given pair of cylinders.

The vertical type was chosen because, besides greatly reducing the floor-space occupied, the cost is about 20 per cent. less than that of the horizontal type; standard steam-engine practice has been followed in the rest of the design. Mr. Chorlton points out that a twin duplex engine of this type, with four cylinders, has only four moving parts subjected to high temperatures, as compared with 20 in the case of an equivalent twin tandem four-stroke-cycle engine, and it can be run at a higher temperature, even with the jacket water boiling. All difficulties

with the jacket are overcome, as the water has free and direct access to all parts exposed to high temperatures, and the outside of the cylinder is accessible for cleaning and inspection.

The author claims that engines of this type can be built for very large outputs; Fig. 2 is a section through one of the cylinders of a large "duplex" engine having four cylinders, each 36 in. in diameter by 43 in. stroke, and running at 140 r.p.m. The flywheel is 12 ft. 3 in. in diameter and weighs 60 long tons, and the floor-space occupied is 475 sq.ft. The mechanical efficiency of an engine of this type having cylinders 15½ in. in diameter by 18 in. stroke was 85 per cent.

The Junkers Engine—IV

By F. E. Junge

The action of forces in side rods and interconnections will be discussed first and then followed by a study of the corresponding actions in the crank mechanism of Diesel and Junkers engines. The side-rod mechanism of the Junkers engine is made up of the following parts, proceeding from the crankshaft:

1. The inside or vertical mechanism; see Fig. 18, consisting of the crosshead traverse piece *G*, the side rods *E* and *F*, the head traverse piece *H*, and the piston rods *A* and *B*.

2. The outside or horizontal mechanism; see Fig. 19, consisting of the side rods *K* and *L*, the traverse piece *I* and the piston rods *C* and *D*. The acting forces are balanced within the mechanism itself. The rods *EF* of the inner mechanism are always in tension and can, therefore, be dimensioned independent of their length. The rods *K* and *L*, of the outside mechanism, however, are alternately in tension and compression. In calculating the buckling loads for these rods, their short length is to their advantage. To avoid unnecessary weight these rods are turned out hollow.

The side rods are guided in such a way that torsional or lateral vibrations cannot be set up, but longitudinal oscillations may be excited in the rods *E*, *F*, *K* and *L*. These vibrations are guarded against by proportioning the cross-sections of the rods and the masses of the pistons with respect to the normal number of revolutions of the engine.

The critical speed of the engine is determined as follows, for the rods *EF*:

$$n = \frac{30}{\pi}\sqrt{\frac{2\,EF}{L}\left(\frac{1}{m_1}+\frac{1}{m_2}\right)}$$

where

n = Critical number of revolutions per minute of the engine;

E = Modulus of elasticity of the material of the rods;

F = Cross-sectional area of the rods;

L = Free length of the rods;

m_1 and m_2 = Masses of pistons *A* and *B* respectively.

For the rods *KL*

$$n = \frac{30}{\pi}\sqrt{\frac{2\,Ef}{l\,m}}$$

where

f = Cross-sectional area of the rods;

l = Free length of the rods;

m = Aggregate mass of the pistons *C* and *D*.

In practice n must be chosen from 30 to 50 r.p.m. above or below the normal speed of the engine.

Forces in the Diesel Engine

The gas pressure acts on the piston and the cylinder cover with the force *P*; see Fig. 20. The pressure *P* on the piston may be resolved at the crosshead into the pressure *N* on the slides and the connecting-rod component *S*. The force *S* in the connecting-rod is resolved at the crankpin into the tangential or turning force *T* and the radial force *R*.

Without affecting the equilibrium, two equal and opposite forces, each equal to and acting along the same line as *T*, may be applied at the center of the crankshaft. This results in:

1. A couple *Tr*, identical with the turning effort of the engine.

2. The force *T* remaining over at the crank center, when combined with the radial force *R*, supplies the connecting-rod force *S* at the same point.

3. The force *S* is resolved into the force *N*, equal, opposite and parallel to the pressure *N* on the guides, and the force *P*, equal to the piston pressure *P*.

From this it will be seen that the force couple *NR* is always equal to the couple *Tr*; that is, the frame reaction moment and the crank effort are always in equilibrium. If the engine possesses a strongly fluctuating crank effort, an equal and opposite moment is set up tending to rock the engine and its foundation accordingly.

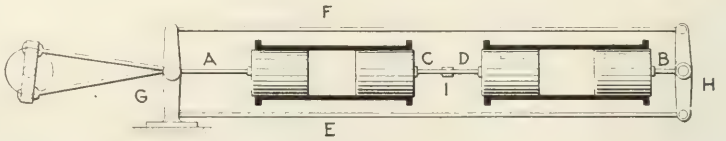


FIG. 18

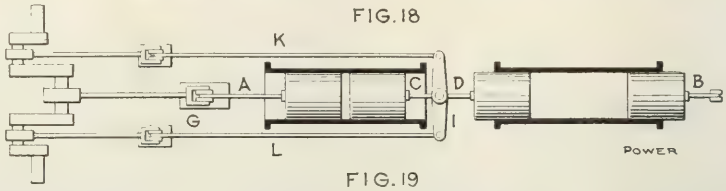


FIG. 19

Connections between Pistons and Crankshaft in Junkers Engine

The force *P* mentioned is balanced by the force *P* acting on the cylinder cover, through the medium of the engine frame and cylinder. Since the gas pressure cannot shift the bedplate in any way, it can only set up internal forces, namely, the forces *P*, tending to pull apart or compress the engine frame and the cylinder, and the bending moments *Ph* which at every part of the frame and cylinder tend to buckle or snap these elements (*h* being the distance of the center of gravity from the center line of the engine).

Due to the periodic nature of the forces and moments mentioned, all the respective parts are exposed to periods of vibration. This exacts a dissipation of energy necessary for the absorption of the oscillations.

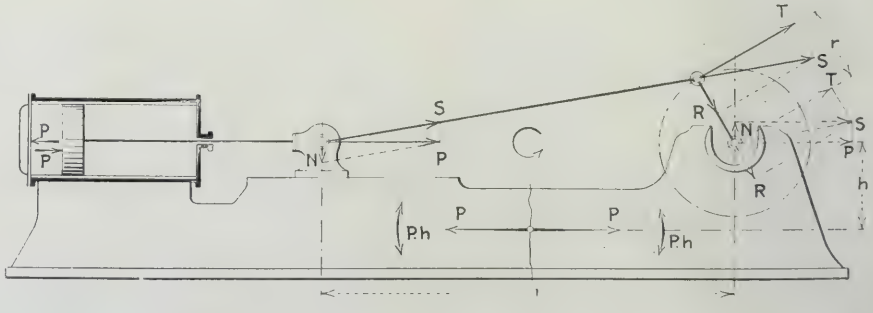


Fig. 20. Action of Forces in Diesel Engine

Forces in the Junkers Engine

Imagine the crank of the ordinary steam-engine mechanism to be reproduced by a distance equal to its own length, on the other side of the crank center. Thus two cranks of equal length set at 180

deg. to each other are obtained. If the new crank be supplied with a connecting-rod, placed opposite to the already existing one, the mechanism of the Junkers engine results. The mechanism of the Junkers engine may, therefore, be virtually considered to be made up of two ordinary steam-engine mechanisms superimposed, the cranks of the latter set at 180 deg., and the connecting-rods disposed on the same side of the crankpin circle; see Fig. 21.

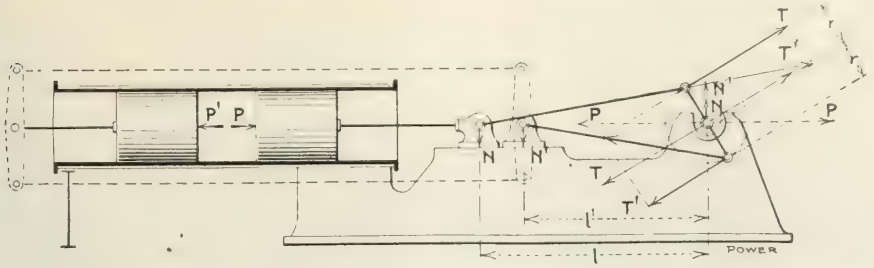


FIG. 21. ACTION OF FORCES IN JUNKERS ENGINE

The force actions are then obviously as follows. For the front piston:

1. A tangential force T.
2. A pressure on the guides N.
3. A force N equal, but opposite, to the pressure on the guides; this force with the latter forms the couple Nl.
4. At the crank center a force of equal value, but opposite in direction to the tangential force T; both forces supplying the couple Tr.
5. A force P at the crank center equal in value and direction to the piston pressure P.

For the rear piston the forces are analogous, namely, the gas pressure P on the rear piston is transmitted as $\frac{P}{2}$ to each of the side rods, by its traverse piece. At the gudgeon pin of the second mechanism these again combine to equal P. Thus results:

1. Pressure N' on the guides.
2. A tangential force T'.
3. Force N' equal, but opposite in direction, to the pressure on the guides. This, and the latter force, form the force couple $N'l'$.
4. A force at the crank center of equal value, but opposite in direction to the tangential force T'. Both forces supply the couple $T'r'$.
5. A force at the crank center equal in value and direction to piston force P.

The couples Nl and $N'l'$ for both mechanisms, augment one another, the same holding true for the turning moments Tr and $T'r'$. Hence

$$Nl + N'l' = (T + T')\, r$$

The piston forces P, mentioned under 5, neutralize each other exactly. Instead of the interlocking of the internal forces being accomplished by means of the frame, as is the case with the ordinary engine, it is effected in the mechanism itself, by the side rods, traverse pieces and crankshaft.

The reaction moments at the frame are, of course, retained. This is a peculiarity of every engine, for whenever an engine produces a turning moment, the engine itself must bear back with an equal moment. However, the existence of these moments is of little importance in arrangements of several units of the Junkers type, for, owing to the excellent balance of the reciprocating parts, the positions of the cranks are not restricted to any special relative spacing in order to attain favorable conditions in regard to the inertia balancing, as is the case in other types. The cranks can, therefore, be set at angles producing the most uniform turning moment, without reference to balancing at all.

PRESSURE REVERSAL

In Fig. 22 the resulting piston pressures for a Junkers engine are plotted (tandem arrangement). An indicator diagram showing a pressure of 590 lb. per sq.in. during combustion is taken as a basis. The pressure due to inertia is assumed at 295 lb. per sq.in. for infinitely long connecting-rods. As will appear from the diagram, the pressure reversal in the Junkers engine takes place shortly before the dead center is reached. This is what is desired and corresponds to the practice followed in steam engines.

In the single-acting Diesel engine, radically different conditions are met. The nature of the resulting pressure curve is shown in Fig. 23, which is based on the same data as in the preceding case. On the forward or expansion stroke no pressure reversal takes place at all. Hence the inertia curve nowhere intersects the expansion line. This, of course, is only the case when the value of the inertia pressure is kept low enough. At higher values of the inertia of the reciprocating parts, one pressure reversal may take place when the expansion line is just touched by the inertia curve. Two pressure reversals will, however, take place as soon as the curves intersect.

On the other hand, during the back or compression stroke the resulting pressure changes its direction twice; once near mid-stroke, and again almost at the dead center. These reversals could only be obviated by considerably decreasing either the inertia pressure or the number of revolutions. This decrease would have to be of such an extent, as to let the inertia curve either touch the expansion line or keep quite clear of it. In the first instance one change of pressure would still have to be reckoned with, while in the second there would be none to fear. Yet, a diminution of the necessary range is impracticable. Thus in practice one is forced to put up with the trying reversal near mid-stroke.

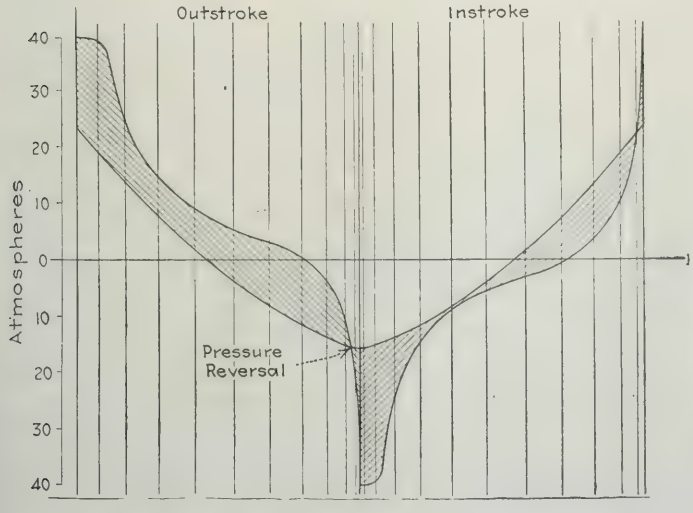


FIG. 22. RESULTING PRESSURES IN JUNKERS ENGINE

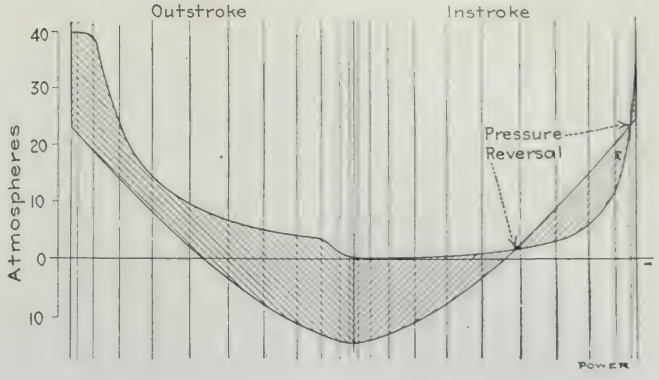


FIG. 23. RESULTING PRESSURES IN DIESEL ENGINE

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Forced Draft Control at Ritz-Carlton Hotel

By Charles A. Fuller

In the Mar. 19 issue, the heating and ventilating system in the addition to the Ritz-Carlton Hotel, Forty-sixth St. and Madison Ave., New York City, was described. At the time when this work was designed, provision was made to increase the boiler plant to take care of the additional load. The original plant consisted of five boilers aggregating a total of about 1600 hp. of the Franklin water-tube type. Two new boilers of 300 hp. capacity each were added.

The stack was 6 ft. in diameter and 225 ft. high, which was considerably too small for the entire load. It was therefore necessary to install some form of forced or induced draft to compensate for the lack of stack capacity. The restricted head room and the location of the smoke breeching was such as to practically eliminate the use of an induced-draft system, which necessitated the selection of forced draft. It was also necessary to divide the system into two separate units, one for the new boilers and one for the old boilers, to avoid carrying ducts through the main building wall which separates the old boiler plant from the new.

For the two 300-hp. boilers, one Sturtevant No. 8 single-width bottom-discharge horizontal multivane fan is used, directly connected to a 5x7-in. vertical engine. This unit is capable of delivering 13,500 cu.ft. of air per minute against a static pressure of 1½ in. when running at 435 r.p.m.

The fan and engine are directly in the rear of the two boilers and the fan discharges into a chamber below the combustion space back of the bridge-wall. This chamber, 8 in. deep, extends from the bridge-wall to the rear boiler wall, the entire width of the setting. The bottom of the space is finished smooth with concrete and the top is formed by T-irons across the area covered with concrete and finished with paving brick. A duct 8x60 in. leads from this chamber up through the bridge-wall and discharges into the ashpit below the grates.

The fan for the old boilers is a Sturtevant No. 9 double-width top vertical-discharge, multivane type, directly connected to a 7x7-in. vertical engine. This unit can deliver 36,000 cu.ft. of air per minute against a static pressure of 1½ in. when running at 370 r.p.m. This unit was too large to place back of the boilers as in the other case, and therefore had to be located in the adjoining basement of the new building. The supply duct leads from the outlet of the fan and extends along the wall back of the boilers, of size shown in Fig. 1. The branches to each boiler are taken from the bottom of this duct and extend along the floor to a chamber below the combustion space, formed in the same manner as the ones described for the new boilers. Excavating was necessary here to avoid reducing the area of the combustion chamber. The bridge-walls were torn down and rebuilt in each of the boilers to form the duct leading up to the ashpit.

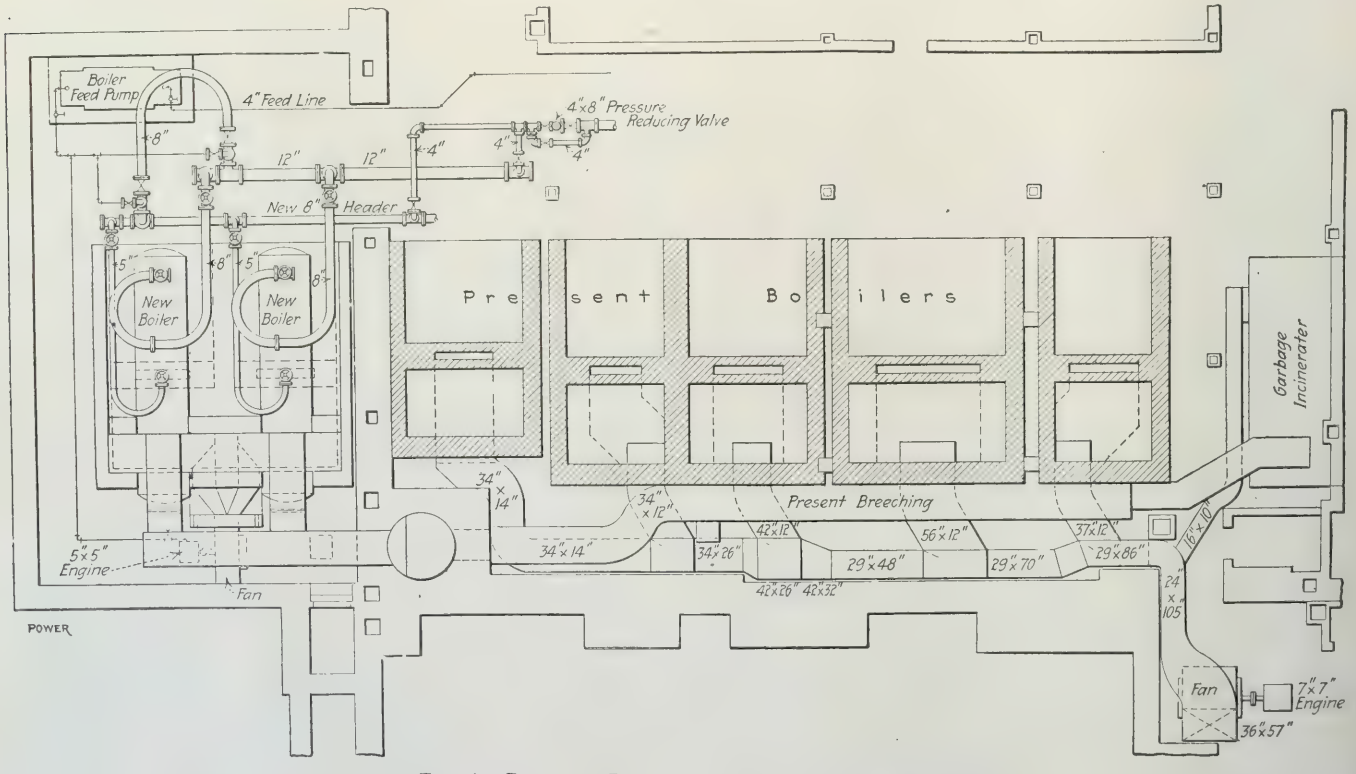


Fig. 1. Plan of Boiler Room at Ritz Carlton Hotel

To each boiler, 22 lb. of air per pound of coal was allowed for proper combustion. Assuming 4½ lb. of coal burned per horsepower-hour and the weight of 1 cu.ft. of air at 80 deg. F. being 0.073 lb., then

$$\frac{22 \times 4.5}{0.073 \times 60} = 22.6 \text{ cu.ft.}$$

of air is delivered per horsepower-minute.

Then multiplying the horsepower of a boiler by 22.6 gives the cubic feet of air to be delivered per minute to that boiler.

The velocity of air in the main duct is figured at 2500 ft. per min. and the discharge velocity into the ashpit is 1800 ft. per min., with a static pressure at the fan of 1½ in. of water.

A Garrett fan regulator controls the speed of the engine directly from the steam pressure in the boilers. This regulator operates like the ordinary damper regulator, the steam pressure being communicated to a diaphragm which actuates a pilot valve on a water cylinder. The piston in the water cylinder is connected by a chain to a balanced valve in the steam-supply main to the fan engine. As the steam pressure drops slightly, the speed of the fan is increased and correspondingly increases the draft which brings the steam pressure back to normal. If the steam pressure drops, the operation is reversed. This balanced valve in the steam supply has an attachment which can be so adjusted that it will not close off entirely but only slow the engine down to its minimum speed.

The flue from the garbage incinerator at the extreme end of the old boiler room discharges into the end of the main breeching as shown. Before the forced-draft system was installed, the main damper was in the breeching near the stack, beyond where the flue from the incinerator entered the breeching. As a result, when the main damper closed with the increase in steam pressure, it also closed the draft from the incinerator, causing the smoke to back up and producing objectionable odors which could be detected up through the house at times. To correct this trouble Canady combustion controllers, which would control the individual dampers in each flue and also the butterfly dampers in the blast connections under the grates, were installed. Then by discontinuing the use of the main damper and allowing the steam pressure to be controlled by the Canady system working in conjunction with the controllers on the fan engines, the gases from the incinerator were not retarded.

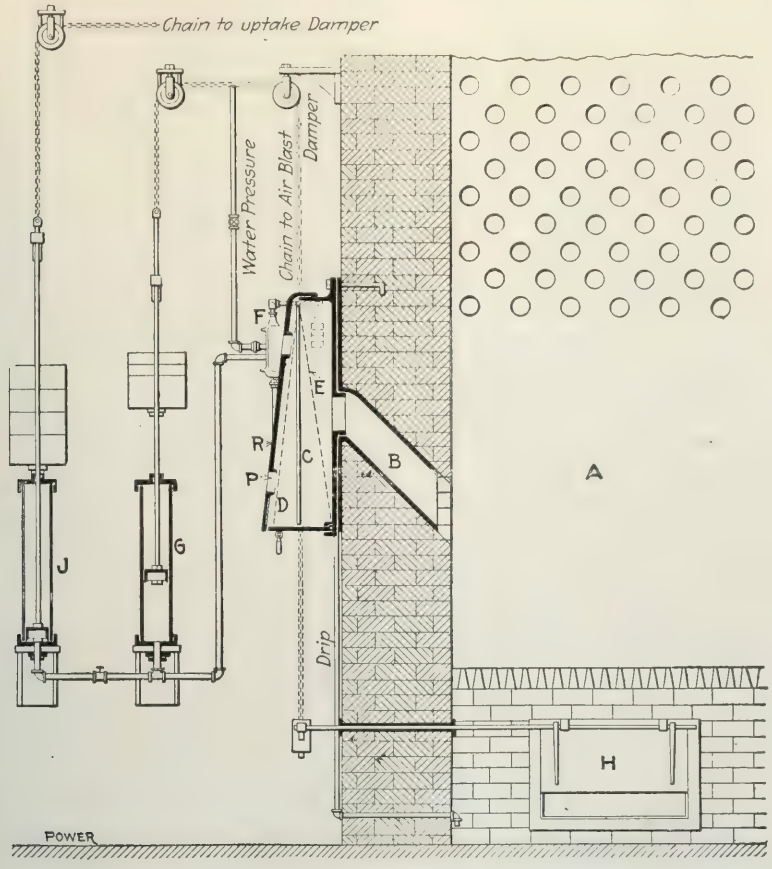


FIG. 2. ARRANGEMENT OF CANADY CONTROLLER

As the control on each boiler is separate and distinct from the others, each boiler will carry its proportionate part of the total load, as the pressure of air under the grates is so regulated that approximately the required amount of air will pass through the fuel bed.

The operation of the complete undergrate draft system is as follows: The normal speed of the fan is such that the quantity and pressure of the air delivered will be sufficient for operating the boilers at the average load, the pressure in the ashpit of each individual boiler being regulated as previously mentioned. In the steam line to the engine driving the fan, there is the balanced valve previously mentioned connected to a steam regulator which operates with change in the steam pressure. Should the steam pressure rise or fall, due to a decrease or increase in the boiler load, the fan will automatically slow down or speed up, the quantity of air delivered will be decreased or increased, and the combustion controllers will automatically readjust the positions of the dampers, admitting less or more air to pass through the fuel bed, corresponding to the necessary decrease or increase in the fuel consumption for the load on the boilers.

In its construction and operation the combustion controller is essentially as follows: Referring to Fig. 2, a cast-iron box *R*, approximately 18x16x4 in., containing a close fitting, swinging vane *C* of special heat-resisting material, is attached to the back of the boiler. The vane communicates on one side with the combustion chamber through a cast-iron pipe *B* set in the boiler wall, and on the other side with atmospheric pressure through ports *P* in the box cover. This vane being well balanced and of considerable area, is caused to swing either in or out, with the slightest variation of the suction in the combustion chamber and in doing so operates a small pilot valve *F* which in turn admits or releases

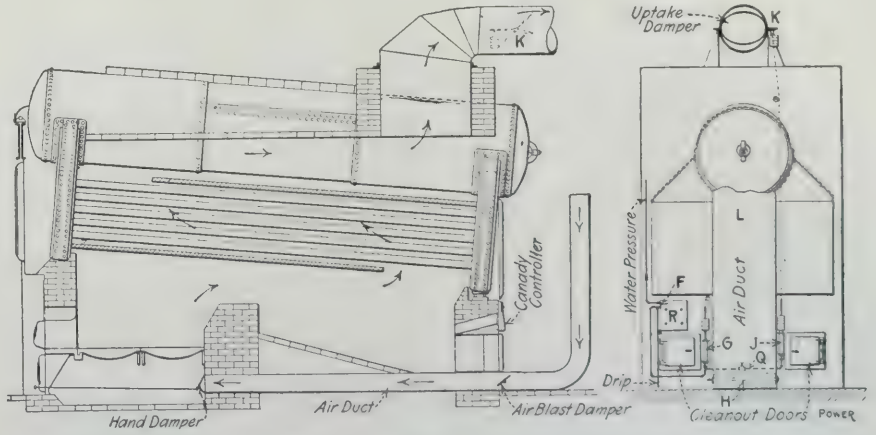


FIG. 3. SECTIONAL VIEW THROUGH SETTING, SHOWING DRAFT CONTROL

hydraulic pressure to the operating cylinders *J* and *G*, whose piston rods are connected by chains and levers to the dampers *H* and *K*, Fig. 3. The pilot valve *F* also acts as a check, locking the dampers in whatever position corresponds to the suction in the furnace for which the controller has been set.

The upward movement of the cylinder pistons is accomplished by hydraulic pressure, and the downward movement by weights on the piston rods. The weights on the cylinder controlling the uptake damper are considerably heavier than those on the cylinder controlling the air-blast damper, so that when water is admitted the piston in the latter accomplishes its full upward stroke, before the former starts to operate. On their downward movement the pistons act in the reverse order, due to the heavier weighting of the uptake-cylinder piston rod. As the draft necessary for removing the gases from the furnace is practically constant, the cylinder piston which operates the uptake or breeching damper, maintains the latter in the open position for which it is set, corresponding to the rating desired from the boiler, except at such times, which will be noted later, when the piston will close, or partially close the damper. Any increase in the thickness of the fuel bed, blow-holes, opening of fire-doors, or other furnace occurrances are taken care of by the piston operating the air-blast damper, which continually alters the position of the latter so as to maintain the desired suction in the furnace, and consequently a practically constant amount of air for a uniform rate of combustion.

The cylinder piston operating the uptake damper comes into play when the fan slows down, due to high steam pressure, the events taking place in the following order: When the steam exceeds the desired pressure, the fan slows down and immediately the air-blast damper goes wide open, but with the fan running at its slow speed there is not enough air to maintain the required vacuum in the furnace.

The piston operating the air-blast damper having gone to the top of its stroke and opened wide the damper, the uptake-damper piston starts to rise and gradually closes the damper, continuing in its upward movement until such position is reached, when the required vacuum is restored. The relative movements of the two cylinder pistons are reversed, so to speak, as when one rises it opens the air-blast damper, while when the second rises, it closes the uptake damper.

When the steam pressure falls and the engine speeds up, the cylinder pistons act in the reverse order, the one operating the uptake damper completing its down stroke before the one operating the air-blast damper starts on its down stroke, the latter coming to rest in the position corresponding to that which the uptake damper occupies with its operating piston at the bottom of its stroke.

In addition to the butterfly dampers in the supply ducts, hand dampers are also provided on the outlets in the bridge-walls, as shown in Fig. 3. These dampers are controlled by levers which extend out through the boiler fronts.

The addition to the power plant and the forced-draft system was designed by Messrs. Clark, MacMullen & Riley, consulting engineers, New York City.

CORRESPONDENCE

Heating the Shop

Some time ago the writer was sent out on a peculiar case in a factory doing good machine work. In the coldest winter weather the production seemed to fall of and the quality of the work was affected. The firm had an idea that the shop was not properly heated or ventilated. The building was about 200 ft. long by 50 ft. wide, set up on supports and without a basement. In summer it was an ideal place, but in winter the men complained of cold feet and parched throats. The machines were set across the shop with an aisle running the entire length in the middle and one on each side. In the roof over the center aisle were skylights to give ventilation. The heat was supplied by a fan pulling the air through a box coil made up of eight sections, having one valve on the supply and one on the return so that all of the sections or none must be used.

The heated air was then blown around the shop through pipes running on the side walls about a foot from the top. About every 8 ft. were branch pipes to distribute the air across the room, the idea being to send it across and then upward out through the skylights. This arrangement had worked fairly well for a few years though the men had complained of cold feet. The summer before the trouble commenced, the firm took out all the shafting and ran the machines by individual motors.

When the heating apparatus was started for the cold months it was found that the shop could not be heated with the skylights open, so they were kept shut and ventilation sacrificed. This resulted in general depression of the workmen and excessive drinking of water. They had tried to keep the side windows open, but it was so cold that they soon shut them. Steam was crowded into the coils at the expense of the engine. Then they tried turning down the branch air pipes, so that the hot air would blow almost upon the men, but soon gave that up.

When the writer had learned all the details, he decided that the overhead shafting and pulleys had acted as deflectors of the hot air as it came out of the pipes, so that the main current was broken up and the air distributed more generally. When the overhead work was taken out there was nothing to prevent the air from going across and out through the openings in the roof, leaving a cold strata of air near the floor.

There had never been a fresh air supply for the fan, the air being used over and over. The only fresh air coming in was the leakage around the windows. It was finally decided to carry the branch air pipes to the floors with deflectors to distribute along the floor, the air eventually rising and leaving through the roof. The firm decided against a fresh air supply for the fan, for if air could come in through the roof openings there would be no need of the additional expense. But when shown the effect of countercurrents and the advantages of thermostat control, it was decided to build a chamber around the steam coil and install mixing dampers automatically controlled by a thermostat. The steam coil was divided into four units under individual control. The result was satisfactory to all concerned and much better work was turned out during the winter months.

P. J. Downey.

Dorchester, Mass.

Leaky Thermostat Causes Trouble

A little trouble I had recently with a heating and ventilating system installed in a theater may be of interest.

Fresh air was taken from the outside and drawn through three heating coils by a fan. Under the coils was a bypass damper operated by a thermostat. When the temperature of the air reached 70 deg. the bypass damper opened and allowed the air to pass under the coils. When the temperature was less than 70 deg. the damper closed and all the air passed through the coils. During the cold weather the system worked all right, but in warmer weather, when cutting off the heat was necessary, the air compressor would not maintain normal pressure. Compressed air was used to operate all dampers except the bypass. On a very warm day the pressure would stay up but if it turned colder in the night the pressure fell off again, and the best we could get was 5 or 6 lb., where 15 lb. was necessary to operate the dampers.

A long search revealed that the thermostat had a leaky port. With the damper closed more air leaked out than the compressor could supply and maintain the pressure, but with the damper open the leaky port was closed so that the air pressure was normal. Reducing the leakage from the port cured the trouble.

Andrew A. Lafferty.

New Bedford, Mass.

In its last fiscal year, the Missouri Pacific carried 4,230,000 tons of coal, an increase of 36.5 per cent. over that of 1911.

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POWER

Reciprocating Engines for Battleships

It is interesting to note that the new dreadnought "New York," launched a few days ago at the Brooklyn Navy Yard, and which when placed in commission will be the most powerful battleship afloat, is to be propelled by reciprocating engines of 28,000 hp.

When the wave of popularity of the steam turbine for marine purposes was at its height a few years ago most of the foreign governments immediately adopted this type and our navy department was severely criticized for not taking similar action. It appears now, however, that the Navy Department was determined not to be influenced by popular opinion, but desired first to determine the status of the turbine for naval purposes from the standpoint of both maneuvering and economy.

Accordingly the three scout cruisers, "Birmingham," "Salem" and "Chester," were built, equipped with reciprocating engines, Curtis turbines and Parsons turbines, respectively. An exhaustive series of comparative steaming tests was made on these ships, the results of which were published in POWER at the time.

The first battleship to be equipped with turbines was the "North Dakota", the "Delaware," its sistership, having reciprocating engines. During the two years that these ships have been in commission we understand that the "North Dakota" has been decidedly the more expensive to maintain from the point of coal consumption, and has spent more time in the navy yard undergoing repairs than the "Delaware."

While it is true that the later ships, the "Florida" and "Utah," also have turbines, they were laid down and well under construction before comparative results could be obtained with the two earlier ships.

A battleship of this class is designed for a speed of about 21 knots, but when cruising, except during special speed trials, rarely exceeds 15 or 16 knots. In stationary practice the speed of a turbine is maintained constant and the steam supply is regulated to meet the load. In marine practice, however, the revolutions of the propeller, and consequently those of the turbine, decrease with a decrease in the speed of the ship. While the load curve of a turbine is somewhat flatter than that of a reciprocating engine, it is well known that the steam consumption increases with a decrease in the rotative speed. This accounts for the increased steam consumption at lower ship speeds.

The success of the turbine in certain fast Atlantic liners is due to their being run always at the speed for which they are designed.

International Expositions

It is hardly to be wondered at that the different countries have found it necessary finally to confer upon and establish agreements relating to the regulation of international expositions or exhibitions. Rather the wonder is that the abuse of them has not brought about some definite action earlier.

There have been various poorly organized attempts to do what a conference now in session at Berlin will probably accomplish, but lack of harmony or failure to coöperate on the part of all countries or interested concerns has hampered the work of earlier organizations.

A particular evil has been the too frequent occurrence of expositions, which has been a hardship to manufacturers, as it is the exception rather than the rule that the exhibitor comes through an exposition with his returns from it anywhere nearly equalling the expenses it has put upon him. The loss has to be charged to the selling expense, and ultimately the public or the purchasers of the commodities exhibited have to make up for it in the higher prices paid.

Another evil of the World's Fair is the debt which the city holding it contracts, always deluded by the idea that the city will be greatly advertised, that people will flock to it, and that its growth will be remarkably more rapid after the exposition is over. Municipal authorities, boards

of trade and enthusiastic citizens are always ready to invite an exposition, but the experience is always the same. The outcome is a large debt from which it takes a good many years to recover and generally several individuals are a few thousand dollars out with nothing to console them but the thought that they have spent their money in a benevolent cause if the community and outsiders, so far as they could reach the exhibition, have been educated by it.

Perhaps worst of all the abuses of the international exhibition as generally handled, is the bad management, sometimes so bad that it is criminal. A few unprincipled people may seek personal profit by graft in connection with the renting of spaces, making of awards, etc., and the awards, unfortunately, have come to mean so little that they are scarce worth striving for.

All these abuses will come within the province of the Exposition Congress in Berlin to act upon. It will be in session probably for several weeks and little will be known of its findings until the end. Enough is known so far, however, to be assured that, if the agreement is carried out, there will be no general expositions in the future except at intervals of three years, and the same country will not hold them more frequently than once in ten years. If the congress accomplishes nothing more than reducing the frequency of expositions it will have done a good work, for which manufacturers the world over will be grateful.

Cost of Central Station Current

A thoughtful analysis of the relative distribution of the expense of central-station current, as shown by the rates charged large and small consumers, is given on page 700. It shows an apparent injustice in the discrimination against the small consumer. Although individually his business is small, there are a great many of him and the aggregate load of all the small consumers is desirable both in quantity, duration and the time of day when it occurs. On this ground then it seems unfair that his rate should be such as to cover the fixed charges on the total capacity, while the large consumer actually obtains his power at less than cost (*pro rata* cost per kilowatt-hour sold).

Graphically this situation is depicted in the cartoon on the first page and helps to emphasize a condition that seems deserving of some discussion. It would be interesting to hear from others that may disagree with the author's conclusions, as to how they justify the existing distribution of expense for central-station current, or on what other basis more of arrangements regarding itinerary, pro, equitable rates might be fixed.

Write-Ups

For the benefit of the reader and advertiser alike we take this occasion to declare our position with regard to publicity matter. By the latter is meant anything emanating from manufacturers, whether news, descriptive matter, test data, or what not, issued for the purpose of keeping themselves before the public.

An editor must judge of the availability of matter from the standpoint of the subscriber. Is it the kind of matter that the readers pay for, the printing of which will satisfy them and cause them to continue their respect for and use of the paper?

Nobody has a more uptodate knowledge of power-plant problems than the makers of power-plant apparatus. Each is, or should be, a specialist in his own line. Access to this knowledge would be a godsend to an editor. Much of it might be printed to the advantage of the manufacturer. Most manufacturers are either so busy that they cannot stop to tell or write what they know, or are afraid it might be of some use to their competitors, and so hide their talents and those of their designers and engineers, under a "talk-and-be-bounced" order, instead of writing themselves and letting their people write themselves and the firm to the front as experts and specialists in their lines.

The publicity men of these firms are in position to get much material which we would be glad to have; material that is as welcome from them as from anybody else, and which we are quite as ready to pay them for as other contributors. Our experience, however, is that, as a rule, either the publicity man fails to sense the value of this class of material or his employer will not let him give it out, fearing that it will help a competitor or feeling that it does not sufficiently advertise him. His idea is that nothing does him any good but a puffy "write-up" filled with extravagant claims, and containing his street and number—the very thing that does him no good at all.

POWER does not give "write-ups." If an article appears to be interesting enough to our readers, we will publish it. If it comes from a publicity man and he is able to exploit his success in getting it used to his advantage with his client, that is his business. If the article is of real advertising benefit to the client, all the better, especially if he is one of our friends and patrons. But the consideration which leads to the acceptance of the article is not whether its publication will help the publicity man or the advertiser, but whether it will interest and benefit the reader.

The Modern Way

We are pleased to note that the American Society of Mechanical Engineers has joined the ranks of progressive organizations, and proposes in future to conduct its meeting on a strictly modern basis. In common with other societies, it has followed the old custom of appointing a local committee in the city where a convention was to be held to attend to the social features of the convention. This involves the soliciting of subscriptions from local members, which is bad, or from local firms, which is worse, in order to defray expenses of receptions, collations, dances or other entertainment.

The modern method is to have every participant in the social features of a convention pay his share of the expense, and we believe that both the visiting and local members will be glad of an opportunity of paying their own way instead of feeling under obligation to members who reside within the convention zone.

We are informed that this step was taken, after due consideration of all phases of the question, as being the only feasible method of handling the expense of the social feature, which, in spite of all economies possible, runs into several thousand dollars at each convention.

And while any change of this kind comes as more or less of a shock, we are quite sure that no member of an organization of this kind desires to have his entertainment paid for by another member, and that, as soon as members become accustomed to the new order, the change will be appreciated by all.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Notes on Water Works Pumps

In the writer's opinion, the design shown in Fig. 1 has many advantages from the operator's point of view. Should either of the steel valve sections *A* or *B* break, they can be removed conveniently without dismantling the entire pump and engine. All that is necessary is to remove the flange bolts and roll out the

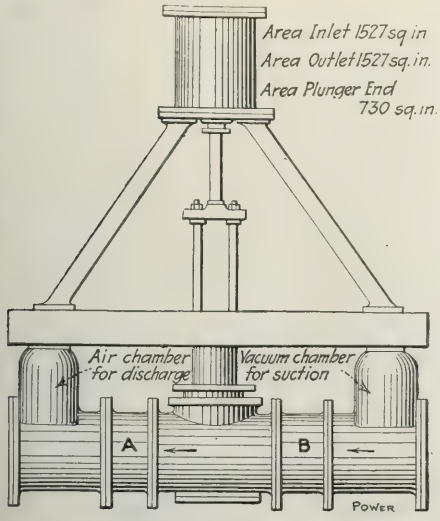


FIG. 1. A DESIGN HAVING MANY OPERATING ADVANTAGES

broken section when a new one can be placed in position.

Fig. 2 represents a vertical triple pump of an earlier design. This type having a duty of 100,000,000 ft.-lb. was considered the height of pump achievement when placed in operation about 1880. The duty of the pump shown in Fig. 1 is 180,000,000 foot-pounds.

The total inlet area of the suction-valve deck (Fig. 1) is 1527 sq.in., and the discharge area is the same. The cross-sectional area of the plunger is 730 sq.in., which makes the discharge and the inlet areas 50 per cent. greater than the plunger area, the displacement being 190 gal. for each revolution and a total of 570 gal. for the three plungers.

The argument has been advanced that the greatly increased duty of the later design is due to an increased steam pressure and the proportioning of cylinders that afford an increased number of expansions. The writer can see no great improvement in the steam engine of the two types; both are steam jacketed and have reheating coils in the receivers. An examination of the water ends shows an entirely different design and quite a difference in the proportioning of valve areas with reference to the plunger displacement.

The area of the plunger of Fig. 2 is 520 sq.in.; the total area of the inlet suction valves 250 sq.in., and the area of the discharge-valve opening is but 169 sq.in., thus making the area of the plunger 3.08 times that of the discharge-valve area and over twice that of the suction-valve area. Consequently the mechanical efficiency was very low, and the cost of repairs became unduly high. The breaking of suction-valve decks around the point *OO* was considered at that time due more to a mysterious providence than to bad design. Repairs of this nature were made by securing a 2-in. brass plate, of the same dimensions as the suction-valve deck, to the valve deck by tap bolts.

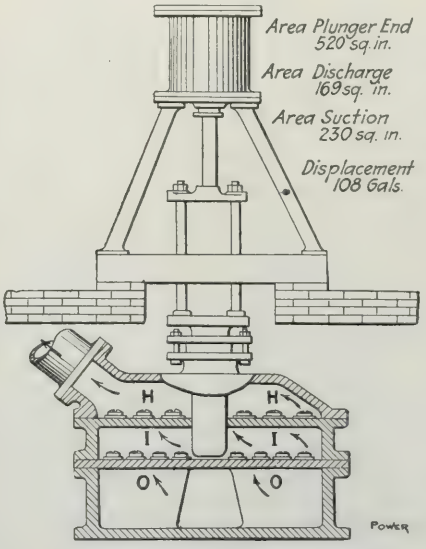


FIG. 2. AN EARLIER DESIGN OF VERTICAL PUMP

At high plunger speeds the pressure in the suction chamber, for the instant that the plunger started inward, would become much greater than that in the discharge line, due to the restricted passages and the inertia of the pump valves, the plunger tending to displace water more rapidly than it could escape. After more mature consideration, this seems to be the cause not only of the low duty, but the breaking of the suction-valve decks as explained.

The writer contends that the increased duty of the later design is due to a more efficient pump, and not to any great improvement in the engine design, other than a construction that affords an initial steam pressure 50 per cent. greater than that of 15 years ago. This extra spurt in the improvement of the displacement pump, came just in time to place it a little ahead in the race for high duty with the steam-driven turbine centrifugal pump.

Fig. 3 shows a horizontal type of pump that is of an earlier design than that shown in Fig. 2. Some designs included a tail-rod with an outside bearing, the intent of which was to support the weight of the plunger. This, however, seems to have been considered unnecessary by most American builders, many of whom considered the plunger simply as a watertight, cast-iron shell, and therefore to be regarded as a large float whose buoyancy is sufficient to keep it from bearing too heavily on the bottom of the packing box.

The writer has in mind one instance where a plunger became filled with water which caused it to run out of true and the bottom of the plunger became scored. The area of the plunger of the pump shown in Fig. 3, is 552 sq.in., the area of one-half of the outlets of the discharge-valve deck, is 379 sq.in., and of

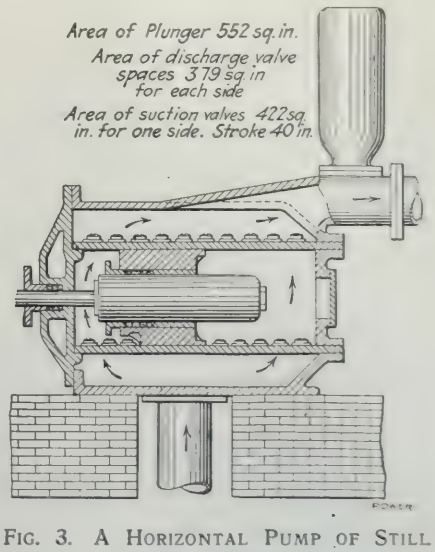

Area of Plunger 552 sq. in.
Area of discharge valve spaces 379 sq. in for each side
Area of suction valves 422 sq. in. for one side. Stroke 40 in.
POWER

FIG. 3. A HORIZONTAL PUMP OF STILL EARLIER DESIGN

one-half of the inlets of the suction-valve deck is 422 sq.in. It will be noticed that the proportions are more generous when compared with the plunger area than in the case of Fig. 2, consequently this pump is still in good condition and is well called "Old Reliable."

JOHN BURNS.

Columbus, Ohio.

Furnace Arch Design

Boiler-arch maintenance is a serious question, and little help is offered to users by the makers of boilers and stokers, who design the particular shape of arch used with their boilers. If trouble occurs, the boiler man will endeavor to shift the blame to the furnace maker, and even the mason contractor will be held responsible.

Some engineers have tried to overcome arch trouble by using the best grades of brick, but as firebrick makers are unable to produce a brick that will meet the requirements of present-day temperatures, this does not cure the evil. Good workmanship helps, as a poor brick arch well made is equal to an arch of good brick poorly laid.

Others have increased the spring of the arches, which helps to a degree, but the higher the spring the more the gases concentrate in the center of the boiler, leaving the heating surfaces near the wall cool, and the gases go to the stack above a normal temperature, causing a loss in efficiency and capacity.

By ventilating an arch it is possible to construct it with 1 in. of rise to each foot of boiler width, which makes a comparatively flat arch with a consequent more even distribution of the gases. In vertical types of boiler the gases enter the tubes at a lower elevation, utilizing greater heating surface and increasing the efficiency of the unit. A saving of 10 per cent. in fuel is being met with in plants now using ventilated arches, as well as an increase in the life of the arches as high as 400 per cent. over that of the ordinary type.

The question of how much the efficiency of the unit is impaired by ventilating the arch is sometimes brought up, but of the hundreds of samples of gas analyzed in a 13,000-hp. boiler plant equipped with ventilated arches, no loss was noticed, and samples taken with the air inlet open have shown a higher CO_2 analysis than when the opening was temporarily closed while taking samples.

I should be glad to see this subject discussed further.

E. E. Carr.

Chicago, Ill.

Near Accident to Boiler

Recently we nearly had a serious accident to one of our boilers which furnished steam to equipment for a mine.

The boiler is 54 in. by 16 ft. and of the horizontal return tubular type; the water column was connected as shown at *A*, trapping the water in the pipe. The pipe continued through the boiler as illustrated. This pipe was originally intended for a feed line, but at this time the boilers were fed through the blowoff connection. The pipe *BB* shows from whence the water was taken to the water column, so that should the top four rows of tubes be exposed to the fire, one could still get water when blowing down the column and thus feel at ease. A valve is in the steam line, but is not shown.

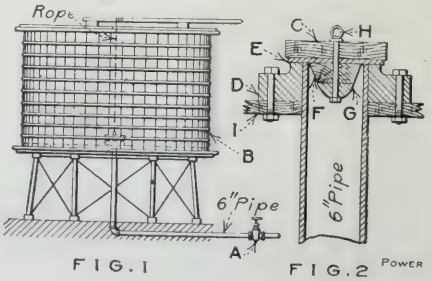
At the time of the accident, water was being forced into the boiler, when suddenly the rear tube sheet commenced to leak and the water disappeared from the glass; the fires were drawn immediately. By investigating we found the rear tube sheet sprung away from the beading of the tubes at least $^1/_{64}$ in. in places; the dotted line shows a somewhat exaggerated condition of the head at the front end. By actual measurement it is sprung about $^1/_8$ in. We have now changed the connection to that shown at the right of the engraving and the feed water now enters at *F*. As a result we are not having any trouble keeping track of the water.

William E. Piper.

Mineral Hill, Nev.

Saving a Tank of Water

The gate valve *A*, Fig. 1, on the pump discharge line to the water tank *B* had to be disassembled to renew some defective parts. As there was no valve in the line between this one and the tank and as the tank could not be emptied on account of the demand for water, the work of renewing the valve parts became a problem.



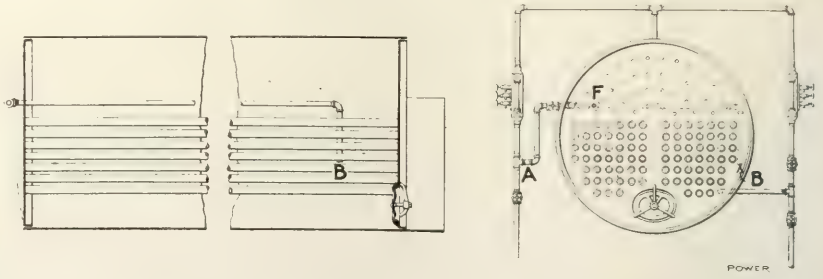
Tank and Details of Plug

To do the job the plug, Fig. 2, was made, in which *C* is a piece of 2-in. plank cut square and of sufficient area to cover the inverted flange *D* which is a permanent fixture in the tank. A gasket *E* of sheet rubber packing was fastened to the board so as to rest on the pipe. The wooden piece *F* acts as a guide and the babbitt end *G* as a weight to overcome the buoyancy of the wood. The eye-bolt *H* holds the plug together and serves as a place to attach the rope.

This plug is let down into the tank with a rope and pole. The water was drained from the pipe and the valve repaired without losing any water from the tank.

J. B. Proctor.

Crichton, Ala.

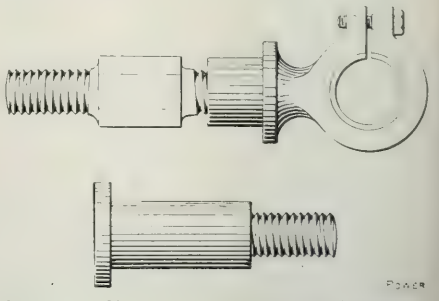


Showing Water Column Connections of Tubular Boiler

Repairs to Worn Bearings and Pins

We repair the bearings and pins on the steam- and exhaust-valve rods of our engines as follows:

When the brass gets worn we insert a piece of brass tubing and when the pins have become worn we turn them down and slip on a piece of steel tubing. The illustrations show the application of the brass and steel tubes. The tubes used for the brasses are split to allow for adjustment.



Showing Tube and Sleeve on Rod and Pin

W. M. Stockwell.

Dansville, N. Y.

Conversion of Calorie to B.t.u.

It is unfortunate that we should have two units in common use to express the unit quantity of heat, but as it is so we should know how they are related and be able to think one in terms of the other.

The calorie is defined as that amount of heat which is necessary to raise one gram of water at a comparatively low temperature 1 deg. C. The British ther-

mal unit is that amount of heat necessary to raise 1 lb. of water 1 deg. F. To get one in terms of the other we must know the values of the English and metric systems and of the Centigrade and Fahrenheit thermometers. To get B.t.u. in terms of calories, change pounds to grams and Fahrenheit to Centigrade.

1 *lb.* = 453.59 *grams*

1 *deg. F.* = $^5/_9$ *deg. C.*

Hence $^5/_9$ of 453.59 is the conversion factor. This gives 1 B.t.u. = 252 calories.

The number of British thermal units in a calorie is the reciprocal of the above of $^1/_{252}$, or 0.00397. This can be proved by changing directly gram to pounds and Centigrade degree to Fahrenheit, for

1 *gram* = 0.002204 *lb.*

1 *deg. C.* = $^9/_5$ *deg. F.*

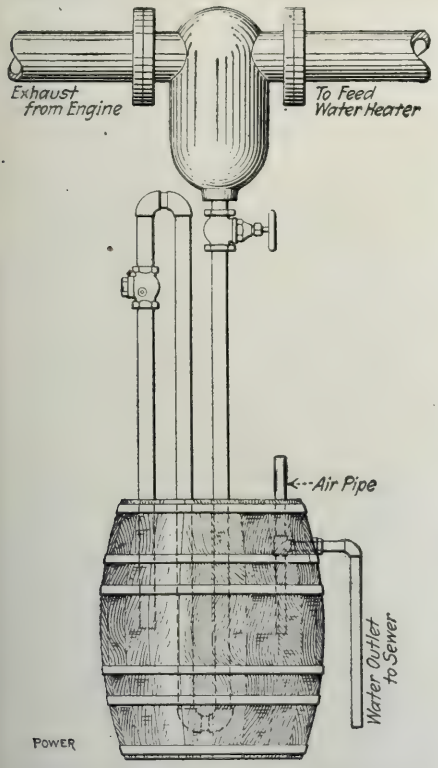
These multiplied give 0.00397.

A. L. Hodges.

Charleston, S. C.

Receiver for Oil Separator

Conditions being admirable for it, I have worked out a scheme for saving the cylinder oil from the exhaust steam in our plant. The main exhaust line is near the ceiling of the engine-room basement. All the exhaust lines from steam-using machinery run into it. Near the entrance



Piping of Receiver for Oil Separator

to the feed-water heater is an oil separator. The distance from the outlet or bottom drain of the separator to the basement floor is about 5 ft. The head was taken out of a new oil barrel, and the barrel placed directly under the separator. A water seal was then constructed as illustrated, extending from the bottom of the barrel up nearly to the level of the bottom of the separator and back nearly to the bottom of the barrel.

This gives a free drainage for the condensate and also condenses what steam would otherwise be forced into the open air. The pressure in the separator amounts to 1 lb. and more at times. A vertical check valve in the line to the seal guards against water being drawn into the exhaust lines in the heating season when we have occasion to pull a vacuum on the lines. A globe valve is inserted to provide for repairs and cleaning.

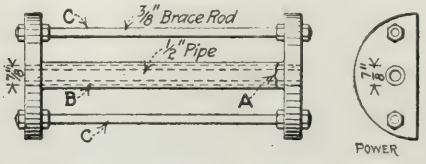
This brings us to the plan for separating the oil from the water. An inverted U-bend is passed through a hole near the top of the barrel with an air vent in its highest point to prevent siphoning. Thus the barrel stands full all the time without running over. The water is constantly draining to the sewer. The oil is dipped off, strained and used again mixed with fresh oil.

Edward T. Binns.

Philadelphia, Penn.

Quick Repair to Cracked Cutoff Valve

Recently the cutoff valve on one of our automatic engines cracked at *A*, the cause being a defective casting, which was less than $^1/_{32}$ in. thick on one side and about $^3/_{16}$ in. on the opposite side; it was, therefore, eccentric in appearance. As it was necessary to use the engine at



Showing Repair to Cracked Valve

once, I made the repair as illustrated in about an hour. The valve is giving good service so far.

A ½-in. pipe was slipped through the ⅞-in. hole *B* in the valve and the $^7/_{16}$-in. valve stem was inserted in the ½-in. pipe, making an excellent fit. The ⅜-in. brace rods *C* were put in and drawn up snug and the ends riveted over to prevent the nuts working off. The crack was then hardly visible.

Hugh L. Russell.

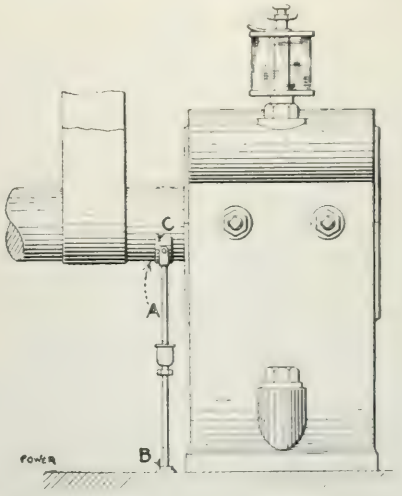
Keams Cañon, Ariz.

Regulating an Overspeeded Engine

On taking charge of a plant which contained one cross-compound Corliss engine 22 and 40 by 48-in., making 79 r.p.m., I found it overspeeding, so I reduced the counterweight until the speed was back to the normal 79 r.p.m. The governor belt was smeared with engine oil, so I told the assistant I would put on a new belt, but he informed me that the present belt had been on only three weeks.

When starting the engine we had to help the governor give an earlier cutoff by pushing on the reach-rod; otherwise excessive racing would occur, due to the governor belt slipping on account of its oily surface, which prevented the governor from gaining speed along with the flywheel. The governor speed lagged so behind that of the engine that there was danger of rupturing the flywheel.

The governor belt ran directly on the shaft and so near the main journal as to get oil soaked very quickly. I put on a new belt with a view to changing and cleaning both belts alternately with gasoline, but on running the new belt I found a clearance of 1 in. between the belt and the shell of the main bearing, as shown at *A*. The speed had dropped



Wiper to Absorb Oil before It Reaches Governor Belt

to 76 r.p.m. on account of the slipping being eliminated so the counterweight had to be set where it was in the first place. I then set a wiper, which consisted of a floor flange *B* for a base, and a pipe and brass oil cup with a drip pipe attached screwed into the floor flange. A rod was screwed into the oil cup and a leather top *C* riveted to the flattened end of the rod.

This wiper was set so that the tip would bear on the shaft just below the center. A guard post with a flange base screwed to the floor was used to prevent crowding the wiper in case the belt tried to take up its old position on the shaft.

John Powers.

New Bedford, Mass.

[Unless the wiper is well cared for, oil will get by it and the trouble will again appear. A metal disk with an overall diameter of about 6 in. greater than the shaft and secured to it near the belt would act as a positive baffle. The disk should be flanged at its base on one side only and fastened to the shaft with the flanged side toward the journal.—Editor.]

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

The Steam Trap as a Boiler Feeder

Relative to this subject it is surprising that Mr. Wickland should question the economy of the steam trap when compared with the ordinary direct-acting feed pump for delivering feedwater to boilers.

I have repeatedly "thrashed" this question out with men who are good engineers, but who do not seem to grasp the different principles upon which the two machines work. Possibly a short analysis of these principles, or conditions, may help to a better understanding of the subject which is one considerably discussed among engineers.

Assume a battery of boilers used for heating purposes principally, but the conditions are such that 100-lb. gage pressure must be maintained. The returns from the heating system come back at a temperature of, say, 257 deg. F. To get this temperature down to, say, 205 deg. F., which is about the highest temperature at which the ordinary feed pump will handle it, we must lower the temperature 52 deg., which is equivalent to over 5.4 per cent. of the fuel used to generate the steam for supplying the system, while the steam trap will return the water at the higher temperature, and the only loss is the small amount of steam that wastes through the air, or vent valve and which if the equipment is properly handled is insignificant, or less than 1 per cent. of the water evaporated.

The saving of heat possible by using a trap is, of course, the trap's greatest merit, but it also has other advantages which commend it. If the condensate is returned to the boilers by a pump it has to be forced in against a pressure of 100 lb. and the steam required to generate the power required is lost with each stroke of the pump while the trap merely takes sufficient steam to balance the pressure and the water drops back into the boiler by its own weight.
boiler by its own weight, thus making possible a big saving.

Advocates of the steam pump point to the fact that when the trap has delivered its charge of water the bowl or tank that has held the water is filled with steam of boiler pressure when it returns to the filling position, and claim this is equal to, or even in excess of the amount that would be used in the steam cylinder of the pump to effect the same results. But this is a great mistake. Admitting that the receiving bowl, or tank of the trap is filled with steam at boiler pressure when it lifts to the filling position and with the air valve open to the atmosphere, or condenser as the case may be, what really happens?

A little steam may escape through the waste pipe as the air valve opens, but instantly the entering water, which we assume to be but 81 deg. lower in temperature than the steam in the trap, begins to condense this steam transferring the heat in the steam to the water, all to be returned to the boiler at the next operation. True, there is some loss by radiation from the trap, but probably much less than from the pump cylinder.

The convincing facts are obtained by testing for the economy of each device with a small independent boiler and this has been done repeatedly. I regret that I have not these data now as they are very interesting, but to the best of my recollection the trap only took about $^{1}/_{30}$ of the steam to handle a given amount of water that was required by the pump, both devices being in good working condition.

Mr. Wickland's question concerning the efficiency being the same whether the temperature of the feed water is 40 or 250 deg., I cannot answer positively, having no data at so low a temperature. But supposing the water enters the trap at 40 deg., what takes place? Would not the steam that enters to equalize the pressure form a thin film of hot water on the surface of the water in the trap and thus act as a nonconductor to the main body of water underneath, thus preventing any further condensation while the charge is dropping into the boiler by its own weight?

As to the question of intermittent circulation, are we not confronted by a theory rather than by a fact? The real circulation in a steam boiler is caused by the evaporation and not by the entrance of the feed water. It would seem that half-minute impulses (which is about right for a well arranged trap plant) would increase rather than retard the circulation. This seems to be the case in practice.

If Mr. Wickland will procure circulars from makers of return or pumping traps and study them carefully, he will find much that will interest him and will probably agree with the above argument.

W. H. Odell.

Yonkers, N. Y.

Diagram from Engine Running Backward

In the issue of Aug. 13 appeared an article entitled "Engine Connected up Backward." The fact that an engine so connected would run and develop nearly full power, interested me so much that I made a short study of the probable diagram from such an engine. In the first place no engine would run with the steam flow reversed unless a balanced valve, such as a piston valve, were used in its

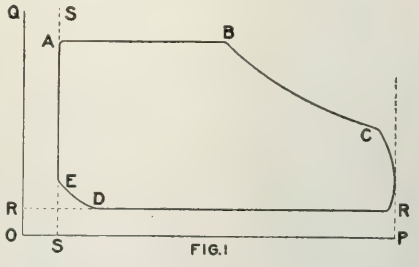


FIG. 1

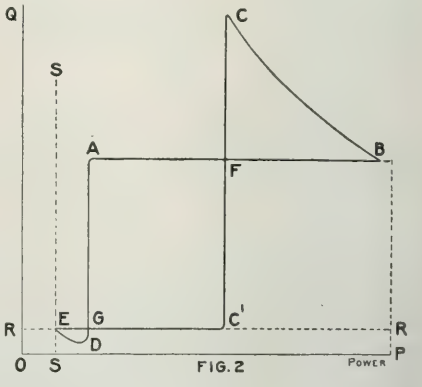


FIG. 2

construction. The direction of rotation would be reversed, but the valve motion relative to it would remain unchanged; hence the events would take place in a reverse direction, that is, admission on the valve-gear would become the exhaust period.

Fig. 1 represents an actual indicator diagram with late cutoff and small compression, and Fig. 2 the diagram resulting from reversed steam flow. In both figures *OP* represents the absolute vacuum line, *RR* the atmospheric line and *OQ* the clearance line.

With the position of the valve-gear remaining relatively unchanged the valve events would take place at the same position of the exhaust line no matter which the direction of rotation; hence the admission line *AB*, Fig. 1, has the same length and position on the stroke line as does the exhaust line *C'E* in Fig. 2; and

the exhaust line *CD*, Fig. 1, has the same length and position as the admission line *AB*, Fig. 2.

To construct the diagram for steam flow reversed from the actual diagram, it is best to start at the end of the exhaust period at *D*. The piston moving in the opposite direction makes this the beginning of the admission period, or *A* in Fig. 2. Steam then flows in for the rest of the stroke and a little of the return until the valve closes at *B*, corresponding to *C*, Fig. 1. From *B* to *C* there is adiabatic compression of the steam, corresponding to the expansion line *BC* of Fig. 1. Then the opening of the valve gives rise to the sudden exhaust shown by the line *CC'E* in Fig. 2. At *E* the valve again closes, the piston moves forward to *D* and gives the expansion curve *ED* which corresponds to the compression curve *DE* of Fig. 1. The valve then opens and the cycle is repeated.

It is now apparent that in Fig. 2, two negative loops *FBCF* and *GEDG* occur, which absorb work from the positive loop *GAFC'G*. In an engine having the governor acting on the cutoff the engine would operate under load when the steam flow is reversed because as the load increases the line *CFC'* is shifted farther and farther to the right, corresponding to a later cutoff in Fig. 1, and the positive area *AFC'GA* is increased while the negative area *FBCF* is decreased.

It can also be seen from Fig. 2 that while any engine would run provided its cutoff could take place later than three-eighths stroke, the exhaust would take place from a pressure considerably higher than boiler pressure, resulting in noisy operation and low efficiency.

From a practical standpoint, such a diagram as Fig. 2 is absolutely worthless, but the study of its evolution from the regular indicator diagram is at once interesting and instructive.

JAMES BAILEY.

Corning, N. Y.

Over Pressure Explosions

A boiler expert, recently lecturing before a body of college students on "Boiler Explosions," stated that the reason for so many explosions just before starting the engines in the morning or during the noon hour or shortly after closing down, was because a boiler accumulated pressure so rapidly when steam was not being drawn from it, that if the safety valve refused to work the bursting pressure was soon reached.

From time to time attention has been called by many writers to the fact, that explosions are likely to occur at the times mentioned, but in my opinion it is hardly likely that more than a small number of such explosions are due to overpressure.

The lecturer supports his argument with figures to show how rapid the rise in pressure would be in a boiler where the connections for the delivery of steam were all cut off. Starting with the heat absorbed per minute by a 100-hp. boiler, when operating at rated capacity, and the amount of water contained in the boiler, he figured that a boiler working at 85 lb. pressure would reach a pressure of 485 lb. in 14 min. after the steam connections were closed. Although not definitely so stated, the lecturer evidently referred to the horizontal return-tubular type in his calculations, and estimated that 5000 lb. of water would be contained in a 100-hp. boiler.

Using the lowest amount of heating surface allowed per horsepower for this type of boiler (10 sq.ft.) a 60-in. boiler with sixty-four 3½-in. tubes 16 ft. long, will just fill the requirements for 100-hp. capacity. Such a boiler filled with water up to 4 in. above the top tubes, would contain about 10,200 lb. of water.

As the time required to raise the pressure is directly proportional to the amount of water contained, this figure, if used in place of the 5000 lb. assumed by the lecturer, would raise the time interval to over 28 min. There must also be considered the evaporation of water to fill the steam space with the higher pressure steam which would probably bring the time up to 30 min., to say nothing of the increased losses due to radiation at the higher temperatures. In explosions of sound boilers due to overpressure, where witnesses have lived to testify, they always reported numerous leaks for a considerable time before the actual explosion, and the steam liberated by such means must add considerably to the time interval necessary to reach the bursting pressure. Again, calculations are based on the continuation of furnace conditions the same as at rated load, and as firemen usually check their fires at noon and bank them at night, it would seem difficult to estimate that boiler explosions from overpressure at such times were all due to a rapid rise in pressure on account of the stoppage of the flow of steam.

Overpressure explosions have often been produced before starting up, by a combination of defective safety valves and steam gage, or where the engineer depended solely on the registration of a defective steam gage and attempted to adjust the safety valve to meet its readings. However, in such cases, evidence has shown that the fireman had been endeavoring for a considerable time previous to the accident to obtain what he assumed was working pressure.

Most explosions at starting time have been the result of disturbance produced in the boilers by the opening of connections to engines or other steam-using apparatus, or by the connecting together of the different boilers. In most of these cases the boilers have been previously weakened by cracks, corrosion or other defects.

I do not wish to in any way belittle the dangers of overpressure, but an investigation of the average explosion which occurs just after the boiler has ceased to deliver steam, will usually disclose a severe weakening of the structure, such as a lap crack or corrosion as the prime cause of the accident. The final separation of the parts at such times is likely due to strains produced by the contraction of the parts caused by temperature changes in the furnace, the fire in which has been banked and the fire-doors opened to admit a blast of cool air and prevent the further formation of steam.

J. E. TERMAN.

Hartford, Conn.

Referring to the comment on my Cornell lecture, Mr. Terman admits by inference that my method of computation is correct, and merely states that my result would be different under differently assumed conditions, which, of course, is true.

I did not use in my formula actual measurements taken from some particular type of boiler, but assumed a hypothetical case involving 50,000 and 5000 because with these figures the arithmetical computation could be made mentally, and thus leave the student's mind free to grasp the principle involved. The time required to reach the bursting pressure after all openings are closed would be 14 min. under the conditions I assumed, and which would apply in the case of a boiler having very little water space in proportion to its rated horsepower. It would be 28 min. under the conditions assumed by Mr. Terman, and which would apply specifically to the horizontal tubular boiler from which he took his measurements; and, of course, it would be something else under still differently assumed conditions. But it seems to me that even 28 min. ought to be an appallingly short enough interval to satisfy anybody.

WILLIAM H. BOEHM.

New York City.

Shutting Down Plant During Summer

The article on the above subject in the Aug. 20 issue is not as good as it sounds. A certain government institution shuts down its power plant two months every summer and buys central-station current at a special rate of 2½c. per kw.-hr. During this time, the men are given a month's vacation at full pay, and the plant is overhauled and put in first-class condition. For the reason that the plant is three times as large as necessary, except the boilers, the overhead charges are excessive and the current in summer costs about 3½c. per kw.-hr. But in winter it costs about 1½c. per kw.-hr., so that

central-station current can be bought at a lower price than it can be made in summer. But this 2½c. is not all it costs; the overhead charges on the idle plant go on just the same, and must be added to the cost of central-station service. The wages of the employees are also constant, and must be added to the cost of outside current. The employees could be laid off without pay, but they would never come back, and as they are all good men it is to the employer's advantage to keep them, rather than hire new ones every year. This makes the outside current cost about 5c. per kw.-hr. with a consumption of about 1000-kw.-hr. per day.

This would not be so bad if the service was satisfactory, but it is not. When the isolated plant is running the voltage does not vary over 2 or 3 volts, while the central-station current varies 10 to 15 volts with the normal voltage at 110, and sometimes goes so low that the lamps are very dim, while at other times they are very much above the proper voltage, which makes lamp renewals more frequent. Also the unsteady current affects the operation of the elevators to some extent.

The conditions met with in this plant are not exactly as they would be in a private plant, but in any plant where the overhead and other charges must be added to the cost of central-station current, its price per kilowatt-hour has to be very low to compete with the isolated plant, even in summer when the exhaust goes to waste, as about all that is saved is the cost of coal and oil and possibly a little in wages.

J. G. Hawkins.

Hyattsville, Md.

Phosphorus for Flue Gas Analysis

The use of phosphorus for flue-gas analysis, given in the Oct. 15 issue, is new to me.

As I understand it, the sticks of phosphorus are placed in the absorption tube. Is any liquid put in the absorption tube with them? If not, how can one tell when the gas is all drawn back into the burette? Is there any heat generated by contact between the phosphorus and gas, i.e., enough to break the absorption tube? The article says it is easy to tell when a new charge of phosphorus is required. How? Is there anything better than cuprous chloride for absorbing carbon monoxide?

L. A. Farrell.

Yonkers, N. Y.

All the space in the absorption tube not occupied by the sticks of phosphorus must be occupied by water. There must be at least 100 c.c. of this free space to give sufficient room to introduce the gas to be tested. The preferable way seems to be to invert the absorption tube, fill it with water, and introduce the phosphorus sticks into it, displacing the water gradually. The sticks should be as long as possible, that is, run all the way from the top to the bottom of the pipette. After the pipette is about full, the cork is put in and the pipette turned right-side up, then, if necessary, the capillary tube and rubber connection are filled with water. The temperature rises only a few degrees when the oxygen in the gas and the phosphorus react. The products of combustion are soluble in water, and the sticks will grow slowly thinner. When the reaction becomes too slow, as shown by the slow evolution of white fumes, more sticks of phosphorus should be put in.

There seems to be no other absorbent for carbon monoxide except cuprous chloride. I personally prefer the acid solution as it is easier to make up, according to directions given in Hempel's "Gas Analysis." This is probably just a matter of personal preference.

Donald M. Liddell.

Elizabeth, N. J.

Slotted Connecting-rod Wedge Bolts

Regarding Mr. Cultra's trouble with broken connecting-rod wedge bolts as described in the Sept. 3 issue, I wish to offer a few suggestions. If his sketch, Fig. 1, represents conditions of the wedge-bolt failure, then he certainly went to much trouble in repairing it.

Instead of losing time slotting the inner end of the broken bolt to take a screwdriver as shown, all he needed to do was to tap the wedge down into its lowest position, and then use a small pipe wrench on the protruding end of the bolt. In trying to slot the end of the bolt in such a difficult position as he managed to do, one would be very likely to burr and rivet the end of the thread. This would make it still more difficult to screw the bolt out, in fact, it might easily "freeze" before making many turns, and defeat the purpose of a quick repair.

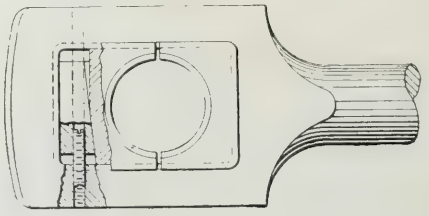


FIG. 1

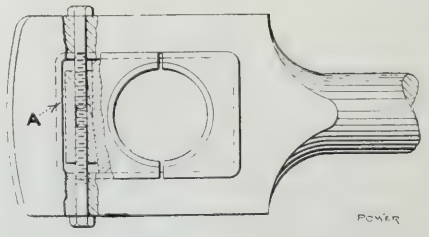


FIG. 2

Showing Wedges and Wedge-bolts of Connecting-rod

Mr. Cultra's suggestion to engine makers to slot the ends of their wedge bolts is good. However, a single through wedge bolt is not so likely to get abused in "keying up" as the double ones. With the latter, engineers have been known to forget to back off one before screwing up the other, and thus put an excessive stress upon the pair. This is especially true where an engineer is in the habit of pinching the brasses tight on the pin first, and then backing off for running clearance. A fracture is often started in this way that results in a broken bolt after a few days' operation of the engine.

I have frequently found broken wedge bolts to be caused by a poorly fitted wedge, as shown at *A*, Fig. 2. A wedge thus fitted is not backed up by the rod itself but must depend upon the bolts to resist the thrust of the brasses. The bolts soon become crystallized and break. Such fitting is generally due to carelessness in drilling the bolt-holes in the rod. The remedy is to place a liner of the required thickness behind the wedge. This will take the shearing stress off the bolts, which with proper care should last indefinitely.

J. A. Carruthers.

Hosmer, B. C.

Coincidental Names

Power's little squib in "Over the Spillway" of Oct. 15 regarding my brother, William G. Snow, tempts me to relate several amusing personal experiences.

Years ago I was sent to Albany, N. Y., to settle a controversy over a heating system. I met Mr. Winter, of the state architect's office, with whom the matter was successfully settled without any undue display of heat. Nevertheless, we parted very warm friends.

A short time ago I received a letter naturally addressed to Walter B. Snow, but somewhat curiously signed by Walter B. Frost. It was once my experience to have the Frosts as near neighbors, while the Hales lived around the corner. Not long since Mr. Winther occupied an office opposite mine at 170 Summer St., Boston.

Walter B. Snow.

Boston, Mass.

Submarine boat "F-1" established the world's record for depth during a six-hour submerged cruise in San Francisco Bay on Sept. 5. The boat attained a depth of 283 ft., which corresponds to a pressure of 122 lb. per sq.in. on the outside of the vessel. It maintained this depth for 10 min. at a speed of 6 knots, then rising to a depth of 19 ft. it proceeded at a speed of 8 knots.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Bituminous Coal

What are the leading physical characteristics which distinguish bituminous from anthracite coals?

F. J.

The bituminous coals are softer, become more broken in handling, lose more weight from air slacking, contain larger percentages of volatile matter, burn with longer flame and some varieties coke.

Brick for Ashpit Walls

Should ashpit walls be faced with firebrick?

S. L.

Good, hard burned brick is cheaper and better than firebrick for lining ashpit walls, as the latter frequently becomes moistened in wetting ashes before being drawn and firebrick rapidly crumbles if alternately heated and moistened.

Prevention of Dust in Indirect Heating

How can dust be prevented from being carried into buildings which are heated by indirect blower systems?

T. G.

The fresh air should be taken into a settling chamber and thence drawn through a number of screens of heavy cheese cloth; the screens should be spaced several feet apart and have from five to ten times the area of the supply ducts, so that the air may pass through the screens at low velocity. The settling chamber and screens should frequently be cleaned.

Water Power per Foot of Fall

What number of cubic feet of water flowing per second would be equivalent to a flow of 1,000,000 gal. in 24 hr.; and what would be the number of horsepower developed per foot of fall?

J. H. C.

One million gallons of water flowing in 24 hr. would be equivalent to a flow in cubic feet per second of

$$\frac{231{,}000{,}000}{1728 \times 24 \times 60 \times 60} = 1.547 \text{ cu.ft.}$$

of water per second; and the horsepower in the water or gross horsepower would be

$$\frac{1.547 \times 62.34 \text{ lb. per cu.ft.}}{550} = 0.17534$$

gross hp. per foot of fall.

Making a Tight Wheel Pit

Can ordinary portland cement concrete be relied upon as a water-tight construction for keeping freshet or tide water out of engine-flywheel pits?

L. B.

No, concrete wheel pits built in engine foundations are likely to become cracked from shrinkage of the material or vibrations from the engine, and may unexpectedly develop leaks from an external pressure of water. A safer plan is to line the wheel pit with a water-tight cast-iron or steel plate pan, anchored down to the engine foundation.

Steam Sizes of Anthracite Coal

What are the dimensions for commercial designation of different "steam sizes" of anthracite coal?

J. S.

Commercial standards for designation of the different sizes vary with the different coal companies, each having its own designation according to standards adopted for different sizes of meshes for revolving screens used for separation of coal sizes, but are approximately as follows: Pea size coal passes through ¾-in. mesh and over ½-in. mesh; buckwheat through ½-in. and over ¼-in. mesh; birdseye through $\frac{5}{16}$-in. and over ¼-in. mesh, rice through ⅜-in. and over $\frac{5}{16}$-in. mesh; culm through $\frac{5}{16}$-in. mesh.

Hydro-Electric Unit

How many kilowatts can be generated by a supply of 600 gal. of water at 150 lb. pressure used on a turbine waterwheel?

R. J. D.

Six hundred gallons of water would have a volume of 600 × 231 = 138,600 cu.in. and if supplied per minute at a pressure of 150 lb. per sq.in. would develop 138,600 × 150 = 20,790,000 in.-lb.

$$= \frac{20{,}790{,}000}{12} \text{ ft.lb of energy per min.}$$

Assuming that a turbine would deliver 75 per cent. of the energy developed by the water and that the efficiency of the electric generator, considering its size, would be 85 per cent., then as 1 kw. is equal to 44,240 ft.-lb. per min., the number of kilowatts generated would be

$$\frac{20{,}790{,}000 \times 0.75 \times 0.85}{12 \times 44{,}240} = 24.96$$

or in round figures, 25 kw.

Allowance for Piston Rod

In computing indicated horsepower of an engine, should not the reduction of piston area by the piston rod be taken into account?

J. N.

Yes; to be accurate it should always be taken into account, but the average size of piston rods for steam engines being only about ⅛ the diameters of their pistons, the error in computation of power developed on the crank side of a piston is only about 2.8 per cent. too much; or only about 1.4 per cent. too much of the total power developed by an engine for the same mean effective pressure on both sides of the piston. For most practical purposes, the error is so small that it may be neglected.

Testing Alignment of Engine Shaft

What is a quick method of determining whether the shaft of a horizontal engine is about square with the center line of the cylinder?

J. P.

If the engine shaft is first made level, a fairly accurate test can be made by comparing the path described by a point on the edge of the wheel with a strong fine line set in a vertical plane which is parallel with the center line of the cylinder, having the line pass over the engine shaft and stretched clear across the diameter of the wheel so as to nearly touch its edge. The line should be drawn taut and with each end made fast to a stationary object, should be set about level. It can be set in a vertical plane parallel with the cylinder center line by measuring equal distances with a rod or tram from plumb lines set over the center of the head end of the cylinder and the center of the piston rod close to the stuffing-box; or by plumbing down and measuring across to a line drawn on the floor parallel to the center line of the cylinder. Then measure the distance from the line to the nearest point on the flywheel rim, mark the point and turning the wheel over about 180 deg. until the same point again comes opposite the line, see whether its distance from the line measures the same as in the first instance. Any difference in the measurements will show which way the shaft should be moved for a distance in the length of the shaft equal to the wheel diameter.

Engineers' Study Course—II

Multiplying or dividing both the numerator and denominator of a fraction by the same number does not alter its value.

$$12 \div 3 = 4 \text{ or } \frac{12}{3} = 4$$

This says that 12 divided by 3 equals 4. Now let us multiply both the 12 and the 3 (both the numerator and the denominator) by 7 before making the division and we shall see that it makes no difference in the final result.

$$\begin{array}{rcl} 12 & \div & 3 = 4 \\ 7 & & 7 \\ \hline 84 & \div & 21 = 4 \end{array}$$

If we make the number we are going to divide as many times as big as we make the number we are going to divide it by, the proportion between them will be unaltered and the divisor will go into the dividend as many times as before.

It is equally true if we divide both the dividend and divisor (both the numerator and the denominator) by the same number

$$84 \div 21 = 4 \text{ or } \frac{84}{21} = 4$$

This says that 84 divided by 21 equals 4. Let us divide both numbers by 7 before making the division and we shall see that it makes no difference in the result.

$$\begin{array}{rcl} 7\,)\,84 & & 7\,)\,21 \\ 12 & \div & 3 = 4 \end{array}$$

In the above the result was a whole number, but the effect so far as altering this final value is concerned, is the same if we deal with a proper fraction. Let us try it with $^2/_5$

$$\frac{2 \times 3}{5 \times 3} = \frac{6}{15}$$

$$\frac{6 \div 3}{15 \div 3} = \frac{2}{5}$$

Here we multiply both the 2 and the 5 by 3 and get $\frac{6}{15}$, but if we divided a thing into 15 equal parts and took 6 of them, we should find that we had the same amount as though we divided it into 5 equal parts and took 2 of them. This is proved by the fact that when we divide the 6 and the 15 both by 3 the fraction comes back to $^2/_5$.

It is easier to work with small than with large numbers. The first thing to do usually is to reduce a fraction to its lowest terms. This is done by dividing its numerator and denominator by the same number so long as you can find a number that will go into both of them without a remainder.

EXAMPLE: Reduce $\frac{64}{256}$ to its lowest term.

We know that 8 will divide 64 evenly. An instant's inspection shows that it also goes into 256, so we have, dividing both the numerator and denominator by 8

This, the second lesson of the course, deals with cancellation and the reduction of fractions to their lowest terms. As we get a little more mathematics to work with, the examples can be made more practical and valuable. A little patience with the preliminary work will soon enable us to handle interesting engineering calculations, but if this initial work is slighted that which comes later will be hard to understand.

We shall commence a series of elementary lectures on Combustion in connection with the course in an early issue.

$$\frac{64}{256} = \frac{8}{32}$$

But 8 will go into both 8 and 32 also, so we can divide by 8 again.

$$\frac{64}{256} = \frac{8}{32} = \frac{1}{4}$$

That 64 is ¼ of 256 is easily proved, for

$$64 \times 4 = 256$$

This dividing of the numerator and denominator by the same quantity is called *cancellation* and it applies just as well when there are several factors in the numerator or denominator or both as where there is only one.

By the way, a *factor* is a quantity by which you multiply or divide.

A *term* is a quantity which is to be added or subtracted.

EXAMPLE:

$$\frac{35 \times 34 \times 12}{48 \times 21 \times 17} = \frac{14280}{17136} = \frac{5}{6}$$

Here are two series of multiplications to get the fraction with the single numerator and denominator and then a lot of division to get that fraction to its lowest terms. But see how easily it is done this way.

$$\frac{\overset{5}{\cancel{35}} \times \overset{2}{\cancel{34}} \times \overset{1}{\cancel{12}}}{\underset{4}{\cancel{48}} \times \underset{3}{\cancel{21}} \times \underset{2}{\cancel{17}}} = \frac{5}{6}$$

We can divide both 12 and 48 by 12. So we cross out the 12 and write 1 over it because 12 goes into it once, and cross out the 48 and write 4 under it because 12 goes into 48 four times.

You do not need to write the 1 because it is a factor, and multiplying or dividing by 1 does not change the value. If, however, there were no other numbers left in the numerator you would have to write 1 as the numerator of the answer as for example,

$$\frac{\overset{1}{\cancel{3}} \times \overset{1}{\cancel{6}}}{\underset{7}{\cancel{42}} \times \underset{5}{\cancel{15}}} = \frac{1}{35}$$

If there are nothing but 1's in the denominator you can forget them. The answer will be a whole number as

$$\frac{\overset{7}{\cancel{42}} \times \overset{5}{\cancel{15}}}{\underset{1}{\cancel{3}} \times \underset{1}{\cancel{6}}} = \frac{35}{1} = 35$$

To go on with our cancellation:

We can divide both 17 and 34 by 17, so we cross them both out and write 2 above the 34. But this 2 will go into the 4 which we have just written in the denominator so we cross them both out and write 2 under the 4 because 2 goes into it twice. We can divide both 35 and 21 by 7 so we cross them out and write 5 above the 35 because $35 \div 7 = 5$ and 3 below the 21 because that is the quotient of $21 \div 7$. All that is left above the line is a 5 and a 1. Below the line there are left a 2 and a 3, neither of which will divide 5 evenly, and no number that will divide 5 evenly will divide either of them.

We therefore multiply all that is left above the line together for a new numerator, and all that is left below the line together for a new denominator and find that the product of all the factors that we had equals $^5/_6$.

Remember that this cancellation applies only to *factors*, i.e., to numbers by which you are to multiply or divide. It does not apply to *terms*, or quantities which are to be added or subtracted. Let us try it.

EXAMPLE: A pump delivers 77 cu.in. per stroke and makes 69 strokes per minute. In a five-hour run 6930 cu.in. are used for miscellaneous purposes and the rest goes to the boilers. How many gallons of water do the boilers use per minute?

The pump would deliver

$77 \times 69 \times 60 \times 5$ *cu.in. in five hours*

Of this

$77 \times 69 \times 60 \times 5 - 6930$ *cu.in.*

go to the boilers in five hours. Five hours $= 5 \times 60$ minutes; so the number of cubic inches per minute which go to the boiler are

$$\frac{77 \times 69 \times 60 \times 5 - 6930}{5 \times 60}$$

One gallon is 231 cu.in.; so the number of gallons which the boilers get per minute will be

$$\frac{77 \times 69 \times 60 \times 5 - 6930}{5 \times 60 \times 231}$$

It would look as though we could cancel at once the 5's and 60's common to both numerator and denominator, divide the 6930 by the 231, and have an easy job of it. But the 6930 in the numerator is

a *term*. It is not a multiplier but is to be subtracted.

If you cancel with it there you must divide it by everything with which you divide anything else above the line.

Working it out the long way it will be seen that the right answer is

$$\begin{array}{rrl} & 77 & \\ & 69 & \\ \hline & 693 & \\ & 462 & \\ \hline & 5313 & \\ & 60 & \\ \hline 231 & 318780 & \\ 60 & 5 & \\ \hline 13860 & 1593900 & \\ 5 & 6930 & \\ \hline 69300 & 1586970 & (22\tfrac{9}{10}\text{ gal. per min.} \\ & 138600 & \\ \hline & 200970 & \\ & 138600 & \\ \hline & 62370 & \\ & 69300 & = \tfrac{9}{10} \end{array}$$

If we had tried to cancel the term it would be (multiplying the uncanceled values in the numerator together)

$$\frac{77\times69\times60\times5\times6930}{5\times60\times231}$$ (30)

$30 \times 69 \times 77 = 15{,}390$ *gal. per min.* which is ridiculous. But if we make of the term a separate fraction, writing it over the same denominator as the other, i.e., dividing it by everything by which we divide the other, we shall get

$$\frac{77\times69\times60\times5}{5\times60\times231}\ (\times 23) - \frac{6930}{5\times60\times231}\ (30) =$$

(21 / 3 over the first fraction; 2 over the second)

$$23 - \tfrac{1}{10} = 22\tfrac{9}{10}$$

which is right and easily and quickly done. Remember, then, when canceling to cancel only figures with the multiplication sign in front of them, and not any thing with a + or — sign. If a quantity has no sign, as when it is the first in a row like the 77 in the above example or as in the expression

$$\text{cir.} = 3.1416\ \text{D},$$

which means that the circumference is 3.1416 times the diameter, and the D, although it has no sign, is understood to be a multiplier.

When a sum or a difference comes in as a factor, that is, when we are to multiply something by the sum or difference of two numbers we can get rid of the plus or minus sign and make a multiplier at once of the sum or difference by performing the operation indicated. The area of a piston is 108 sq.in. Of this area, 12 sq.in. is taken out by the rod. The pressure is 77 lb. per sq.in. and the piston makes 1800 strokes of 18 in. each in 45 min. What horsepower does it develop?

The area upon which the pressure acts is

$$108 - 12\ \textit{sq.in.}$$

The pressure is 77 lb. per sq.in., or

$$77\ (108 - 12)\ \textit{lb.}$$

in all. You can leave out the multiplication sign, for 77 (108 — 12) means the same thing as $77 \times (108 - 12)$, but you must inclose the "terms" in parentheses, to show that it is their difference which is to be multiplied

$$108 - 12 = 96$$

$$77\ (108 - 12) = 77 \times 96 = 7392$$

But if it had been written without the parentheses it would be

$$77 \times 108 - 12$$

One is supposed to perform the operations in the order indicated, so that this expression would equal 12 less than the product of 77 and 108, which is

$$8316 - 12 = 8304$$

quite a different thing from what was intended.

Instead of the parentheses a mark over the quantities which are to be taken together may be used thus

$$77 \times \overline{108 - 12}$$

The total force then tending to produce motion is

$$77\ (108 - 12)$$

This force is exerted through 18 in., or $\frac{18}{12}$ ft. per stroke, and there are 1800 strokes so that the number of foot-pounds of work developed in the 45 min. is

$$77\ \ 108 - 12)\ \tfrac{18}{12} \times 1800\ \textit{ft.-lb.}$$

To find the number of foot-pounds developed in 1 min. we must divide this by 45, giving, since we may drop the denominator, 12 of the fraction $\frac{18}{12}$ in the numerator below the line.

$$\frac{77\ (108 - 12)\ 18 \times 1800}{33{,}000 \times 12 \times 45}$$

the division by 33,000 being because there are that number of foot-pounds per minute to a horsepower. Performing the subtraction in the parentheses this becomes

$$\frac{7\ \ 8\ \ 3\ \ 8}{77\times96\times18\times1800}$$ (8 over 1800)

$$\overline{33000\times12\times45}$$ (3 over 45)

(3500 over 100)

You could cancel further by dividing the 8 and 100 by 4, but 100 is easier to divide by in the final reduction.

Multiplying the uncanceled figures in the numerator together we have

$$7 \times 8 \times 3 \times 8 = 1344$$

and this divided by the 100, which is the only figure left in the denominator, is

$$\frac{1344}{100} = 13\tfrac{44}{100} = 13\tfrac{11}{25}\ hp.$$

Answers to Last Week's Questions

1. $\frac{21}{41}$.
2. $2\frac{3}{16}$.
3. $11\frac{9}{11}$.
4. $18\frac{14}{231}$.
5. $18\frac{68}{324}$.
6. $\frac{312}{847}$.
7. Divide 12 by 4. Divide the circumference by the diameter. Divide the quantity indicated by *a* by that indicated by *b*.
8. An integer.
9. The numerator.
10. The denominator.
11. $\frac{21}{5}$.
12. The numerator is 17x4 or 68 and the value is $\frac{68}{3} = 22\frac{2}{3}$.

Examples for Practice

(1) Reduce $\frac{96}{144}$ to its lowest terms.

(2) A cubic foot contains 1728 cu.in., a gallon 231 cu.in. What fraction (expressed in its lowest terms) is a gallon of a cubic foot?

(3) There were fired to a boiler 2456 lb. of coal, of which 307 lb. were taken out of the ashpit as ash and refuse. What fraction of the coal does this represent?

(4) The area of a piston is 64 sq.in. and of the rod 12 sq.in. The pressure in the cylinder is 40 lb. per sq.in. What fraction of a ton (2240 lb.) is the pressure tending to move the piston?

(5) What is the value of

$$\frac{16 \times 27 \times 6 \times 2}{9 \times 12 \times 48}$$

(6) What fraction of a foot is 4 inches? To make a pound of water at 32 deg. into a pound of dry saturated steam at 155 lb. absolute pressure requires 1194 B.t.u. Of these 861 are required to evaporate the water after it has been raised to the boiling point. What fraction, expressed in its lowest terms, if the total heat is employed for the evaporation?

(7) The stroke of an engine is 42 in. and the admission valve closes when the piston has moved one foot. At what fraction of its stroke is the engine cutting off?

(8) The coal supplied to a boiler contains 12,000 B.t.u. per lb. Of these, 8000 are absorbed by the boiler, 600 are lost by radiation and the rest go up the chimney. What fraction of the whole goes into the steam, what fraction is lost by radiation and what fraction goes up the stack?

(9) What fraction of an hour is 15 minutes?

(10) A circle is divided into 360 deg. When a crankpin has traveled through 120 deg., what fraction of a revolution has it made?

(11) An engine is supposed to make 72 r.p.m., but only makes 69. At what fraction of its rated speed is it running?

(12) To make a pound of steam out of the feed water which a boiler is getting requires 1155 heat units. Assuming (which is approximately true) that it takes a heat unit to heat a pound of water 1 deg., what fraction of the coal would be saved if the temperature of the water was raised 165 deg.?

(13) What fraction of a horsepower would it take to lift 275 lb. 90 ft. in a minute, friction neglected? (A horsepower is 33,000 ft.-lb. per min.)

The Ashcroft Thickness Gage

A convenient and accurate instrument for quickly measuring the thickness of sheet metal, sheet rubber, leather, paper, boxboard and any sheet material not exceeding 0.110 in. in thickness is a new product of the Ashcroft Manufacturing Co., 85 Liberty St., New York City.

As shown in the illustration the gage is held in the hand, and material to be

ASHCROFT THICKNESS GAGE

measured is inserted in the jaws which are opened by pressing the push button with the finger. When pressure on the push button is released the jaws automaticaly close and the thickness of the material is indicated on the dial.

The white-enameled dial is easy to read for the graduations stand out distinctly. The lines represent thousandths and the dots one-half thousandths. As the spaces between the dots and lines can be easily divided with the eye, the gage can be accurately read to one-quarter thousandths of an inch.

The gage has no sliding parts, the movement being mounted on steel pivots. Friction and wear are thus reduced to a minimum. It can be carried in the vest pocket if desired, and with proper care will last for years. As compared with the ordinary micrometer caliper, this gage has the advantage of being more easily read and the setting is automatic and independent of the individual's sensitiveness of touch.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Aging cheese by electricity, using a 0.2-ampere alternating current with 10,000 volts, is the scheme of a Rotterdammer (a somewhat suggestive name). As nothing is said of the deodorizer's capacity, we are forced to turn up our nose and declare the scheme "cheesy."

England must be enjoying "perfectly unexampled prosperity," says a Britisher; "have to go to the workhouse for men." This bally statement hardly proves the case, old top. My word! these beggars cawn't work, else they wouldn't be in the workhouse, d'ye see? It's the same at 'ome as in the States: There's men aplenty, but they lack skill, absolutely—and quite so!

A fly found a flea in a flue.
Says the fly: "Now what shall we do?"
Says the flea: "Let us fly."
Says the fly: "Let us flee."
They flew through a flaw in the flue.

The secretary of the New York Entomological Society has courteously invited us to hear a paper on "Experiences in Collecting Hydrophilidæ." N. Webster says this "hydro" is a gentleman clavicorn bug—not of hydro-electric development; therefore we must dodge this invitation.

A newspaper says that a workman fell into a boiling vat of pulp, and is in a "serious condition." Remarkable, how these newspapers are showing discernment.

Engineer lost his pocketbook containing ten real dollars while washing out a boiler. Months later, while again "laundering" the said boiler, he found the $10—bleached white, but the figures were there. No, it's not remarkable that an engineer should have ten whole dollars, but it's unusual to lose it that way.

The United States Treasury allows its clerks $1 a week to keep their trousers properly creased. When someone removes the jinx from the cost of high living—which is bagging at the knees at present—we can afford to increase our nattiness, decrease our sartorial negligence, and keep our unmentionables in crease.

Someone split Clutch's "block" with a disk plate last week. Clutch says he will have to readjust himself to new conditions before he will be able to make any more engagements.

Engine Speed Regulation in Mills

BY J. E. CROSLANDS

Regularity of speed in mills has been brought into considerable prominence by advocates of the electric drive; hence the following particulars of tests, relating to the governing of a vertical triple-expansion engine at a large English mill will be of interest. The tests were made with the object of ascertaining the regularity of speed which can be obtained with a three-crank engine of the most modern design, the engine being controlled by a center-weight type of governor.

The principal results of these tests show that the variation in speed, inclusive of momentary fluctuations, under normal working conditions, is about 3.4 per cent., and with apparatus of the most modern type, the total variation can be reduced to one-half of this amount. Also, in consequence of this improvement in regularity, a marked increase in production and quality is obtainable.

The engine, which had cylinders 20, 32 and 52 by 48 in., was designed to furnish about 1000 i.hp. when taking steam at 175 lb. gage, but at the time the tests were made the load varied between 650 and 760 i.hp. An exact record of the revolutions for 10-min. intervals was made for a period of about 3 hr. each day and during the whole of this period records of the speed variations were taken by a tachograph.

The engine was also indicated at frequent intervals to ascertain the load, and records were made of the machinery, which was being driven on different days in order that these should be available for comparison. The first test was made with the engine running under ordinary conditions and with the governor supplied by the makers. With average conditions the speed varied only 1.6 per cent. and under the worst conditions 2½ per cent. For the first hour of the test the steam pressure varied between 173 and 182 lb. per square inch. Later this was reduced to 140 lb. for a short period under which conditions the engine speed fell to such an extent that difficulties were experienced in spinning. In consequence, instructions were given for the pressure to be brought up again to normal.

The second test was made after the engine had been fitted with a new patented governing system known as the "Lumb system." This test was carried out, so far as possible, under the same conditions as in the former test, except that owing to the new governing system it was found that the speed of the engine was not visibly affected by a reduction in steam pressure. Therefore, the reduction of pressure was carried down to 126 lb. without any complaints from the workers in the mill.

Table 1 gives the comparative results of both tests, showing that the total variation in mean speed was equal to 1.9 revolutions when the engine was controlled by the governor supplied by the makers, and that this was reduced to 0.15 revolution when controlled by the Lumb system. These results are for the total period and include that part of the test in which the steam pressure was reduced.

TABLE 1. VARIATION IN MEAN SPEED OVER 10-MINUTE INTERVALS, INCLUDING PERIOD WITH STEAM PRESSURE REDUCED.

	Maker's governor	Lumb system
Average speed, r.p.m.	76.2	76.00
Variation in mean speed, r.p.m.	77 to 77.5	76.05 to 75.9
Equivalent variation, revolutions	1.9	0.15
Equivalent variation, per cent.	2.5	0.21

It is the custom in textile mills to reduce the pressure immediately before each stopping time, to avoid an abnormal increase in pressure after the engines stop and a loss of steam at the safety valves. The conditions in which reduced pressure is included are therefore in accordance with ordinary practice. The speed and variations, however, have been separately calculated for such a period of the test as was carried out under full steam pressure and the results are shown in Table 2.

TABLE 2. VARIATION IN MEAN SPEED OVER 10-MINUTE INTERVALS WITH REGULAR STEAM PRESSURE.

	Maker's governor	Lumb system
Average speed, r.p.m.	76.4	76.02
Variation in mean speed, r.p.m.	77 to 75.8	76.05 to 76
Equivalent variation, revolutions	1.2	0.05
Equivalent variation, per cent.	1.6	0.07

The tachograph records agree with the revolutions shown by the counter as regards the mean speed over 10-min. intervals, but they also show the momentary variations of speed, which are constantly occurring due to the change of load upon the engines. These changes in speed are an addition to the changes shown by Table 2 and are of great importance on account of the suddenness of the fluctuations.

The application of the new governor was accompanied by a marked improvement in the momentary changes of speed. Table 3 gives particulars of the total variation of speed, momentary and otherwise, as obtained by analysis of the tachograph records, which further show that under the Lumb system the momentary fluctuations are not only reduced in value, but in the length of their period.

TABLE 3. TOTAL VARIATION OF SPEED, INCLUDING MOMENTARY VARIATIONS UNDER ORDINARY WORKING CONDITIONS WITH REGULAR PRESSURE.

	Maker's governor	Lumb system
Average speed, r.p.m.	76.4	76.02
Equivalent variations, in revolutions	2.6	1.3
Total variation in speed, per cent.	3.4	1.7
Equivalent	77.7 to 75.1	75.97

From these details it is apparent that by the adoption of the improved system,

TABLE 4. TOTAL VARIATION IN SPEED, INCLUDING MOMENTARY VARIATIONS AND ALSO THE PERIOD WITH STEAM PRESSURE REDUCED.

	Maker's governor	Lumb system
Average speed, r.p.m.	76.2	76.00
Equivalent variation, in revolutions	3.4	1.4
Minimum speeds, r.p.m.	77.9 to 74.5	76.70 to 75.30
Total variation in speed, per cent.	4.5	1.8

the total variation in speed was reduced from 4.5 to 1.8 per cent.

It may be pointed out that there is an absence of reliable records of speed fluctuations in cotton spinning-mill engines and that many of the figures quoted in reference to uniformity of speed are far from accurate. Owing to the improvement in speed regularity, the production of a cotton-spinning mill can be increased and the quality of the yarn can be improved. With the original system of governing, the maximum speed was equal to 77.9 r.p.m., but the average speed was only equivalent to 76.2. With the improved system of governing, the engines could be run at 77.2 r.p.m. and the maximum speed would not at any time exceed that under the old conditions, namely, 77.9. This increase of speed is equivalent to a gain of no less than 42½ min. in a 55-hr. week.

Isolated Plant vs. Central Station

By G. H. McKelway

The items which enter into the real cost of the plant and decide whether it would be better to use central-station service are not only the ordinary ones of labor, fuel, water, lubricants, etc., but interest on the first cost of the plant, depreciation, cost of maintenance, cost of superintendance and the rental value of the room or building containing the plant.

For these latter charges much will depend on whether the plant is already built or is to be built. If an electric plant has already been installed and paid for, the substitution of central-station service for that already supplied by the isolated plant will not lessen this first cost which has already been paid for directly or indirectly, and if bonds have been issued for constructing the plant the interest on them will still have to be paid whether the private plant is abandoned or not. If the private plant is sold when the central station begins to supply power there will be some return, but it will not amount to anywhere near the first cost and probably not much more than the scrap value of a worn-out plant.

While there will be no depreciation if there is no private plant, yet by selling it and obtaining only a small fraction of its value the depreciation loss comes in a lump instead of being spread over a number of years. Again, if the room occupied by the plant cannot be utilized for other purposes, the plant need not be charged with rent.

Another point often adding to the paper cost of an isolated plant is the charging to it of part of the time of some man or men who would be needed in some other capacity around the building whether the power was generated or purchased. The mere changing of a man's title from fireman to engineer, or mechanic to switchboard operator is no reason for supposing that the company could get along without him if the power should be purchased from an outside source.

As a rule, when a manufacturing plant requires steam or hot water for use in its work, it is very hard for an outside plant to show a saving over an isolated plant, because then the comparison is not merely on the basis of supplying electric power but whether the central station can supply power at such a low rate that it will pay to purchase that power and at the same time operate boilers for heating. Where the central station can also supply steam for heating it may have a chance in some cases.

It is seldom profitable to purchase power when the isolated plant is run by water. The cost of maintenance and operation is so much less than that of a steam plant, unless an auxiliary steam plant should have to be installed, that all will depend upon the first cost of the private plant. If this has already been built the central station will have little chance, even if it as well is operated by water power.

After an isolated plant has been built, the fixed charges must be met anyway and should not be included. The comparison should be between the running expenses of the isolated plant and the fixed charges and running expenses of the central station, to which must be added some profit for the stockholders. Under such conditions it is not surprising that the private plant, in spite of usually greater cost per unit of capacity, smaller and therefore less efficient units, and relatively higher cost of fuel and wages, can make a better showing than its larger rival, which is also handicapped by having more or less distribution loss before the consumer's plant is reached.

When the private plant has to be built, the fixed charges must be included so that the cost will more nearly approach and perhaps even exceed that of the central station, depending, of course, on the cost of distribution and the profits of the corporation.

Fight to Reduce Coal Rates

On a charge that their transportation rates are excessive and prejudicial, five of the principal carriers of hard coal are defendants before the Pennsylvania State Railroad Commission.

The complainants are the three most influential business men's associations of Philadelphia, Harry E. Bellis, municipal railroad-rate investigator, and Prof. Ward W. Pierson, of the University of Pennsylvania, who has been making an investigation for Mayor Blankenburg.

The complainants petition the commission to issue an order recommending that the through rates or the proportion of through rates on hard coal, charged by the defendant companies from the Wyoming district to Philadelphia shall not exceed $1.066 a ton, and that the rate from the Lehigh Valley shall not be more than $0.844 per ton and from the Schuylkill district not more than $0.859 per ton.

NEW PUBLICATIONS

PRINCIPLES OF HEATING. By William G. Snow. David Williams Co., New York City. Cloth; 224 pages, 6x9 in.; 60 illustrations; 60 tables; indexed. Price, $2.

The book is a revised edition of one published in 1907 and has considerable acceptable data in the part devoted to the principles proper, as the chapters on combustion, boilers, transmission of heat, flow of steam and water in pipes, humidifying and cooling air, and pipe covering. The additional matter is mainly from papers describing different heating systems, most of them apparently written to exploit these systems. To make them appear advantageously, their sponsors have infringed some of the principles so carefully treated in the previous chapters. One is left with a sense of utter confusion with no idea as to the advantages of any one system that would best adapt it to any particular case. They are all "best."

Separate systems are described in detail which are essentially the same in action and principle except for a particular mechanical contrivance used to distinguish them in the trade. If they had been properly classified, some of the advantages omitted and a few disadvantages added, the book would be less confusing and possess more merit. The diagrams and description of the different steam systems are clear and the field well covered, and merit lies in having them all described between two covers.

In the numerous lists of advantages, misleading statements occur in regard to the increase in steam velocities by using vacuum, thereby reducing the pipe sizes and eliminating back pressure, while nothing is said of the reduction in capacity of the radiation due to the attendant lower temperatures, and pressures and corresponding increased volumes. The statements are all true taken separately, but taken relatively together would become impossible. If any considerable difference in pressure is created below atmosphere, the steam volumes increase very rapidly, which requires larger pipes for the same capacity while the temperature of the radiation is reduced. No engineer would claim smaller-sized piping for atmospheric pressure than for 20-lb. gage pressure. An inspection of a steam table will readily show the absurdity of claiming smaller sizes for steam below atmospheric pressure. Air removal is assumed and possible in all cases. It is true the mains may be reduced at any pressure by increasing the drop on the line.

The chapter on forced hot-water heating is extracts from papers, merely descriptive and with no illustrations or practical information. A paper by Prof. Woodbridge condemns its use for mill heating where power is not available and another by the same author favors its use for a central plant for the government buildings at Washington. The main idea being that hot water is inadvisable where intermittent heating is employed.

Another article by A. G. Hosmer compares the heating of a large storehouse with 10-lb. steam continuously and with 60-lb. steam from 6 a.m. to 6 p.m.; to accomplish the same results 35 per cent. more steam was required in the latter case. It would seem that the same difficulties attend the intermittent heating of large plants by steam as are complained of where hot water is used.

Considerable space is given to the use of a tank to store the water heated by exhaust steam during periods of peak load to be utilized later when the supply of exhaust steam is inadequate. This practice has been entirely abandoned and a few calculations will show that the size of tank to store available heat at proper temperatures for a hot-water system of size would be entirely impracticable.

In an article on central heating the author states that the hot-water systems require different amounts of surface along the line due to the drop in water temperature, and that with a vacuum-steam system all of the radiator temperatures will be the same regardless of the distance from the power house. It would be shameful to pay for the size of main necessary for a plant of any size and extent to accomplish this object.

In one article, A. G. Hosmer shows the advantage of utilizing exhaust steam for heating with even slight back pressure and in another doubts the practice of placing a heater of a hot-water system between the engine and condenser. The reviewer has actually installed several plants with the hot-water heater between turbines and the condenser and operated the heating under partial vacuum with results in economical operation on the combined heating and power that could be accomplished in no other way.

Several of the papers were written ten to twelve years ago and rapid progress has been made since that time. It is unfortunate that the author in his revision did not make a selection of later material and on this subject bring the book up to the present time.

"THE UNA"-FLOW STEAM ENGINE. By J. Stumpf. D. Van Nostrand Co., New York. 1912. Cloth; 229 pages; fully illustrated.

One of the most interesting European developments of the last few years is the "Una"-Flow engine, the invention of J. Stumpf. Reports of its economical performance have been brought to this country from time to time and one of our prominent engine-building firms has announced that an American engine of this type is now ready for the market. Until recently only in scattered foreign periodicals has it been possible for the American engineer to obtain particulars of the construction and economy of these engines but the rapid adoption of the type by European builders has convinced Mr. Stumpf that a book was needed on this subject and the present volume is the result.

Mr. Stumpf has here collected a series of pictures, drawings, details and tests covering the history, theory, construction and operation of his "uni-directional flow" engine and its application to mill and ordinary service engines, rolling mill and hoisting work, locomotives, marine engines, compressors, pumps, and the semi-portable type known in Europe as the locomobile.

He claims, and the tests presented bear out his contention, an economy for his single-cylinder construction equal or even better than that obtained by compound or triple-expansion engines of the usual type. The extent of their use in Europe is shown by the statement that in June, 1911, 600,000 hp. were in actual operation.

Chapters 1 to 5, about 20 per cent. of the book, are devoted to the general theory of the construction of the engine, and the influence of the condenser, steam jackets, the prevention of leakage and the loss by clearance surfaces.

Chapter 9 treats of the influence of the clearance volume on the steam consumption. The remainder of the book relates to the various types in service with their details.

The book will be of value not only to the designer and student but also to the engineer.

In a 120-page book entitled "DeLaval Steam Turbines, Multistage Type," the DeLaval Steam Turbine Co., Trenton, N. J., has presented more than an ordinary trade catalog. Fully a third of it is devoted to a discussion of the "speed-

compromise" problem; that is, finding the best means of reconciling the high speed natural to steam turbines with the low or moderate speeds of driven machinery. The relative advantages of the several fundamental types of turbines, as affecting this and other matters, are discussed under the following chapter heads:

The Field of the Single-stage Turbine; The Necessity for Multi-stage Construction for Large Turbines; Advantages of Speed Reduction by Gears; Considerations Affecting Choice of Type of Turbines; Relation Between Rotative Speed and Number of Stages; Effects of Different Methods of Compounding; Choice of Type of Turbine; Benefit to the Driven Machine from Being Able to Obtain the Proper Speed; Alternator Speeds; Direct-Current Generator Speeds; Centrifugal Pump Speeds; Fan and Blower Speeds, and Speeds for Rope and Belt Drives.

The remainder of the book is occupied by a detailed description of the design and construction of the DeLaval multistage or multicellular turbine, which is built in capacities of 500 hp. and up, and of the DeLaval speed-reduction gear, through which this turbine is applied to driving standard-speed direct-current generators, centrifugal pumps and blowers, and for rope and belt drive. Novel in the construction of the DeLaval multicellular turbine is the use of solid steel rings shrunk over the stationary guide vanes of the diaphragms, entirely encircling the wheels. These rings provide an impenetrable steel armor, which will effectually prevent injury to surrounding objects or persons if the turbine should be overspeeded, as through lodgement of objects under the governor valve. A final chapter argues that because of their lower first cost, steam-turbine-driven centrifugal pumps can compete successfully with the most efficient triple-expansion pumping engines wherever coal does not cost more than $6 to $8 per ton. The book also contains a Mollier or total heat-entropy chart, accompanied by a scale by using which may be read off directly the energy available from the expansion of steam between given limits, as heat units, the resulting velocity of the steam in feet per second, the duty in foot-pounds per thousand pounds of steam, and the number of pounds of steam consumed per horsepower-hour.

BOOKS RECEIVED

THE MODERN GASOLINE AUTOMOBILE. By Victor W. Page. The Norman W. Henley Publishing Co., New York. Cloth; 700 pages, 6x9 in.; 500 illustrations. Price, $2.50.

STEAM BOILERS. By W. Inchley. Longmans, Green & Co., New York. Cloth; 412 pages, 4¼x7½ in.; 141 illustrations; plates; tables. Price, $2.40.

THE FREEZING POINT, BOILING POINT AND CONDUCTIVITY METHODS. By Harry C. Jones. The Chemical Publishing Co., Easton, Penn. Cloth; 75 pages, 5x7¾ in.; illustrated. Price, $1.

SOCIETY NOTES

A. R. Maujer, Western editor of POWER, delivered an illustrated lecture on "Western Power Plants" before the Robert Fulton Association, Illinois No. 28, N. A. S. E., on Nov. 2.

O. Monnett, chief smoke inspector of Chicago, presented a paper, "New Methods of Approaching the Smoke Problem," at a regular meeting of the Western Society of Engineers on Nov. 4. The paper outlined the policy employed by the smoke-inspection department and described the newly developed method of making smoke observations with the Ringlemann charts as outlined in POWER of Aug. 6.

OBITUARY

Vaughan Pendred, for many years editor of *The Engineer*, of London, Eng., died recently at his home in Streatham. He was born in Ireland in 1836. His father was a classical scholar of note and his mother a highly educated woman. From his parents and from two resident governesses he obtained his early education, and from his love of books and reading he acquired a mastery of many subjects. His parents were poor and his success in engineering was the result of pertinacity and hard study.

Vaughan Pendred was the inventor of an electric arc lamp and of other devices, which, however, he never sold or exploited, with the exception of the Nozzle boiler, invented when he was still a young man. This boiler was of the vertical type with a rectangular firebox. Water tubes crossed the firebox in both directions with a slight inclination. Across the higher ends extended a trough, with partitions, directed upward; across the lower ends was a similar trough pointing downward. The troughs, or nozzles, assisted circulation and good evaporative results were attained.

In 1862, Mr. Pendred first came to England to become an engineer. He obtained a position with a small firm in Covan, Staffordshire, which made traction engines and agricultural machinery. He had early begun to write letters and articles for the technical magazines and shortly after coming to England he became editor of the *Mechanic's Magazine*. In 1865 he became editor of *The Engineer* at 29 years of age. From 1865 to 1905 he devoted his entire attention to this work. To him credit is given for bringing out illustrations of current practice in machine design. Previous to his editorship engineering journals had depended upon what they could glean from patent-office drawings, and it was his influence which persuaded machinery and engine builders to permit the reproduction of their working drawings.

Mr. Pendred was a member of the Iron and Steel Institute, the Institution of Mechanical Engineers, the Society of Engineers, and an honorary member of the Cleveland Institute of Engineers. He contributed few papers to these or other societies; all his time and energies were devoted to *The Engineer*.

He is survived by a widow and three sons, one of whom, L. St. L. Pendred, is now editor of *The Engineer*.

PERSONAL

P. A. Hoffman, New York, former sales manager of the B. F. Sturtevant Co., is now with Evans, Almirall & Co., contracting engineers, New York City, in the engineering sales department.

F. L. Johnson, formerly connected with the construction department of the Brooklyn Edison Co., has accepted charge of the power plant of the Trenton & Mercer County Traction Corporation, Trenton, N. J.

Herbert C. Goodwin, engineer of the power station of the Willimantic (Conn.) Gas & Electric Light Co., has recently become assistant superintendent of the New Britain Gas Co. He was presented a sum of money from the Willimantic company and a handsome memorial from his fellow employees.

Robert Renton Hind, engineer of the Hawi Mill Co., Hawaii, T. H., which operates 35 miles of private railway on the islands, visited POWER while in New York recently. Mr. Hind says the islands offer good opportunities to competent operating engineers. He is a member of the American Society of Mechanical Engineers.

Capt. Asher Carter Baker, U. S. N. retired, will be the director of exhibits of the Panama-Pacific International Exposition. Capt. Baker has had most unusual experience and important service in general exposition affairs, having been connected with the Chicago Exposition in 1893, the Paris Exposition in 1900, and the St. Louis Exposition in 1904. He was vice-president of the class jury, vice-president of the group jury and a member of the superior jury at the Paris Exposition. While he gave more particular attention to the transportation exhibits and still more attention to modern architecture and development, yet he has had a great deal of experience in all of the departments of the exhibits division and has become an acknowledged expert in classification of the world's products and occupations of man.

It is said that the farmers of the United States have on an average $6.25 per acre invested in animal power, and it costs at least $4.38 a year per acre for maintenance. Talk about maintaining a power plant, gasoline engines and gas tractors can beat that!—*The Gas Engine.*

Moments with the Ad. Editor

Along the lines of what we have been saying on this page, here's an item from *Printer's Ink* that will interest you:—

Denver Merchants to Stop Misleading Advertising

——

The Denver Retail Merchants' Association has appointed a committee to take steps toward the passage of a state law which will regulate to an appreciable extent fire, bankrupt and other misleading advertisements which have been appearing in local newspapers. The new law will be made to include local advertisers as well as itinerant vendors.

"But," you ask, "what has this to do with power plants?"

The answer to that is that advertising today is a national—even an international—force. It has no limits as to trade or profession. Its principles are the same in all lines and those principles stand for truth and honesty.

If the advertiser is so small that he believes he can reap a profit from dishonest advertising, then the law is going to step in and stop him.

But there are few advertisers these days who need *legal* restraint—

And especially is this true in your field.

It isn't because it's against the law in so many states that they don't make untrue claims and statements—

It's primarily because they are good business men and realize that in advertising honesty is not only the *best* policy, but the *only* possible policy that can win.

And when you get business men on that basis you've got a protection and an assurance of the quality-you-want that is better than any other forms of "guarantees" can possibly be.

There are a lot of interesting ads in this issue—have you looked them over?

POWER

Vol. 36 NEW YORK, NOVEMBER 19, 1912 No. 21

Concentration

WHEN you sit down to a game of cards, checkers or some similar pastime, you do not permit your mind to wander to other things, at least not if you expect to win. On the contrary, your sole purpose for the time being, is to make as good a score as possible. Consequently you give the contest your undivided attention. If you lose, it is because luck was against you, or your opponent played a superior game. Certainly you never lost because your "mind was not on the race."

When it comes to prosecuting your calling —that which is the means of procuring your bread and butter—do you concentrate with equivalent intensity?

You would not attempt to enter into a game until you had digested the rules pretty thoroughly, for you realize that your chances of winning depend largely on your knowledge of them and your ability to apply them effectively at the right time.

Similarly the mastery of the principles of a science such as steam engineering, is the first essential toward playing a successful part in its application. The question of how to succeed, although ages old, is in truth simple and easily answered. In its shortest form the answer is the single word, "concentrate." We use the term in its broadest sense, to include both physical and mental activity. Continuously apply the same intensity of thought and effort to your daily work that you do to your favorite pastime and you will succeed.

Close attention to the study of a subject leads to complete knowledge thereof, while a roving mind, at best, will be but half informed. A man may be a slow thinker, but if he centers his entire attention on his task, excluding all else and resolutely persisting with the one idea in mind, he will accomplish greater results than a man of much more ability, but whose thoughts are distracted by new ideas and plans for future work or play, allowing him to stumble from one error into another.

So much then for mental concentration.

The other important factor in the battle to achieve is concentration of effort. "Bunch your hits," as they say at the ball grounds. Do the work that you now have to do, both routine and special, the best you know how, and make sure that you know the best way. Apply all the rules of the game and be thorough.

It is widely and wisely quoted that "The reason why the man who minds his own business succeeds is because he has so little competition." Do not make the mistake of trying to play every position on the diamond, for you never could "get away with it." Concentrate on your own. That little position which you hold down is probably plenty big enough for you at present. If it is not, it is only a question of time before you will progress to a bigger one.

In the meantime it does no harm to wait and work!

Practical Flue Gas Analysis

By Harry A. Cox

A knowledge of chemistry is not necessary to operate the Orsat. Read the following article and see how easy it is to interpret results and reduce the amount of coal burned.

Some writers claim that to operate an Orsat requires a knowledge of chemistry. It naturally follows that, since the average engineer's knowledge in this branch of science is limited, he is loath to interest himself in any instrument with these requirements. However, such is not the case for an ordinary fireman will easily learn how to operate an improved gas-analysis instrument, such as is shown in Fig. 1, and interpret the results obtained. The instrument can be bought all set up and ready for the chemicals which are furnished with the outfit. The instructions are so plain that no man need have any difficulty.

The three gases entering into the combustion of coal that need to be taken into account are carbon dioxide (CO_2), oxygen (O) and carbon monoxide (CO). It is desirable to keep the percentage of carbon dioxide as high as possible without obtaining carbon monoxide.

The accompanying table for which the author is indebted to Joseph B. Hays, shows the saving to be made by proper furnace operation.

Per cent. CO_2	Air excess per cent.	Fuel loss per cent.	Preventable loss per cent.
15	38.0	12.0	1.0
14	47.8	13.0	2.0
13	59.2	14.0	3.0
12	72.5	15.0	4.0
11	88.1	16.0	6.0
10	107.0	18.0	8.0
9	130.0	20.0	11.0
8	158.7	23.0	14.0
7	195.7	26.0	18.0
6	245.0	30.0	24.0
5	314.0	36.0	33.0
4	417.0	45.0	48.0
3	590.0	60.0	78.0
2	935.0	90.0	....

The data refers to pure carbon, so for anthracite the results given will be nearly correct, but a little high for bituminous coal.

The factors which make a furnace efficient are the proper distribution of air through the fuel bed, methods of firing which will keep the percentage of carbon dioxide comparatively constant, absence of air leakage between the furnace and damper, proper adjustment of the draft and lastly the human factor.

Take, for example, a boiler fired with No. 1 buckwheat coal and test for the distribution of air. The sampling pipe is placed at a point where the flue gases leave the heating surface on its way to the damper and as near the center of the current of flue gases as possible. Connect the inspirator to the Orsat and sampling pipe and with a freshly covered clean fire on the grate draw a sample of gas and analyze it for carbon dioxide. As soon as the fire has become incandescent draw another sample and analyze it. This percentage will be larger than the first. Then when the fire has burned through and holes show on the grate, analyze a third sample. This result will be smaller than the other two. It will be noticed then that a green fire does not need as much air as an incandescent bed of fuel, and that a fire full of holes is not conducive to a high percentage of carbon dioxide, because the resistance of the fuel bed is decreased and too much air is admitted to the furnace.

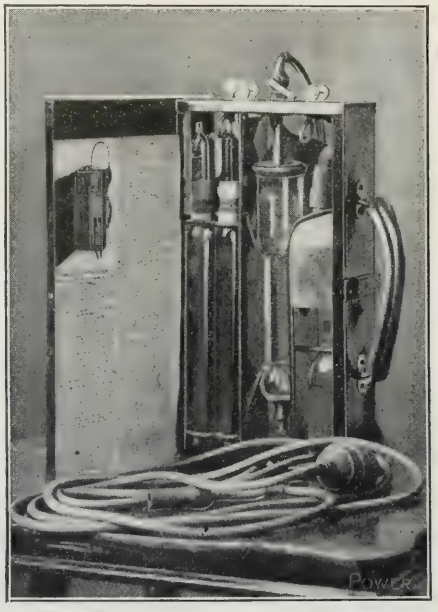


FIG. 1. APPARATUS FOR FLUE GAS ANALYSIS

It is important to keep the fire level and free from holes, especially along the side walls, corners and between the doors. Low carbon dioxide indicates that too much air reaches the fire, so partially close the ashpit doors and use the Orsat as before and a position of the doors will soon be found for various thicknesses of the fuel bed.

After the fire has been burning for a time and is getting dirty, analyze for oxygen and carbon monoxide. Probably from ½ to 1½ per cent. of carbon monoxide with a low percentage of CO_2 will be obtained. It is then time to clean the fire, for there are clinkers on the grate and incomplete combustion is taking place. According to a report of the Bureau of Mines, 1 per cent. of carbon monoxide lowers the efficiency of the furnace 20 per cent.

Fig. 2 shows the relation of carbon dioxide, oxygen and carbon monoxide to the supply of air. These readings were taken from a boiler before any changes were made in the firing or draft. It may be noticed that as the air supply increases, due to holes in the fire, the carbon dioxide decreases and when the air supply becomes insufficient the carbon dioxide runs up as high as 16 per cent., with the formation of excessive carbon monoxide, as there is not enough air for complete combustion. This condition is shown at 2:15 and 4:15. At 7:30 the clinkers interfered with the air distribution and carbon monoxide was formed with a low percentage of carbon dioxide.

It has already been noted that the carbon dioxide drops when the whole grate is covered with fuel at one time, as the temperature of the furnace is lowered beyond the point at which the gases will ignite. To keep an even furnace temperature and a more constant percentage of carbon dioxide, it is better to cover one-half or one-third of the grate at a time.

As previously noted clinkers cause a double loss in lowering the carbon dioxide and causing the formation of carbon monoxide. The writer has learned these facts with the Orsat: Do not disturb the fire more than is absolutely necessary. Fire level instead of using a leveling bar. If the grate is of the shaking and dumping type, do not use it to break the clinkers except during the cleaning period, for as soon as the grate is opened the fine ash and a great deal of live fire is dropped into the ashpit and the clinkers remain on the grate. Use plenty of water on the green coal as the evaporation of the water tends to prevent clinkerous formation. Do not cut down a fire for it takes more coal to level it than to clean it. So when the percentage of carbon dioxide begins to drop, due to a dirty fire, clean it.

Every particle of air, entering the boiler beyond the fire, is raised to the temperature of the flue gas and sent up the chimney at a dead loss. This amounts to from 1 to 5 per cent. in different plants. Take a candle and examine the setting for leaks, also around the inspection doors and those to the combustion chambers, where the door frames fit the setting, where the blowoff pipe comes through the wall and where the breeching fits the setting. If a Babcock & Wilcox boiler, look for leaks at the point where the flues are blown and around the doors on the boiler fronts.

Asbestos putty will stop the leaks except on the doors and to decrease the

leakage at these points coat the door jam with shellac and stick on a single strand of asbestos wicking. After this has been done, try the Orsat again and it will probably be necessary to readjust the ashpit doors for the different thicknesses of the fire.

To attempt any further economy a draft gage and flue-gas pyrometer are necessary. The least draft that can be used, and still keep up steam without the formation of carbon monoxide, is the best for the coal pile. With this end in view connect the draft gage to the boiler in the same manner as the Orsat. Take account of the thickness and condition of the fire, position of ashpit doors and damper, then draw a sample of flue gas and analyze it. Increase the draft and analyze another sample. If the percentage of carbon dioxide is increased more draft is needed, but if the carbon dioxide is decreased cut down the draft as it is already excessive. A thin fire requires less draft than a thick one, and within certain limits the greater the draft the use of the Orsat, draft gage and pyrometer a saving of 1.3 lb. of coal per kilowatt-hour was made.

This question of flue-gas analysis has a human factor. Results require the cooperation of the fireman. Some plants having a continuous CO_2 recorder, pay the fireman a bonus for a high daily average of carbon dioxide; others post the daily averages of their men on a blackboard and engender a spirit of rivalry. With only the hand instrument it is necessary to get the men interested in some other way.

In the first place, the fireman does not shovel coal because he wants to. He would be happy if he were shown how to keep up steam without that labor. When the Orsat is brought into the boiler room, show the instrument to him and explain what it is for. Teach him to operate it and do not be afraid to let him use it. Tell him what per cent. of coal can be saved by increasing the carbon dioxide and work out each step together. He will then begin asking questions and

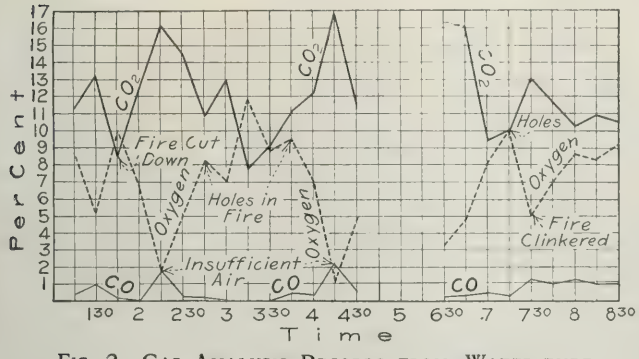


FIG. 2. GAS ANALYSIS RECORDS FROM WATER-TUBE BOILER USING NO. 1 BUCKWHEAT COAL

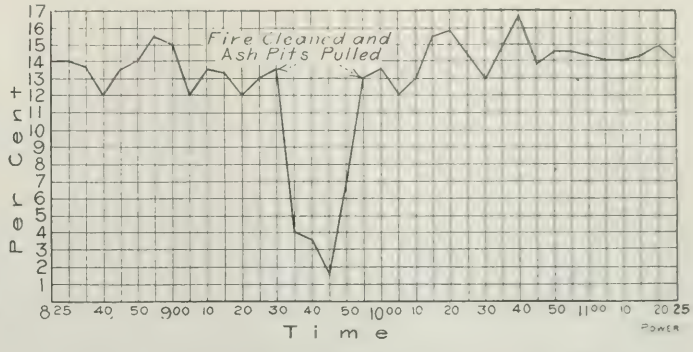


FIG. 3. CO_2 CHART FROM SAME BOILER USING NO. 1 BUCKWHEAT COAL

higher the temperature of the escaping flue gases.

Some engineers advocate a flue-gas temperature of 100 to 150 deg. above that of the steam. For instance, if the steam pressure is 150-lb. gage, its temperature being 366 deg., then the flue-gas temperature should range from 466 to 516 deg. These temperatures require economizers to regain some of the heat, but with no economizer these high flue-gas temperatures represent a loss of 20 or 25 per cent. of the heat value of the fuel, unless a large amount of draft is necessary for combustion. So where no economizer is used it is more economical to carry the flue-gas temperature from 40 to 80 deg. above the temperature of the steam.

If a thin fire is necessary in any boiler, coal can be saved by decreasing the draft. A short time ago the writer saved 300 lb. of coal in 6 hr. by reducing the draft on one boiler to 0.24 in.

Fig. 3 shows a chart taken from the same boiler as Fig. 2. While not perfect it shows how nearly constant the carbon dioxide can be kept. With the make suggestions, and when he has been shown the answers with the Orsat, he will be interested and it will be an easy matter for him to analyze the results obtained with the apparatus.

To obtain carbon dioxide averages over a longer period of time, it will only be necessary to assemble a collecting bottle as described in the third section of Mr. Terman's article, recently published. When all this work has been done and the results are studied it will be found that the coal consumption has been reduced and the fireman's knowledge of his work has been greatly increased.

Hunting for Engine Troubles

BY C. A. GILSON

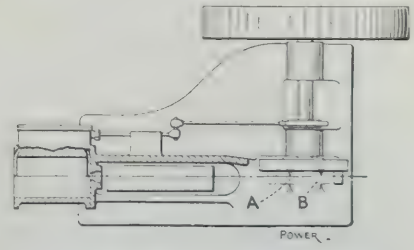
An engine that has steam leaking around the piston rod or that gives two distinct knocks in the wrist at each revolution, is not a very good recommendation for the engineer in charge. A few simple tests may assist in locating the cause of the trouble.

The crankpin may be out of line with the center line of the engine shaft. To test for this fault remove the connecting-rod and level the main shaft. The crankpin should then be tested with a level and should prove level in any position. If it is out of level in any position, the crankpin is out of line. The pin can be easily tested with calipers to see whether it is worn out of round.

Wear and cutting in the main bearing, due to a heavy belt pull, a poor job of babbitting or a mistake in adjusting the quarter-boxes, may throw the engine shaft out of line.

To realign the engine, remove the cylinder head, crosshead, piston and piston rod. Then run a line through the center of the cylinder and stuffing-box, as shown in the sketch. This line should be carefuliy centered either by using an X-centering device in both cylinder and stuffing-box or by using calipers.

Fasten the line at some point beyond the crank and turn the crank to a point near the head-end center and measure the distance *A* shown in the illustration. Then turn the crank to the opposite center and carefully measure the distance *B*, as shown. If these two measurements are equal, the engine is in line; if not, the shaft should be shifted in the proper direction. Some times the shaft can be realigned by adjusting the boxes, preferably the bearings farthest from the



LINE CENTERED IN THE ENGINE CYLINDER

crank. A shaft seldom needs endwise adjustment, as a mistake in design would be the only cause of this fault.

A leaking stuffing-box may be caused by the piston rod and crosshead guides not being parallel; this causes the piston to have a slight vertical motion together

with the regular sliding motion. This vertical motion destroys the packing and may cause the rod to bind and heat in the stuffing-box.

Worn or badly adjusted crosshead shoes may throw the piston rod out of line with the guides. The piston rod can be tested with calipers to determine its position relative to the guides and the proper adjustment made until the piston rod is the same distance from the guides at all points when the crank is on the crank-end center.

These tests should show aligning faults in the engine.

High Vacuum with Surface Condensers

By P. E. Reynolds

The article refers principally to large turbine units, 5000 kw., or over, used with surface condensers having separate hot-well and dry-vacuum pumps, although much of it applies to condenser practice in general.

With vacuums* between 27 and 29 in. of mercury on large turbines it has been determined by tests that 0.1 in. increase in vacuum corresponds approximately to 0.1 lb. decrease in steam consumption per kilowatt-hour, so that a turbine operating with 27 in. of vacuum and having a consumption of 15.5 lb. of steam per kilowatt-hour would reduce its consumption to 14.5 lb. if the vacuum were raised to 28 in. Figuring the cost of generating 1000 lb. of steam at $0.20, this gain in vacuum would mean a saving of approximately $1.75 per kilowatt per year or for a 10,000-kw. output a saving of $17,500 per year.

Just what would be considered a good vacuum depends on the design of the condenser apparatus in question and the supply and temperature of cooling water available. Most surface condensers are calculated to condense from 6 to 8 lb. of steam per hour per square foot of cooling surface, so that an 8000-kw. turbine using a maximum of 150,000 lb. of steam per hour would require a condenser having about 20,000 sq.ft. of cooling surface. Sufficient cooling water should be available to keep the temperature of the circulating water discharge below 90 deg. F., and the dry-vacuum pump should have a volumetric capacity of at least 30 times the volume of the condensed steam. Under these conditions a vacuum above 28 in. of mercury would be considered good. However, nothing but the highest vacuum obtainable with the apparatus at hand in the best of operating condition should be considered satisfactory.

Naturally the highest vacuum obtainable with any given condensing outfit is limited by its design and the amount and temperature of the cooling water, but there are many items affecting the maintenance of a high vacuum which come entirely within the control of the operating engineer and require careful attention. The most important of these items are:

Leakage of outside air into the condenser.

Dirty condenser tubes and accumulations of trash on the tube sheets which would restrict the flow of circulating water.

The condition of the dry-vacuum pump air cylinder and valves.

The condition of the circulating pump.

To locate the cause of low vacuums it is necessary to install thermometers at the hotwell, at the circulating-water inlet and outlet and on the air suction to the dry-vacuum pump. A mercury column or gage, preferably the former, should be connected to the top and the bottom of the condenser to give the vacuum, and all readings should be corrected to a 30-in. barometer for comparison. Whenever possible a thermometer should be installed to give the temperature of the turbine exhaust steam, as this temperature can be used as a check on the mercury column. All the thermometers mentioned should be installed in permanent form for use in regulating the regular operation of the condenser as well as in determining troubles.

Fig. 1 shows a general turbine and condenser installation and indicates the positions for the various thermometer and mercury-column connections. One column may be connected as shown to measure the vacuum both at the top and the bottom of the condenser.

If the cause of poor vacuum is insufficient cooling water, it will be shown by the high temperature of the circulating-water discharge. With a constant load on the turbine this temperature is controlled by the inlet temperature and the amount of water used. Since the inlet temperature is not under control the only way to lower the discharge temperature is to increase the amount of cooling water. This may be done by overhauling the circulating pump and by making sure that the suction and discharge pipes and the tube sheets are clear of trash or any obstruction which might restrict the flow.

An approximate formula for calculating the amount of circulating water required is as follows:

Assume 1000 equals the latent heat of steam in B.t.u. at 1 lb. abs. times its quality plus the difference in temperature between the hotwell water and the tem-

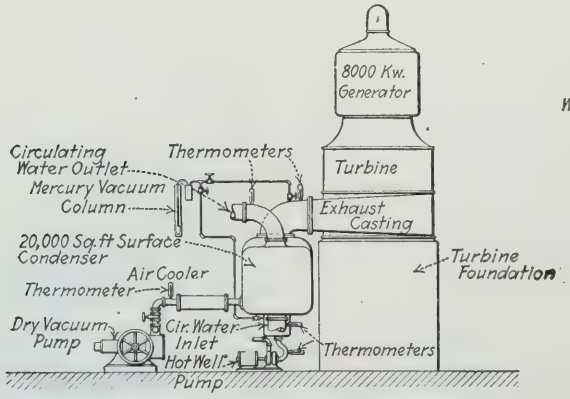


FIG. 1. LOCATION OF THERMOMETERS

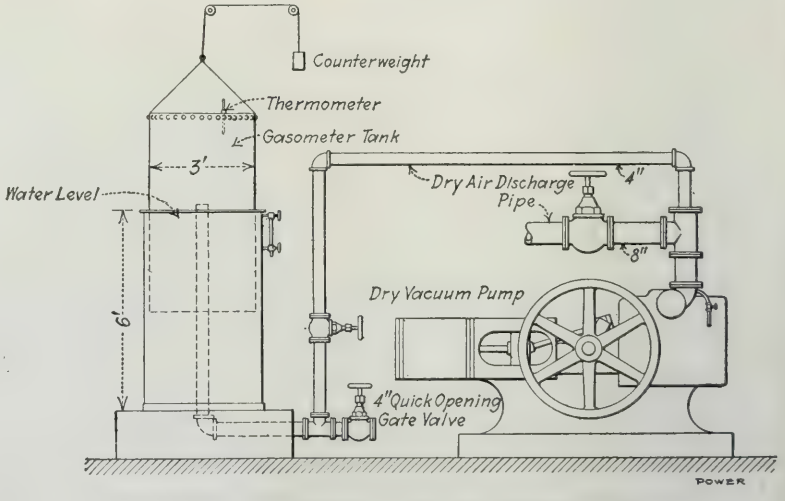


FIG. 2. GASOMETER TANK FOR METERING AIR

*All vacuums are referred to a 30-in. barometer.

perature of the steam corresponding to its pressure, then

$$(A \times 8.33 \times 60)(t_o - t_i) = 1000\ S$$
$$A\ (t_o - t_i) = 2\ S$$

where

A = Circulating water in gal. per min.;

t_i = Inlet temperature of water, deg. F.;

t_o = Outlet temperature of water, deg. F.;

S = Condensed steam in lb. per hr.

To solve for A transpose the above equation to

$$A = \frac{2\ S}{t_o - t_i}$$

or if t_i, S and A are known, it may be made to read

$$t_o = \frac{2\ S + A t_i}{A}$$

This equation shows that with the load S remaining constant, the temperature of the circulating-water discharge t_o depends entirely on the inlet temperature t_i and the flow A, and is independent of the vacuum maintained.

Another cause of low vacuum may be air leakage at the valves and piston of the dry-vacuum pump. To determine if such is the case the pump should be tested at a time when the turbine is not in use by closing the valve in the air-suction line and noting the vacuum obtained when operating the pump at its normal speed. Under these conditions a pump for high-vacuum work should produce a vacuum of within 0.4 or 0.5 in. of mercury of the height of the barometer. If the pump does not produce this vacuum, a general overhauling or repacking and tightening of the glands may produce the desired result. It may also be possible that the air cylinder and valve are not being properly lubricated and cooled.

Probably the most prevalent cause of poor vacuum is air leakage into the condenser. This may happen at any of the numerous connections and joints in the condenser or vacuum stages of the turbine. The table following gives the results of a test made on a 10,000-kw. unit to determine the effect of different sized air leaks on the vacuum obtained. Air was let into the condenser near the top through various sized orifices in flat disks and no readings were taken until after the vacuum had become constant.

Size of orifice, in.	0	1/16	1/8	3/16
Load on gen. kw.	9000	9000	9000	9000
Vacuum at top of condenser—corr. to 30 in. barometer	28.90	28.88	28.72	28.51
Size of orifice, in.	1/4	5/16	3/8	7/16
Load on gen. kw.	9000	9000	9000	9000
Vacuum at top of condenser—corr. to 30 in. barometer	28.31	27.96	27.74	27.30

It may be seen that leaks aggregating the area of a 5/16-in. hole will cause a drop of 1 in. in vacuum and increase the cost of operating a turbine of this size by nearly $50 per day.

To test for air leaks the condenser and lower stages of the turbine should be filled up with warm water and all joints showing leakage carefully tightened and then given a heavy coat of asphalt paint or shellac. The water should be warm to reduce the strains caused by cooling and prevent the forming of condensate or sweat on the outside, as the latter might preclude the possibility of finding small leaks. The use of a candle flame is unsatisfactory, especially if there are drafts around the condenser. It may be possible that the condenser castings are porous or contain minute cracks. For this reason the complete surface under vacuum should be carefully examined and given a good coat of paint. Considerable air leakage may also occur at the seals where the main turbine shaft leaves the casing. Care should be taken to have sufficient pressure, either steam or water, at these points to prevent any possibility of air leaking into the condenser.

It has been determined by numerous tests in plants using open heaters and where a good part of the feed water is returned from surface condensers that the amount of air carried into the condenser with the exhaust steam is too small to appreciably affect the vacuum, the bulk of air coming in through leaks in the shell. It is evident that to maintain a constant vacuum this air must be removed by the dry-vacuum pump at the same rate at which it leaks in. Otherwise it would accumulate and the condenser become air-bound.

To determine the rate of air leakage it is only necessary to measure the air discharged by the dry-vacuum pump. A method for doing this, which has been tried by the writer and found very successful, is to meter the air in a gasometer tank, as shown in Fig. 2. The air is discharged by the dry-vacuum pump into an inverted cylindrical tank which is sealed with water at the bottom and balanced with a counterweight. This tank rises as air is discharged into it, so by timing any definite rise of the tank and knowing the corresponding volume, the leakage into the condenser can be calculated in cubic feet per minute at atmospheric pressure. A gasometer tank 3 ft. in diameter and 6 ft. high and capable of measuring 30 cu.ft. per min. is large enough for ordinary condenser work. The gasometer method of determining air leakage is simple, accurate and inexpensive and requires only a few minutes for its operation.

To obtain a good vacuum the leakage should be kept below 20 cu.ft. of atmospheric air per minute, which can easily be accomplished by careful attention.

Crucible steel was first successfully produced in the United States in 1832 at the works of William and John H. Garrard, at Cincinnati, Ohio.

Economic Riddance of Turbine Scale

By Morgan G. Johns

In a certain generating station operating about 65,000 kw. of turbines great difficulty was experienced by having the wheel vanes, or buckets as they are commonly called, constantly fill up with a hard scale deposited from the river water used in the boilers. With the increased obstruction to the passage of the steam the stage pressure would increase at a rapid rate, making it necessary to take a side out of each turbine about once every four or five months, and in the spring, when the freshets would bring down the refuse from up the river, the vanes would have to be cleaned as often as every three weeks. This was expensive, costing approximately $80 per turbine every time it was undertaken.

A sample of the deposit cut from the vanes with a cold chisel was taken to a chemist for analysis with the intention of arriving at some substance which could be put in the feed water as a boiler compound to counteract the scale producer and bring it down as a precipitate in the boiler. In this way fairly good results were obtained.

One day as the writer was passing a machine being cleaned, he stopped to talk with the foreman in charge and asked the nature of the substance he was using to partly loosen the scale before using the chisels, and was told that it was coal oil. Immediately the idea was conceived of pumping the oil in with the steam just as it entered the turbine and in that way cut the scale out without dismantling the machine or even taking it off the line.

To do this the load on the turbine was reduced to about one-tenth of normal and the vacuum pump was stopped, the free exhaust valve opened and the turbine thrown over on high pressure. About six barrels of coal oil were then pumped into the steam line just in front of the admission valves on the running side by a small triplex motor-driven pump. The time consumed was about one hour, after which the pump was stopped, the condenser pump gradually started up until the vacuum had picked up enough to close the free exhaust valve, and when normal conditions were reached the load was again put on the machine.

With coal oil selling at $3.50 a barrel, the cleaning cost $21 as against $80 by the old method, and without having taken the machine out of commission for one minute of the time.

To make sure that this method was having no ill effect on the vanes, a side was taken out of one of the machines which had just been washed. The scale had been entirely removed from both the guide and wheel vanes, leaving them absolutely smooth and highly polished.

Tom Hunter, Hoisting Engineer

By Warren O. Rogers

The operation of ordinary pump valves are explained, why a pump operates with a shock at the end of the stroke, and how cut-off valves prevent the evil

It is strange how unusual experiences will disturb one's rest at night. I found it so, for, after bidding Hunter good night upon our return from visiting the mine pumps, I tumbled into bed to dream of gigantic pumps working to free a small chamber of water, in which I was imprisoned. The pumps were unable to keep the water down and I soon found myself climbing up the perpendicular walls in an endeavor to escape, but slipping, plunged into the black, surging, gurgling water with a splash. Ugh! it was cold. Then I awoke and found that my cold plunge was the result of wind and rain blowing upon me through an open window.

Morning finally came, but not before I had escaped death in a dozen ways from one danger or another. After a cold plunge and a good breakfast, I boarded the first street car that came along, determined to go to the end of the line, wherever that might be, and as the rain of the night before had clarified the atmosphere, it was most enjoyable.

When the car stopped and the conductor shouted "All out!" I found myself within a stone's throw of No. 2 shaft of the Susquehanna Coal Co. Stifling any fears of possible explosions and flooding mines I began to hunt for Superintendent Koalbreaker, who, the men about the mine said, "would be difficult to locate, as he was all over the lot and didn't wear out his clothing very much holding down the office chairs. But when you find him he will talk with you."

One man had seen Mr. Koalbreaker headed for a culm pile about a mile distant from the shaft. Jumping on one of the small locomotives that was going that way, I eventually located him, and, after stating my errand, how I would like to examine his underground pump rooms, was told to go as far as I liked. He turned me over to Master Mechanic Morgan, who introduced me to a fireboss and we were soon 600 ft. under ground.

The first pump room visited was close to the foot of the shaft; it is shown in Fig. 1. The construction of the walls and roof was different from what I had before seen. The walls were of concrete to a height of about 5 ft. with I-beams imbedded in the concrete at the bottom end, and supporting heavy I-beams at the top which held the ceiling in place. The side walls were built up above the concrete with discarded 3-in. pipe lengths; the ceiling was lined with discarded 6-in. sump pipe. The space back of and above the pipe was filled in with earth. The interior of the pump room was painted white and the floor was of cement. As I had expected to see something far different, the equipment and surroundings impressed me with their neat appearance.

We were given a welcome by the pump men, even the cat on the upper rung of the ladder, a favorite resting place, I was told, seemed pleased to see us.

The compound, duplex, outside-packed Goyne pump was running, pumping 2400 gal. of water per min. against a head of 608 ft., which included the suction lift.

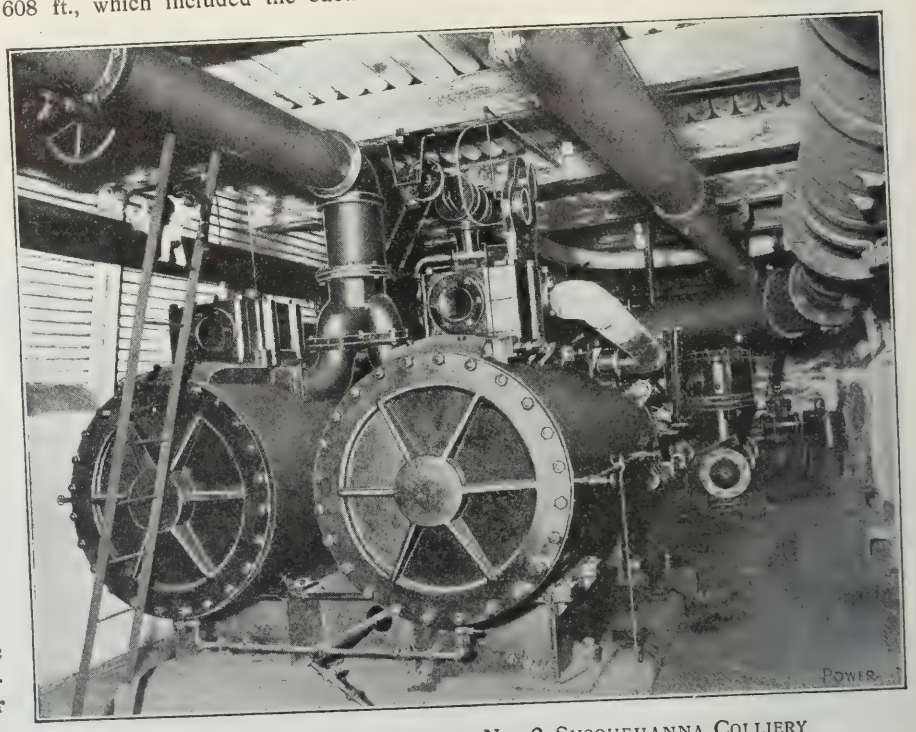


FIG. 1. GOYNE STEAM PUMP AT NO. 2 SUSQUEHANNA COLLIERY

The steam cylinders were 23 and 40 in. and the water cylinders 40 in. in diameter, and 48-in stroke. A perspective view of the pump is shown in Fig. 2.

I found what I was told was a new kind of valve-gear in that, besides the usual type of slide valve common to duplex direct-acting pumps, there were also adjustable cutoff valves. These were not for the purpose of using steam expansively at all speeds, but to obtain economy in steam consumption when running at slow speed or at speeds considerably below normal, also to obtain quick operation without shock and jar when running at high speed.

While examining the pump, several men stopped at the entrance to the pump room and turning, I saw Hunter was one of the party. Having finished his business with his companions, he suggested that he accompany me to act as a sort of guardian angel, to which I gladly assented.

"Know anything about the valve action on this pump?" I asked.

"I think so," replied Hunter. "It is like this, with a duplex pump the steam valves on one side of the pump are operated by the movement of the piston on the opposite side, and *vice versa*, by intervening mechanism by which each piston in its stroke movement reverses the valve on the opposite side, which releases the exhaust steam from one end of one cylinder and admits steam to the other end of the opposite cylinder. Neither piston movement cuts off the admission of steam, which follows the piston full volume with the valve wide open up to the end of the stroke.

"The stroke is ended by the piston covering the exhaust outlet near the end of the cylinder, entrapping the remaining steam, after which a slightly further forward movement of the piston compresses the steam so that the piston is brought to a stop before striking the cylinder head."

"Makes a sort of cushion," said I.

"Exactly; but when the piston has made approximately three-fourths of its stroke, its movement has reversed the valve on the opposite steam cylinder and its piston begins its stroke. The relative

positions of the pistons are such that by the time the one that has made three-fourths of a stroke reaches the end of its travel, the other piston will have made one-fourth of its stroke and will have the second and third fourth of its stroke to make before it reverses the valve that starts the piston that has been at rest against its cushion on its return stroke.

"Now, here is something for you to remember. During the time the piston is resting on a steam cushion, about one-half the time, the steam-chest admission valve has been wide open, and as the throttle valve between the steam line and each steam cylinder is wide open, the direct-acting pump must have a considerable power margin to meet the varying conditions, particularly in mine work. In the average case, there is a large difference in the main steam-line pressure and the required cylinder pressure to overcome the water-end resistance."

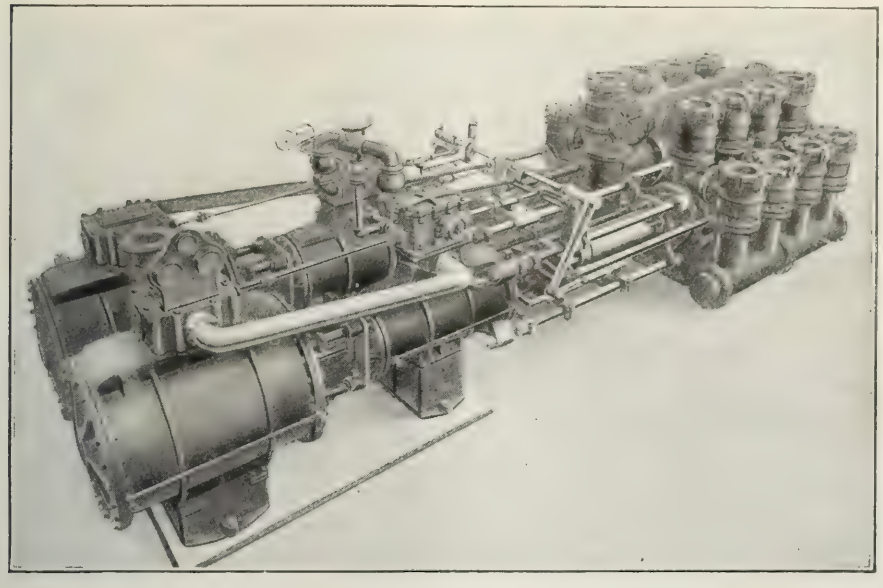


FIG. 2. PERSPECTIVE VIEW OF THE GOYNE STEAM PUMP

"Yes, you explained that clearly only yesterday. It looks to me as though there might be considerable cylinder condensation."

"Sure, for during the time the steam pistons are at rest, at the end of the stroke, all of the steam that flows uninterrupted from the high-pressure steam pipe into the steam of less density in the steam cylinder is wasted as the pump cannot return any work. If a cutoff were arranged and used, better economy would be obtained."

"Well, how does this cutoff take place?" I inquired, watching the cat frisking with a piece of waste.

"When steam is admitted to the steam cylinder by the movement of the opposite piston, the cutoff is operated by the movement of the piston of the cylinders to which it is applied and controlled, and is adjusted by turning a handwheel to suit the speed at which the pump runs. The cutoff takes place, depending on the speed, just as the piston reaches the end of its fixed stroke or at any point earlier up to within 7 in. of the end of the stroke, which is enough for the highest speed, because in direct-acting pumps there is no flywheel in which to store up power that can be utilized in completing the stroke after steam has been cut off. And the only available power for the purpose is the momentum of the moving parts of the pump, and the moving column of water. It has been found in practice that these are enough with pumps running at high speed, even when working against heavy pumping heads, to complete the last 6 in. of the stroke with the steam cut off entirely."

"From what you have just said, the advantage acquired with the adjustable cutoff in the last 7 in. of the stroke and its resulting economy are obtainable only when running at high speed."

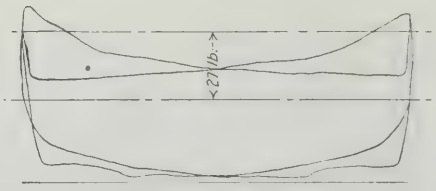


FIG. 3. DIAGRAM OF PUMP OPERATED WITHOUT THE CUTOFF ATTACHMENT

"Yes, that is right, and the economy gained by expansion is only secondary to the economy gained in slow and normal speed operation by preventing the equalization of the high steam pressure in the supply pipe and the low steam pressure in the steam cylinders after the pistons have completed their stroke, and are resting on their steam cushions where they can give up no work.

"In slow running, the cutoffs are adjusted so that steam is cut off just before the ending of the stroke.

"There were some tests run to obtain a comparison of the steam consumption calculated from indicator diagrams and speed of pumps combined with smooth running with and without the cutoff attachment and controlled by the main valve and with the supplementary valve in each case."

Hunter at this point pulled out his ever-ready notebook and, after referring to its pages, said, "The piston speed obtained without excessive shock and without cutoff was 126, and with cutoff, 153 ft. per min. for each piston. Under these conditions the steam consumption per indicated horsepower per hour was 57.05 lb. without and 38.7 lb. with cutoff."

"Why should cutoff adjustment cause the pump to operate without shock or jar? Just tell me that," I demanded.

"Easiest thing in the world," replied Hunter. "In a duplex direct-acting steam pump, the steam-admission valve is moved wide open at the beginning of the stroke and is not closed, nor can it be closed, to cause the end of the piston stroke, which is accomplished by the action of a steam cushion, as I have already told you, by entrapping the exhaust steam in approximately the last inch of the stroke. This brings a heavy mass of moving parts from full speed to a dead stop in a second's time usually with a slam.

"In a small pump this does not cause much inconvenience as the parts are of light weight, neither would it be noticed much in large pumps running at normal speed. But pumps used in pumping mine water are frequently required to operate at speeds beyond their maximum rating and could safely do so were it not for the shock at the end of the stroke caused by the abrupt stop, and there is danger of accidents when operating to any considerable extent above normal speed."

"Yes, you told me that before. But that doesn't explain why the cutoff valve removes shock."

FIG. 4. DIAGRAM OF PUMP OPERATING WITH THE CUTOFF ATTACHMENT

"Gee, but you are like an old woman; you ask questions, and before they can be answered, you kick because they aren't. Now, listen; you take the same pump and provide it with cutoff valves. After each piston has made about two-thirds stroke, the cutoff valves begin to close, gradually reducing the volume of steam flowing into the cylinder and even

though the mean piston velocity is far in excess of the maximum rating, its progress is retarded by the adjustment of the cutoff to correspond with the increased speed that will cause an early and quiet ending of the stroke without the slightest shock, because the velocity of the piston is retarded before it is brought to a sudden stop by the compressed steam."

Hunter detached a loose leaf from his notebook on which were two diagrams, Figs. 3 and 4, taken from the pump under discussion.

"When the pump was running without the cutoff, the greatest steam economy was obtained when the pump was controlled by the main throttle valves. When the pump was running with cutoff the greatest steam economy was obtained while the pump was controlled by the supplementary throttle valves." He drew a dotted line through the two diagrams, remarking as he did so: "This dotted line shows the relative steam economy of the two conditions of pumping, the two diagrams being taken with the same piston speed and with the pump working against the same head, one using 40, the other 64.8 lb. of steam per indicated horsepower."

Just then the cat made a jump for a rat and further conversation was dropped for the time being, due to the excitement of the chase which resulted in the rat escaping in the black depths of the mine.

Design of Indicator Reducing Rigs

By A. L. Westcott

The author describes several types of indicator reducing motions he has operated with different steam engines, an air compressor and gas engine.

In selecting a reducing rig for a given engine, numerous considerations should be taken into account. A rig suitable for a long-stroke, slow-speed engine may be unsuitable for a high-speed short-stroke engine and the resulting large inertia of moving parts.

Indicator-reducing rigs may be divided into two classes based on geometrical design, those which produce correct reductions and those which approximately do so. Figs. 1 to 4 belong to the approximate classification, and Figs. 5 to 8 represent the correct forms.

Fig. 1 indicates a rig of the pendulum form, arranged for use on a horizontal engine. The swinging arm *OA* is connected by a link *AB* to a pin *B* on the crosshead. A cord attached to a pin *E* leads over a guide pulley *D* to the indicator. The guide pulley should be so located that the line *ED* is horizontal, or, if the distance is short, so that the horizontal line tangent to *D* bisects the versine of the angle *EOF*. The position of the pivot *O* should be such that the link *AB* swings equally above and below the horizontal line, and the swing on each side of the vertical line *OA* should be equal, making the angles *JOA* and *AOH* equal.

The degree of accuracy of this rig may be determined by the following considerations. Produce the line of the link in any position, as *JG*, until it intersects the vertical center line *OA*, at some point as *M*. The relative velocities at points *A* and *B* are as the ratio of the radial distances *OA* and *OM*. The distance *OM* varies with the position of *B*, between the extreme values of *OM* and *OA*.

The velocity of the drum cord is proportional to the length of the perpendicular let fall from *O* upon the line of the cord. If the length of the pendulum *OA*, represents to scale the velocity of the

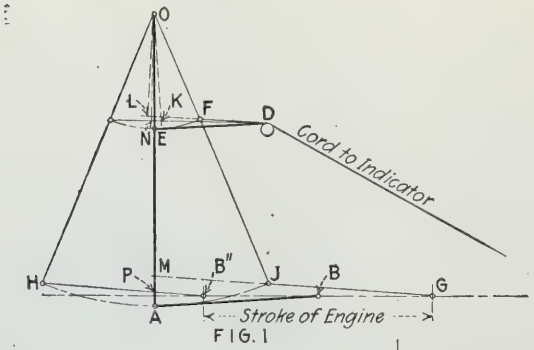


FIG. 1

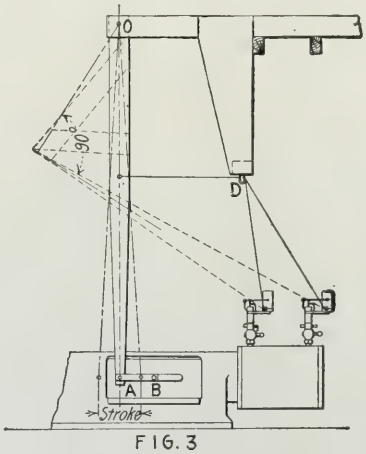
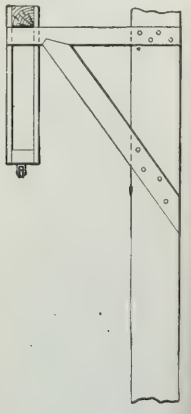


FIG. 3



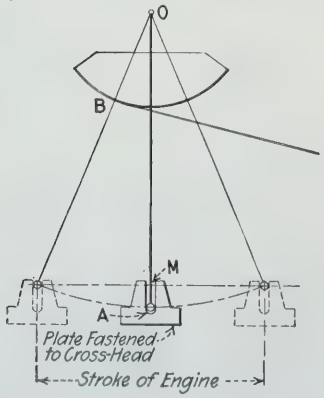
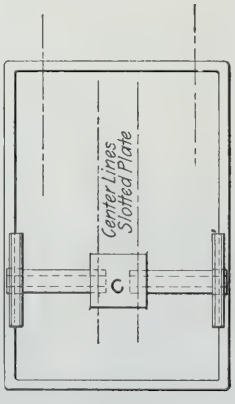


FIG. 2



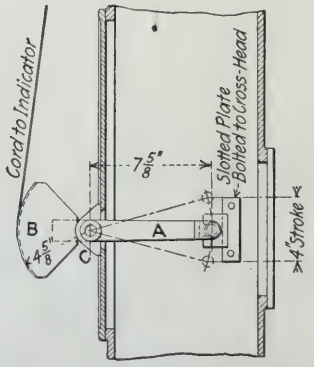


FIG. 4

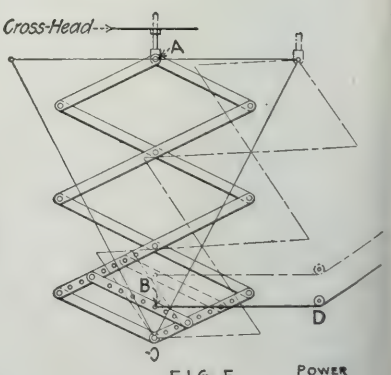


FIG. 5

POWER

FOUR TYPES OF REDUCING MOTIONS WHICH GIVE APPROXIMATE REDUCTIONS

point A in any position, then to the same scale the perpendiculars OK, OL, etc., represent the corresponding velocities of the drum cord. The ratio of the velocities of drum cord and crosshead, then is given by the expression

$$\frac{OL}{OA} \times \frac{OA}{OM} = \frac{OL}{OM}$$

for the position of the pendulum OF. Similarly, for the position OA, the ratio is $\frac{OK}{OA}$, etc. With the mechanism proportioned as shown, the values of this

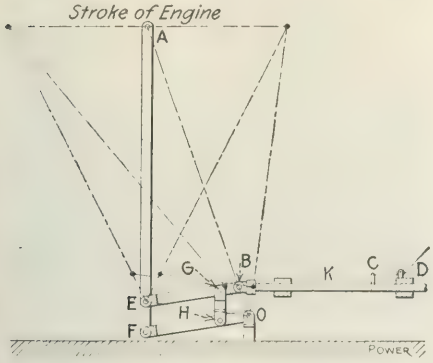


FIG. 6. MODIFICATION OF PANTOGRAPH

ratio in the two extreme and the mid positions are as follows:

$$\frac{OL}{OM} = 0.389$$

$$\frac{OL}{OA} = 0.395$$

$$\frac{ON}{OP} = 0.388$$

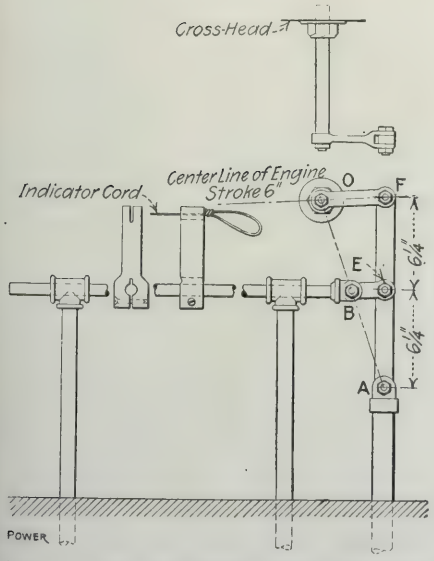


FIG. 7. REDUCING MOTION FOR AIR COMPRESSORS

The stretching of the cord by the inertia of the drum may cause some appreciable error at high speeds. And this is increased because the cord is deflected over a guide pulley. At high speeds it is preferable to so arrange, where possible, that the cord leads directly to the indicator without change of direction.

This may be accomplished by substituting the brumbo pulley, which is a circular segment, whose center is at O. The cord may lead off tangent to this in any direction, as in Fig. 2. With this modification, the length of the perpendicular OK, Fig. 1, let fall from O upon the line of the cord, becomes constant for all positions. The variation in velocity ratio of the cord to the crosshead is thus increased, the values for the three positions becoming 0.438, 0.395 and 0.418, respectively. The sacrifice of theoretical accuracy may be offset by the gain, due to the decrease in cord tension.

Fig. 3 shows a rig constructed for a 6x8-in. high-speed engine. It is intended as an example for students in the construction of an inexpensive rig for temporary use. With these proportions of swinging arm to the stroke of the engine the motion is practically exact. An improvement in the device by which the cord could be led in a straight line to the indicator, is indicated by dotted lines.

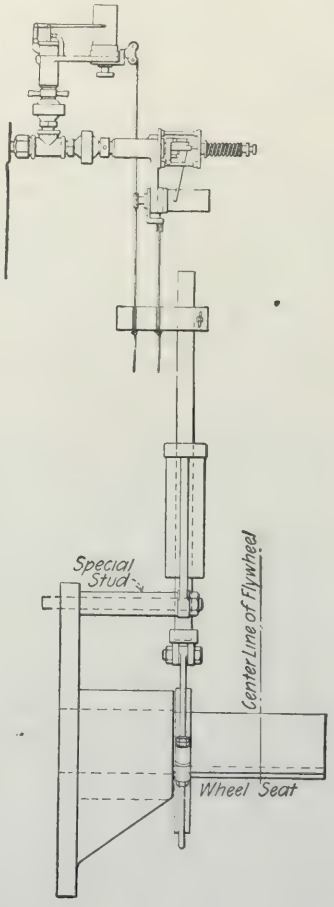


Fig. 2 shows a modification of the pendulum form adapted to high speeds. Besides the brumbo pulley, the swinging arm is driven at A, by a slotted plate attached to the crosshead; the center line of the slot is perpendicular to the stroke of the engine. The velocity ratio of the pin A, to the crosshead for any position is determined by drawing a line through the center of the pin perpendicular to OA, and noting its intersection with OA. The length of this intercept from O represents the velocity of the crosshead to the same scale that OA represents the velocity of the pin.

At the ends of the stroke, the ratio is $OA : OM$. At mid-stroke it is equal to unity. The ratio of cord velocity to that of the crosshead is, therefore, $OB : OM$ at the ends of the stroke, and $OB : OA$ at the center, its value varying between these two limits. With the same proportion of arm radius to the stroke of the engine, this form is more exact than the combination of brumbo pulley and link AB, shown in Fig. 1.

Fig. 4 shows a rig of slotted plate and brumbo pulley combination designed for a small vertical, compound engine with an inclosed crank case. With the proportions chosen, the error is very small. A roller, mounted upon a pin passing through the forked end of the lever arm A is actuated by the two sides of the slot in the plate. It has the advantage of lightness and a small number of moving parts, which is important when dealing with speeds of over 400 r.p.m. The splash system of lubrication employed oils all pins and bearings, and a cover C, of sheet metal, over the spindle bearings catches oil that splashes through the opening in the cover plate and drains it back into the crank case.

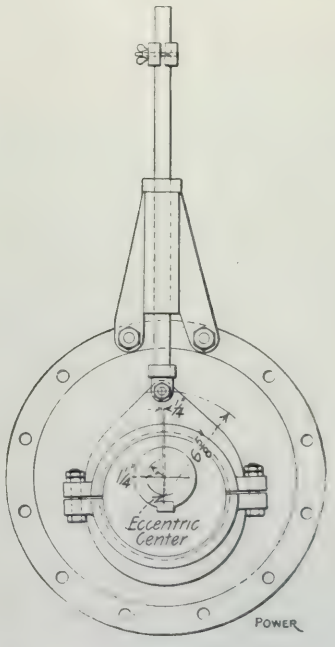


FIG. 8. REDUCING MOTION FOR GAS ENGINE

For long-stroke low-speed engines, a form of the pantograph shown in Fig. 5 is used, where the lengths of stroke of various engines varies from 30 to 60 in. The point O is fixed, but A is attached by a pin connection to the cross-

head. To give a correct motion, the pin *B*, to which the cord is attached, must lie on a line with *O* and *A*, and the cord *BD* must lead off parallel to the stroke of the engine. The ratio of the length of the indicator diagram to the length of the stroke will then be equal to *OB*: *OA*. Means are provided for changing the position of *B* by moving the cross-bar, as indicated by the dotted lines, so that in working on engines of different stroke, the lengths of indicator diagrams can be kept approximately constant.

Figs. 6 and 7 are modifications of the pantograph, and built for a permanent fixture on a single engine. In Fig. 6 the point *A* is attached to the crosshead, and is disconnected when not in use. The essential conditions to give a correct reduction of the piston motion are that the point *B* must be on the line *OA*, the link *OF* must be parallel to *BE* and *GH* parallel to *AF*. The point *B* moves in a straight line parallel to the path of *A*. The sliding rod *K*, to which the motion of *B* is transmitted, is guided by supporting bearings and is long enough to reach both cylinders of a tandem-compound engine, so that the indicator cord connections may be short. The cord is guided over a pulley, so that *CD* is parallel to the motion of *K*.

The rig shown in Fig. 7 was designed for an air compressor of 6-in. stroke. The point *O* in this case is attached to the crosshead and *A* is the fixed point, but the character of the movement is the same. As in Fig. 6, *OF* must be parallel to *BE*, and the point *B* must be on the straight line *OA*. The link *GH*, Fig. 6, is omitted, as the sliding rod is guided by bearings and the stroke is short, so that the maximum angle of *BE* with the center line of the rod is small. The compressor speed is 150 to 200 r.p.m., and to avoid carrying the indicator cords over guide pulleys, wooden brackets were made and clamped on the sliding rod. The method of attaching and detaching the cord to the bracket is shown.

The purchase of a gas engine brought the problem of a suitable reducing rig. The engine was of the two-cylinder, vertical, inclosed-crank case type. Fig. 8 shows the design of reducing rig that was adopted. The flywheel was removed, and an eccentric set on the shaft next to the main bearing, with the eccentric and crank angles coinciding. The eccentric strap was attached by a pin connection to a drawn-steel tube slide, guided by a babbitt-lined bearing. Two long studs replaced two of the cap bolts which held the end housing to the frame of the engine, and these support the guide casting. A wooden bracket, clamped to the slide tube, and slotted at the proper radial distance was used to attach the indicator cords.

Two indicators were used, one with a stiff spring for high-pressure diagrams, and one with a weak spring and stop, for taking the exhaust and intake lines. The former was set in a horizontal position with a very short direct drum-cord connection. The cord of the weak-spring instrument made a right-angle bend between the bracket and the drum. As the work represented by the weak-spring diagram is but a small per cent. of the total work, any slight error from this cause was considered negligible. The mechanism was designed to exactly reproduce the motion of the engine piston. Thus, the radius of the engine crank being 5 in., and the length of the connecting-rod 26½ in., and it being desired to obtain a diagram 2½ in. long, the eccentric radius was made 1¼ in., and the length of the eccentric strap was made one-fourth of 26½, or 6⅝ in. As the center line of the engine cylinders is 1 in. in front of the vertical center line through the crankshaft, the center line of the slider was set ¼ in. forward. This arrangement gave an exact reproduction, on a reduced scale, of the motion of the engine piston.

Derrick Hoist Safety Devices

By R. Reddie

Two of the most prolific causes of injury and death to the operator of a derrick-hoist are a fall when climbing the back-leg to oil the mast-head pin; or being hit by the loose end of a broken rope as it comes lashing into the engine house.

Fig. 1 shows a black-oil force-feed oiler which is attached at the ground level to a back-leg of a derrick. A ¼-in. pipe leads up the back-leg to the mast-head and is held in place thereon by spikes, as shown. The oiler is filled through the plug cock by running the plunger back above the top level. The ¼-in. globe valve is kept closed except when forcing oil to the mast-head.

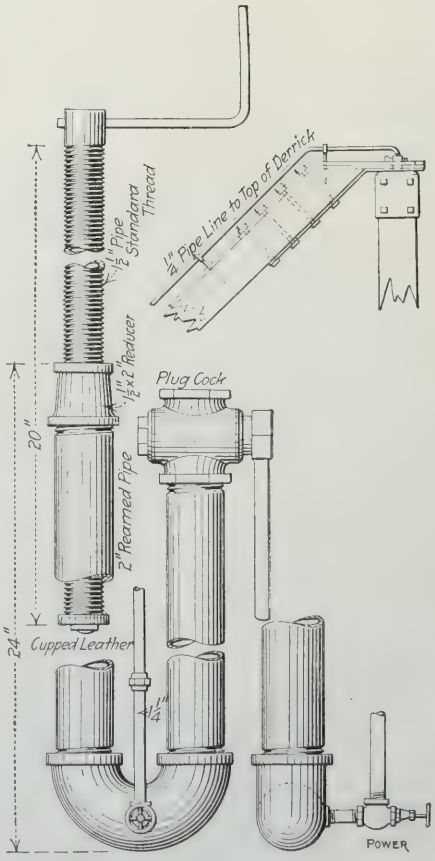


Fig. 1. Masthead Oiler

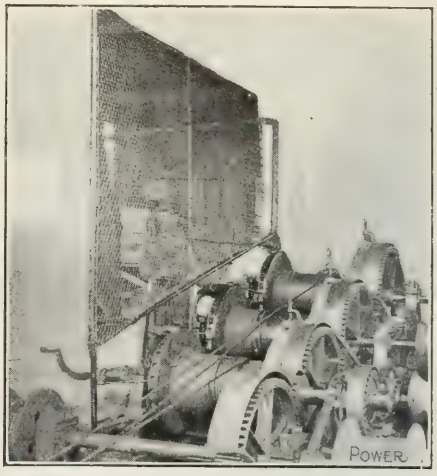


Fig. 2. Engine Room Screen

Fig. 2 shows a screen between the engineer and the drums of his engine, and is self-explanatory.

Steam Boiler Furnace Design

The design of special furnaces suited to local fuel conditions has been neglected until recently. The downtown district of Chicago was built largely during the early nineties and prior to 1900 contained perhaps not a furnace suitable for the proper combustion of Western fuels. Marked developments have been made and suitable furnaces are now built.

Anthracite coal contains from 3 to 10 per cent. of volatile matter and burns with a clear flame 1 or 2 ft. long for a few seconds after firing, after which the bed of hot carbon remains active and complete combustion is obtained. A furnace having a length or height of only 2 or 3 ft. is ample with such fuel; the main object is to insure sufficient and uniform air supply. A deficiency of air can exist with such fuels without producing smoke, but not without loss in economy. The most approved furnaces for this service have flat or slightly inclined grates, the boilers are set between 2 and 3 ft. above the grate surface, 4 or 5 ft. above the floor level and the air is supplied by a mechanical-draft system. Ten years ago these dimensions and proportions were given as standards and little or no consideration was given to fuels. It was also believed that smokeless stacks could

be obtained only by burning anthracite coal.

For semibituminous fuels containing from 8 to 25 per cent. of volatile matter, more ample furnace space must be allowed. These fuels have a large amount of hydrocarbons which, under heat, are liberated and, being long flaming gases, require a large chamber in which complete combustion can occur and sufficient air at high temperature can be mixed with the gases. This cannot obtain in the anthracite furnace and if such coals are used, unconsumed hydrocarbons, or smoke, will result.

The fuels of the Pittsburgh district border between coking and free burning coals. They are relatively low in smoking volatiles and do not require the extreme treatment of the Middle Western coals, but require a larger combustion chamber and a longer time for combustion to take place than the semibituminous coals.

In the Middle West the free burning coals, particularly well adapted to chain grates, contain from 25 to 40 per cent. of volatile matter and are difficult to handle from the standpoint of economical and smokeless combustion.

The length of flame given off from the high volatile Western coals is from 15 to 20 ft. and to prevent loss in economy, and also objectionable smoke, the flame should not enter a region capable of extinguishing combustion until it has traveled at least 10 ft. in a chamber surrounded by incandescent firebrick and having suitable mixing devices, causing the air to mingle intimately with the hydrocarbons.

With the vertically baffled boiler a suitable furnace is obtained by setting the boiler 8 ft. 6 in. high under the front header, and applying a dutch-oven furnace 6 ft. long. This permits a setting with an ignition arch 5 ft. long and a secondary arch at a higher elevation of the same length. The mixing effect, produced by the offset between these two arches, does much to complete combustion and prolong the interval while the gases pass between the fuel bed and boiler-heating surface.

The furnaces used with vertical boilers, for structural reasons, need an extensive arch construction which necessitates a "throat" over the bridge-wall which must be large unless unusual draft pressures are provided. The action of this furnace is similar to that of a horizontal water-tube boiler except that it tends to make the brickwork of the side walls hotter near the inspection door. This furnace is difficult to keep in repair, because all accumulation on the bridge-wall reduces the area of the furnace throat. Such a furnace is not well designed unless amply providing for cleaning the bridge-wall and front bank of tubes. A net area of 30 per cent. of the grate area should be allowed at this section.

The horizontally baffled boiler with tube tile on the lower row of tubes, provides a furnace roof easily constructed and repaired, and producing practically the same effect as an arch-constructed roof. With an ignition arch 5 ft. long, and the boiler set 7 ft. from the header to the floor, a simple furnace design is possible with a 42-in. projection in front of the boiler fronts. Combustion is perhaps carried out to its most complete stages in this furnace and the true furnace efficiency is usually very high.

Good results can be obtained at uniform rates of firing with a return-tubular boiler having an ignition arch only 4 ft. long. However, the boiler shell is a poor substitute for a tile roof for prolonging combustion, and a checking of this action results, usually indicated by No. 2 smoke which with variable loads becomes No. 3. This boiler should have a setting such as is used for vertically baffled boilers.

The furnaces mentioned herein are those to be considered where standard chain grates are used, burning standard coals at rates up to 30 lb. per square foot of grate surface per hour. Often the fuel requires special treatment or higher combustion rates are desired.

For higher combustion rates, the best furnace would have a horizontal arch 12 to 18 in. above the grate surface, and 6 to 8 ft. long, the greater height going with the longer arch, and both being desirable for high-ash coals and extra-long grate surfaces. The feature of such an arch is that there is virtually a 50 per cent. reduction in the draft pressure from the front to the rear or from the inspection door to the front of the arch. In other words, with 0.50-in. draft pressure through the inspection door only 0.25 in. would be obtained at the front of the arch. From this it is evident that such a furnace cannot be successfully used where the draft is only 0.20 to 0.30 in., as the reduction would result in a draft so low that combustion could not be sustained at the ignition point of the fuel.

Conversely the 0.50-in. draft could not be successfully used with the arch furnace setting as the reduction of draft is only about 10 per cent. and the resultant draft pressure would be too great at the point of ignition, and permit too much unnecessary air to enter the furnace, cooling the arch and reducing the combustion rate to below 30 lb. per square foot per hour. This capacity furnace is well adapted to coals having 20 per cent. or more of ash as high boiler ratings are so difficult to obtain with the usual grate sizes that high combustion rates are made necessary.

Coals tending to coke and cake should be treated somewhat similarly if used on chain grates. It is possible to coke such coals in a zone of relatively low draft, and then bring them into a small section of high draft and burn them fairly well.

The object of this article is merely to present for consideration, some common types of furnaces and the reasons for their design. Structural considerations and cost of floor space often prevent the most approved furnace design, but the fact is becoming more universally known that the boiler room is the important end of the power plant and permits of many times the variation in efficiencies that can occur in the engine or turbine room.

True Boiler Efficiency

Without the aid of an external agency heat flows from a hot body only to bodies at lower temperatures; therefore a boiler can absorb only that heat which is above the temperature of the water in it. The heat that is above the temperature of the boiler water is called the "heat available for the boiler." Heat below this temperature will not flow into the boiler and therefore is not available for it. The quantity of heat a boiler will absorb per unit of time depends almost entirely on the quantity of available heat delivered to it. Any commercial boiler absorbs only part of the available heat. The heat absorbed, expressed as a percentage of the heat available, is termed the "true boiler efficiency." This efficiency depends somewhat on the way the heat is presented to the boiler, but chiefly upon the arrangement of the boiler's heating surface. The true boiler efficiency is thus a ratio that is expressed by the equation:

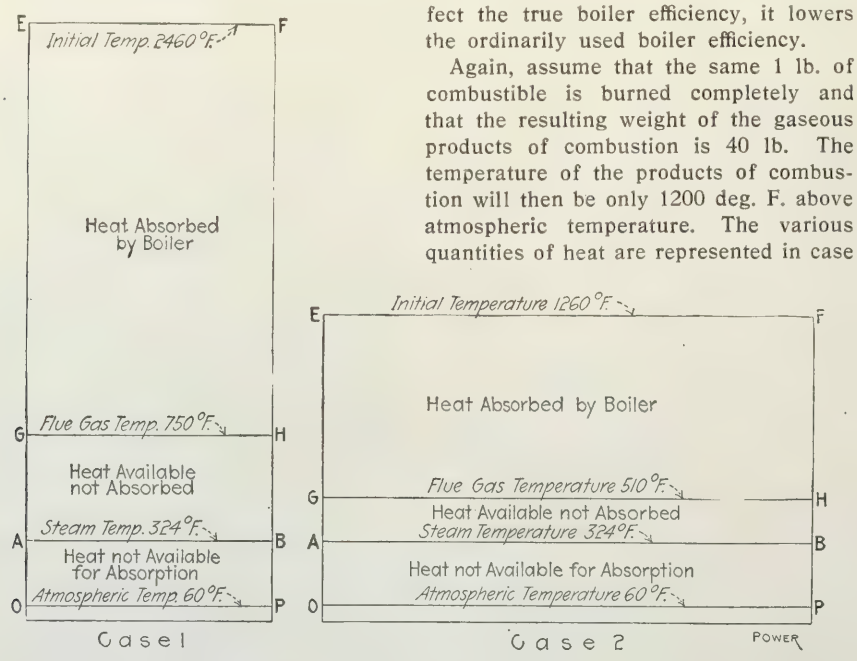
$$\textit{True boiler efficiency} = \frac{\textit{heat absorbed by boiler}}{\textit{heat available for boiler}}$$

The true boiler efficiency has been devised because it is the only true measure of the boiler's ability to absorb heat. All other boiler efficiencies used in commercial boiler tests blame the boiler for not absorbing heat that is below the temperature of the boiler water and therefore not available for absorption.

It is perhaps well to stop here and explain more fully by specific examples what the true boiler efficiency is and how it is related to the boiler efficiency ordinarily used.

Assume that 1 lb. of a combustible having a heating value of 12,000 B.t.u. is burned completely and that the gaseous products of combustion weigh 20 lb.; assume also, for simplicity, that the specific heat of the gases is 0.25; then the temperature of the products of combustion will be 2400 deg. F. above the temperature of the atmosphere. If the latter temperature is assumed to be 60 deg. F., the temperature of the products of combustion is 2460 deg. F. Then if, as in case 1 of the accompanying diagram, the results are plotted, the temperature being used as ordinates and the product of the weight of the gases multiplied by their specific heat as abscissæ, the heat generated by the burning of the combustible is repre-

sented by the area *OEFP*, the line *OP* representing the atmospheric temperature. Assume that this hot gas is passed through a boiler generating steam. If the pressure of the steam were 80 lb., the temperature of the boiler water would be about 324 deg. F. Let the line *AB* indicate this temperature. Then of the total heat generated, only the heat above 324 deg. F., or the heat represented by the area *AEFB*, is available for absorption. The remainder of the heat, represented by the area *OABP*, being below the temperature of the water in the boiler, will not flow into the water, and therefore is not available for absorption. Only the available heat represented by *AEFB* can be absorbed by an ideal boiler; therefore this heat is taken as the basis for measuring the heat-absorbing ability of a boiler.



CHARTS REPRESENTING HEAT AVAILABLE FOR ABSORPTION BY A BOILER UNDER TWO DIFFERENT TEMPERATURE CONDITIONS

Now, assume that the boiler cools the gases down to 750 deg. F. If this temperature is indicated in the diagram by the line *GH*, the heat absorbed by the boiler is represented by the area *GEFH*. Then the true boiler efficiency is shown by the ratio of the area *GEFH* to the area *AEFB;* and the ordinarily used boiler efficiency is shown by the ratio of the area *GEFH* to the area *OEFP*. The numerical values of the two efficiencies in the assumed cases are:

The true boiler efficiency,

$$E_4 = \frac{2460 - 750}{2460 - 324} = 80 \text{ per cent.}$$

Efficiency ordinarily used,

$$E = \frac{2460 - 750}{2460 - 60} = 71 \text{ per cent.}$$

If the steam, instead of being generated under 80 lb. pressure, were generated under 200 lb., the corresponding temperature of the boiler water would be 388 deg. F. On account of this higher boiler-water temperature, less heat would be available for absorption and less would be absorbed by the boiler, even though the ratios of the two heats might remain the same. If the temperature of the escaping gases assumed in the second case be 800 deg. F., the two boiler efficiencies are:

True boiler efficiency,

$$E_4 = \frac{2460 - 800}{2460 - 388} = 80 \text{ per cent.}$$

Efficiency ordinarily used,

$$E = \frac{2460 - 800}{2460 - 60} = 69 \text{ per cent.}$$

This second case shows that, although the rise in steam pressure may not affect the true boiler efficiency, it lowers the ordinarily used boiler efficiency.

Again, assume that the same 1 lb. of combustible is burned completely and that the resulting weight of the gaseous products of combustion is 40 lb. The temperature of the products of combustion will then be only 1200 deg. F. above atmospheric temperature. The various quantities of heat are represented in case 2. Although the quantity of heat generated is the same as that in the first case, twice as much of the heat is below the temperature of the boiler water, and, therefore, not available for absorption; the quantity of the heat available is smaller by the same amount. Thus, even should the true boiler efficiency remain constant, a smaller percentage of the total quantity of heat generated will be absorbed because less heat is available for absorption. If the assumed temperature of the gases leaving the boiler be 510 deg. F., the two efficiencies are:

True boiler efficiency,

$$E_4 = \frac{1260 - 510}{1260 - 324} = 80 \text{ per cent.}$$

Efficiency ordinarily used,

$$E = \frac{1260 - 510}{1260 - 60} = 62.5 \text{ per cent.}$$

The above figures show that, although the true boiler efficiency is the same, the ordinarily used efficiency is 8.5 per cent. lower than that in case 1. Finally, so much air could be used for combustion that the temperature of the products would equal that of the boiler water; then no heat would be available for the boiler, and no heat would be absorbed by it. Thus it can be seen that the useful effect of the heat generated diminishes with the drop in its initial temperature. Any factor that reduces the temperature of the products of combustion is the fault of the furnace and not of the boiler. It is therefore wrong to charge, in the above cases, the decreases in the useful effect to the boiler; they are the fault of the furnace and should therefore be charged against the furnace.

True boiler efficiency has the advantage over the ordinarily used boiler efficiency that it takes care of the variation of the temperature of the furnace gases as well as the variation of the temperature of the boiler water, caused by different steam pressures. In other words, true boiler efficiency does not blame the boiler for a lessened steam production caused by a lower temperature of the furnace gases, which really is a fault of the furnace, nor does it blame the boiler for absorbing less heat when the temperature of the boiler water is raised by raising the steam pressure.—*Kreisinger and Ray in Bureau of Mines Bulletin* 18.

Coal Output Doubles in Ten Years

An interesting fact about the United States coal production, according to the United States Geological Survey, is that in each successive decade the output is practically doubled. If the production of bituminous coal alone were considered, the record for the last 50 years would show an increase somewhat in excess of this ratio. It has been estimated that the anthracite output will reach 100,000,000 long tons annually before it begins to decline.

Statistics of coal production show that up to the close of 1865 the total output had amounted to 284,890,055 short tons. From 1876 to 1885, the output was 847,760,219 tons, more than double the total production during the preceding decade. In 1885 the production was 1,552,075,478 tons, and during the 10 years ended in 1895 it was 1,586,098,641 tons. In the decade ended Dec. 31, 1905, the total production was 2,832,402,746 short tons, and the grand total from the beginning of recorded coal mining in the United States was 5,970,576,865 short tons.

The Virginia Power Co. contemplates the erection of three power plants on New River, at Cabin Creek, W. Va., and is preparing plans for the work. It is planned to supply the New River and Kanawha coal fields with power. Complete details of the plants have not yet been worked out. It is estimated that the system will cost about $2,000,000.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

The Synchroscope

By John A. Randolph

The most widely known type of synchroscope is the invention of P. M. Lincoln which was first brought into practical use at the power house of the Niagara Falls Power Co. It successfully met the purposes for which it was designed and was soon generally recognized as almost indispensable in alternating-current switchboard operation. It is used in paralleling synchronous alternating-current machines, its functions being to indicate the speed of the starting machine relative to the running machine, the angle of phase difference and the moment when the machines are in step.

It is far superior to the older method of using incandescent lamps. The latter show the relative difference in speed, but do not indicate whether the starting machine is running too slow or too fast. They also give no indication of the angle of phase difference. Moreover, while a skillful operator can generally determine by the lamps the moment of synchronism with sufficient accuracy for practical purposes, he has no means of ascertaining this instant with precision. The synchroscope, on the other hand, not only shows at all times the relative difference in speed, but it also indicates whether the speed is too high or too low. Furthermore, it gives an indication of the angular difference in phase and shows accurately the center of the beats. By the number and accuracy of its indications, it renders a material service in time saving when starting.

In construction, the instrument is similar to a bipolar, two-phase, synchronous motor with the exception that its field is excited by alternating instead of direct current. Consider a simple coil of wire *A*, Fig. 1. Looking at the bottom of the coil, when current is applied in a clockwise direction a magnetic field will be established with north and south poles, as shown. If the coil is now placed in a field from another source, as in Fig. 2, and is free to revolve about an axis through one of its diameters, it will act similar to a bar magnet under like conditions, the north and south poles being attracted to the opposite poles of the external or main field. The force thus exerted will turn the coil until its field is in the same direction as the main field and will hold the coil in this position as long as the two fluxes remain unchanged. However, upon the reversal of the main field, coil *A*, following the laws of magnetic attraction and repulsion, will take a position 180 deg. from its last position. If the currents in coil *A* and the main field winding are alternating and vary in phase by 90 deg. electrically, no effective turning force will be exerted. Coil *A* may, therefore, be freely revolved and will come to rest in a position independent of the main field. In explanation, consider the sine wave diagram shown in Fig. 3.

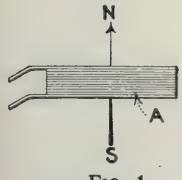


Fig. 1

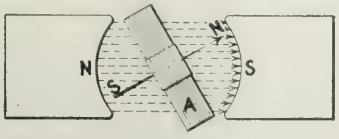


Fig. 2

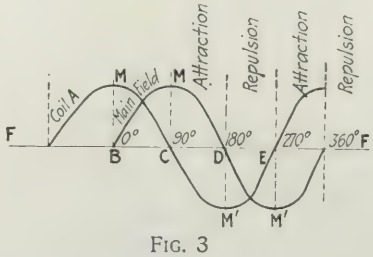


Fig. 3

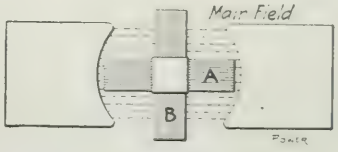


Fig. 4

Illustrating Principles of the Synchroscope

Let the two curves represent respectively the currents in coil *A* and in the main field winding, the frequency being the same for both. Starting from zero value on the line *F-F*, the currents rise in one direction to a maximum at points *M* from which they fall to zero again as the curves cross the line *F-F*. They then rise in the opposite direction to a maximum at the points *M'* whence they again fall to zero, completing the cycle. Consider the 360 electrical degrees representing the cycle divided into quadrants of 90 deg., as shown by the dotted lines at points *B*, *C*, *D* and *E*. It will be seen that when one current is at a maximum the other is at zero and *vice versa*. As a result, no turning force will be exerted at the points *B*, *C*, *D* and *E*. However, between these points a different condition exists. Between points *B* and *C* the currents (and consequently the fields set up by them) are in the same direction, hence the north pole of coil *A* is repelled by the north pole of the field. At the point *C*, however, the current in coil *A* changes its direction, causing a change of polarity in the field set up by the coil. The turning force between points *C* and *D* will therefore be opposite in direction to that between *B* and *C*, and the south pole of coil *A* is attracted by the north pole of the main field. Between the points *D* and *E* repulsion again exists followed by attraction as the curves pass point *E*. It is therefore evident that as the respective currents rise and fall, four impulses will be exerted upon coil *A* in alternate directions, the total or integrated rotating force of which will be zero. If coil *A* had no inertia, it would tend to oscillate through an angle of 90 deg. in response to these impulses. However, in practice, the inertia and friction of the coil must be considered together with the extremely short period of reversal. These elements retard the movements to such an extent that only an indiscernible vibration is produced.

Now let a second coil *B* of the same dimensions and construction as coil *A* be placed on the axis with coil *A* at the same point, but at right angles to coil *A* as in Fig. 4. Let alternating currents be sent through coils *A* and *B* and the main field winding. Consider the current in coil *B* of the same value and relative direction as that in coil *A*, but differing in phase from the latter by 90 deg. Owing to this phase difference, it follows from the previous explanation, that when the force of attraction between the main field and the coil *A* is at a maximum no turning force is exerted upon coil *B* and *vice versa*. When neither of the coils *A* or *B* are in phase with the main field a rotating flux is produced which will cause the movable element to revolve. However, when the current in one of the revolving coils comes into synchronism with that in the main field winding, equilibrium is reached and the coils come to rest.

This principle is utilized in the synchroscope, as shown diagrammatically in Fig. 5. The armature coils *A* and *B* are wound upon a laminated iron core, the angle between them being approximately 90 deg. Upon the armature shaft are three slip rings. The ring *E* is connected to one end of coil *A*, the other end of the coil being connected to coil *B*. From the common connection between the coils

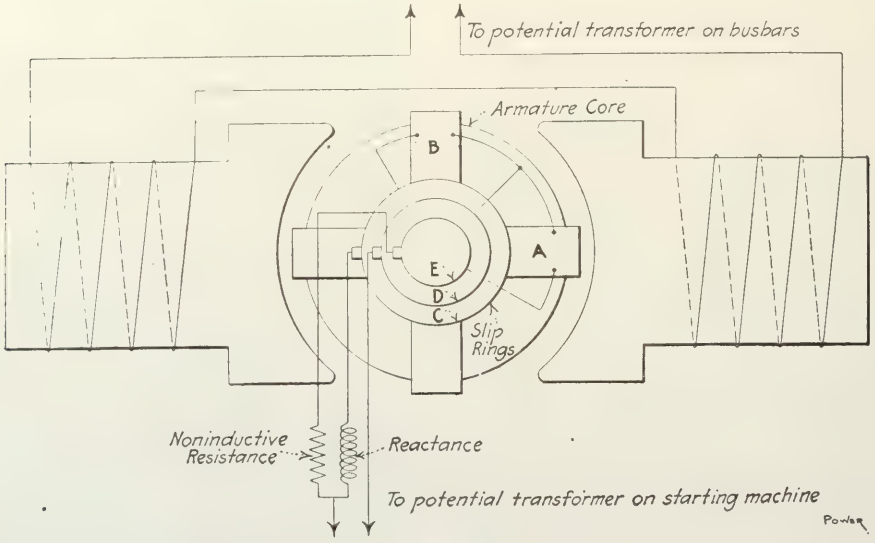


FIG. 5. SHOWING CONNECTIONS OF THE LINCOLN SYNCHROSCOPE

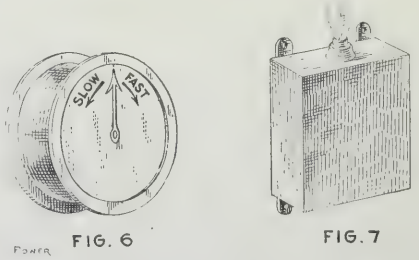
A and *B* a tap is led to the middle ring *D*. The other end of coil *B* is connected to the remaining ring *C*. To secure the required phase difference, the ring *C* is connected through a sliding contact to a reactance coil. The ring *E* is likewise connected to a noninductive resistance. The other ends of the reactance coil and the resistance are connected together and from this connection a lead is carried to the secondary of a potential transformer whose primary is connected across one phase of the starting machine. The other end of the secondary is connected to the middle slip ring *D*. The field coils are wound upon a laminated iron frame, the supply leads being connected to the secondary of another potential transformer connected across one phase of the busbars.

As commercially constructed, the armature and field are inclosed in a case as shown in Fig. 6. One end of the armature shaft extends through the front of the case and carries a pointer which swings over a white dial. At the top of the latter is an index denoting the position which the pointer will take at synchronism. Upon either side of the index are arrows pointing respectively in a clockwise and counter-clockwise direction. The words "fast" and "slow" designate the respective indications relative to the speed of the starting machine when the pointer revolves in the direction of either of the arrows, a clockwise direction meaning that the machine is running at a speed above synchronism and *vice versa*. A complete revolution of the pointer indicates a gain or loss of one cycle by the starting machine as compared with the frequency of the main busbars. The angle which the pointer makes with a vertical line through the index represents the phase difference between the busbars and the starting machine.

The reactance and noninductive resistance are placed in a convenient location outside the synchroscope proper. A common method of constructing these elements is shown in Fig. 7. A coil of wire is used for the reactance and an incandescent lamp for the noninductive resistance. The reactance coil is placed in an inclosing cage and the lamp is fastened to the top of the latter.

THE INDUCTOR TYPE

In this type of synchroscope, all the coils are stationary. The movable element consists simply of an iron armature or core carried upon a shaft; the construction is shown diagrammatically in Fig. 8. Two fixed coils *A* and *B* have their axes in the same plane but at right angles to each other. A third fixed coil *C* is placed centrally to the other coils with its axis perpendicular to the axes of the other coils at their point of intersection. Upon a shaft whose center coincides with the axis of the coil *C* is an iron core located centrally to the coil and carrying at each end a radial projection or polepiece perpendicular to the shaft.



SYNCHROSCOPE DIAL AND REACTANCE BOX

The two polepieces, however, extend in opposite directions from the shaft. When a current is passed through coil *C*, the iron core becomes magnetized and acts as a bar magnet, the polepieces becoming of opposite polarity. If now a current is passed through coil *A* an attractive force will be exerted upon the polepieces which will turn them until the flux emanating therefrom lies in the same direction as that of coil *A*. When the currents passing through coils *A* and *C* are alternating and in phase, the force exerted upon the armature will remain at a maximum holding it in the one position. However, if the two currents differ in phase by 90 deg. no force will be exerted upon the armature; hence it will be independent of the main field and can be brought to rest in any position. (See explanation of Fig. 3.) If an alternating current is passed through coil *B*, differing in phase from that in coil *A* by 90 deg. and coil *A* is out of phase with coil *C*, a revolving field will be produced which will impart a rotary motion to the armature. However, when the current in either coils *A* or *B* comes into step with that in coil *C*, the field will come to rest, bringing the armature in equilibrium simultaneously.

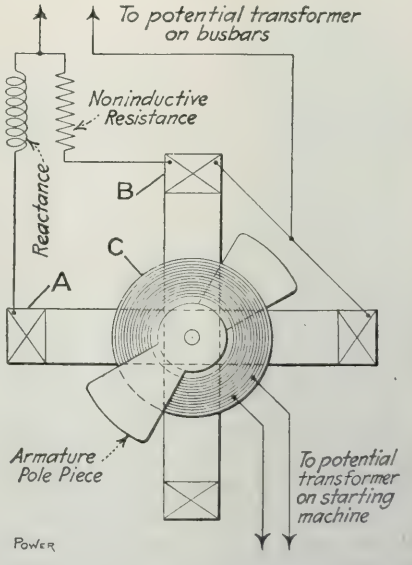


FIG. 8. ILLUSTRATING INDICATOR TYPE OF SYNCHROSCOPE

The phase difference in coils *A* and *B* is produced by a reactance and a noninductive resistance as in the movable coil type. The connections of the coils are as follows; one end of coil *A* is connected to the reactance and an end of coil *B* to the noninductive resistance. The other terminals of the reactance and resistance are joined to a common connection as are also the remaining ends of the coils *A* and *B*. From these common connections, leads are carried to a po-

tential transformer connected across one phase of the busbars. The coil *C*, on the other hand, is fed by a potential transformer across one phase of the starting machine. A pointer carried on one end of the shaft moves over a dial giving indications as in the movable coil type.

General

In practice, the synchroscope is placed upon a swinging bracket in a position such that it can be readily seen from all positions assumed by the operator in starting the various machines. In the larger central stations, an additional synchroscope is placed upon the instrument panel at the machine. The latter instrument acts in unison with that on the switchboard, indicating to the engineer whether the speed is too high or too low.

CORRESPONDENCE

Switch Panel Box

The accompanying illustrations show the details and construction of a panel box for controlling lights, where circumstances make it necessary or desirable to keep all live contacts covered, to prevent persons using them from sustaining shocks and prevent tampering with the fuses, or where sparks caused by the arcing of an improperly closed or opened knife switch might cause a fire.

As shown, the sheet-iron box is divided horizontally into two compartments, the upper containing the snap switches while the lower contains the cutouts and fuses. The cutouts and switches are mounted on wood or slate bases with a barrier of the same material between the switch and the fuse compartments. The cabinets are usually 4 in. deep and 14 in. high, their width depending on the number of switches used. Indicating snap switches are used. The mains are brought in from which ever side is most convenient. In the case shown they are brought in at the right; the branch circuits passing out at the top.

The upper door is secured with an ordinary spring catch while the lower is locked to prevent tampering with the fuses and coming in contact with live parts. The use of snap switches renders their manipulation practically free from shocks. Moreover, their positive opening and closing is entirely independent of the skill of the person operating them and makes them preferable to knife switches in this respect. A person unskilled in the use of a knife switch soon destroys it on account of opening or closing it too slowly and causing an arc. This will burn the blade and clips, sometimes fusing the copper across the clips and preventing the blade from entering. Or, the arcing may cause beads of copper to form on the blade which is closed, resulting in poor contact and heating.

Others again, probably due to ignorance or some previous unpleasant experience with knife switches (in groping for them in the dark when standing on a wet floor) will resort to a stick of wood and use it for closing or opening the switch, sometimes substituting a piece of iron with disastrous results to themselves and to the switch.

P. Justus.

Cleveland, Ohio.

Protecting Reinforced Conductors

In the Oct. 1 issue, C. V. Shear states a method of protecting reinforced circuits by fuses. I cannot agree with his conclusion that it would be better to use but one fuse to protect both wires than to fuse each separately.

The advantage claimed by Mr. Shear for the single fuse is that the two wires are connected together beyond the fuse and therefore the current would divide between them in exact proportion to their conductivity. Although with the second fuse in the circuit there would be a possibility of the resistance of its contacts differing enough from that of the first to make a noticeable difference in the amount of current carried by the second wire, yet this would be only a possibility and would have to be the result of very poor contact, causing a high resistance there, and at the same time a very low resistance in the wires of the circuit. Even if such an unbalancing should occur there would be no chance of its doing any damage by overheating the wire as each wire would be protected by its individual fuse which would blow before the current could reach a dangerous value. Furthermore, if there should be trouble on account of the fuse blowing it would indicate where the trouble might be looked for and, when found, it could be remedied by improving the contact.

As against this remote possibility of annoyance, and nothing more, due to the blowing of the fuse in the unbalanced circuit, there must be considered the actual danger that might be caused by overloading one of the wires when both are protected by a single fuse.

Even when both wires are so large as not to be easily broken there is always the possibility of the insulation on one becoming slightly damaged and permitting the current to leak to some pipe, conduit, or other wire and cause a high resistance short-circuit which would not be heavy enough to blow the one large fuse but which would blow a fuse of half the size.

In one case there is no opportunity for a fire being started by an overheated wire, while in the other there is a chance.

G. H. McKelway.

Brooklyn, N. Y.

Power Dissipating Rheostat

The dissipation of large amounts of energy in electrical testing work by means of rheostats becomes a serious problem on account of the heat generated, and special means of cooling the rheostat must be provided.

For testing a large storage battery at St. Louis, requiring the continuous dissipation of 1,500,000 watts, the device was adopted of using iron pipe as the resistance conductor and of cooling it by circulating water through the pipe. A total length of 369 ft. of 2-in. pipe was used, made up into a grid which was tapped at several points to allow the water to escape without having to pass from end to end of the grid.

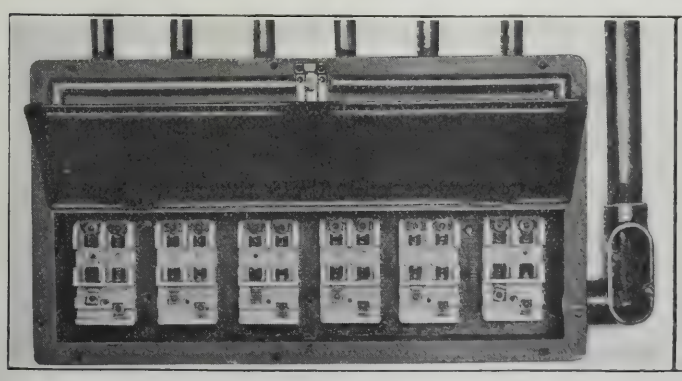


Fig. 1. Panel Box Open

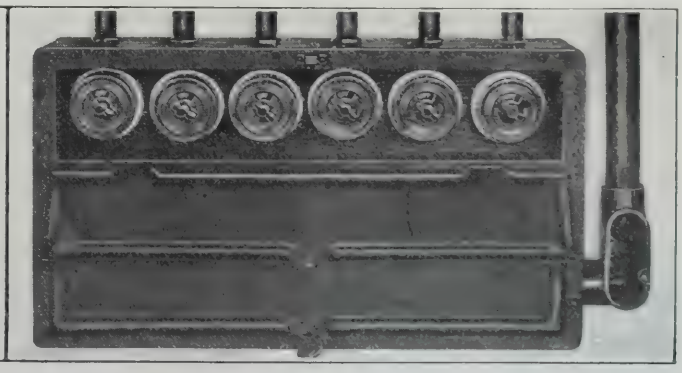


Fig. 2. Panel Box Closed

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gas Engine with Corliss Releasing Valve Gear

The Hilker-Wiechers Co., of Racine, Wis., after operating its factory for five years with a 50-hp. gas engine and producer, decided to double its capacity in 1910. After careful investigation a 200-hp. Smith combination hard- and soft-coal producer and a 200-hp. complete expansion gas engine were selected; this installation has now been in continuous operation for the past 18 months.

While it was not expected that the present load would be more than 125 hp., the problem of buying an engine which would take care of this load economically, yet have a capacity for 200 or 225 hp. when required, was solved by the installation of a 16x24-in. tandem double-acting engine which would develop one-half, three-quarters or full load with a maximum thermal efficiency by simply cutting out one or more explosion chambers at will. Direct connected to a 150-kw. direct-current generator and with a flywheel sufficiently heavy to produce a satisfactory turning moment with but one, two, three or four explosion chambers working, this engine takes care of any load within its range with a high efficiency. The method of cutting out one or more explosion chambers to suit the load will be apparent under the valve-gear description.

Fig. 1 shows a general view of the engine, from which it will be seen that no piping comes up higher than the bottom of the cylinder and that no moving mechanism extends above the side shaft, leaving the tops of the cylinders plain. The side shaft which is driven by spiral gears running in oil and which may be sued to pump oil through a filter and to all bearings, drives the governor and a single eccentric for each explosion chamber.

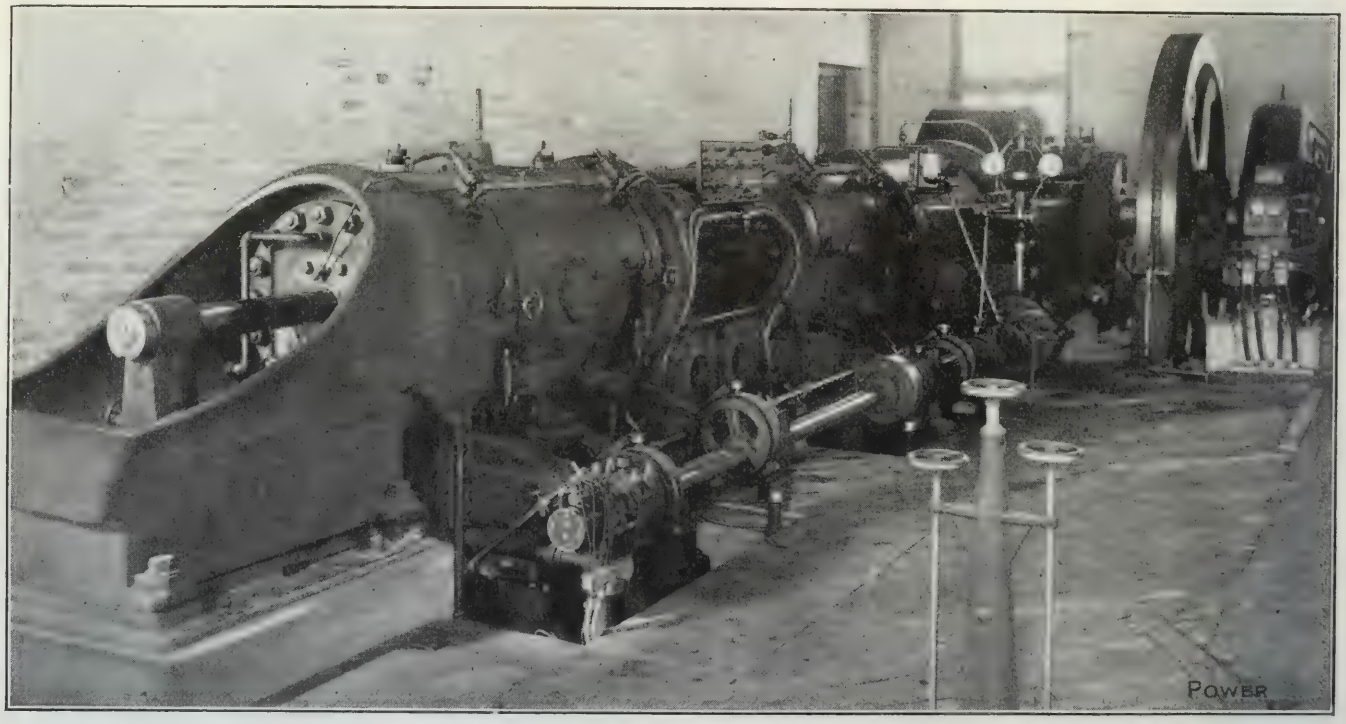


FIG. 1. SARGENT ENGINE IN SERVICE

A cross-section through the explosion chamber and valve mechanism is shown in Fig. 3, and Fig. 2 is a longitudinal section through the valve cage, valve, cylinder and head. When the valves are in the position shown in Fig. 3, the engine is on center and the piston at the end of the exhaust and the beginning of the suction stroke, the exhaust valve having just closed and the admission valve ready to open.

Fig. 4 shows the position of the valve and eccentric after the piston has made two strokes and the crankshaft has turned 360 deg. from that shown in Fig. 3; that is, the beginning of the power stroke.

When the crankshaft has turned 160 deg. during the working stroke, the eccentric has moved clockwise 80 deg. from the position shown in Fig. 4, raising the cam *A*, which is fulcrumed at *B* until the cam *C* through the push rods *E* (Fig. 2) and the exhaust ring *F* raises the poppet valve *G* against the pressure in the cylinder and the air pressure on dashpot plunger *J*. This allows the exhaust to escape through the exhaust ports *H* to the exhaust pipe. A further movement of the eccentric 100 deg. brings the exhaust ring *F* to the position shown in Fig. 3; that is, the end of the exhaust stroke.

When the exhaust is wide open and the cam *A* is in its highest position, the plate *K* engages the hook *L* journaled to the yoke *M* pivoted at *B*, and through rod *N*, levers *O*, arms *P* and collar *Q*, integral with the poppet valve stem, lifts the cut-off valve *R* and poppet *G* against the air pressure on the dashpot plunger *J*, letting in gas at *S* and air at *T*.

As the cam *A* continues to go down during the suction stroke, the inlet and poppet valves rise until the hook *L* engages the knockoff lever *U* (Fig. 4), the position of which is determined by the

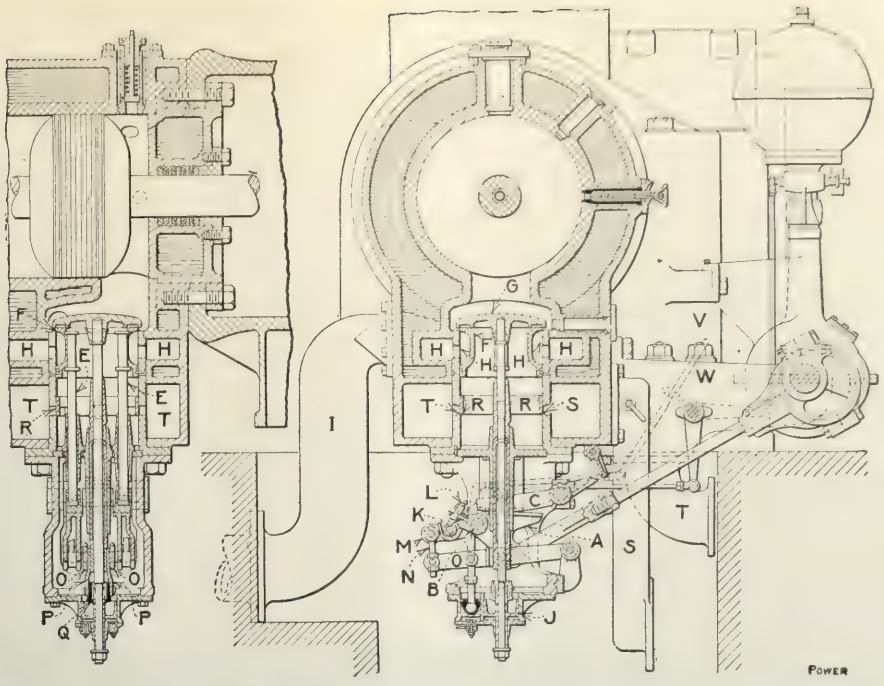


FIG. 2. LONGITUDINAL SECTION THROUGH VALVE MECHANISM

FIG. 3. SHOWING VALVE GEAR, THE EXHAUST VALVE JUST HAVING CLOSED

governor and the load. When the hook *L* is released, air pressure on the dashpot plunger closes both the cutoff valve and the poppet valve quickly and quietly, preventing wire drawing in the cylinder

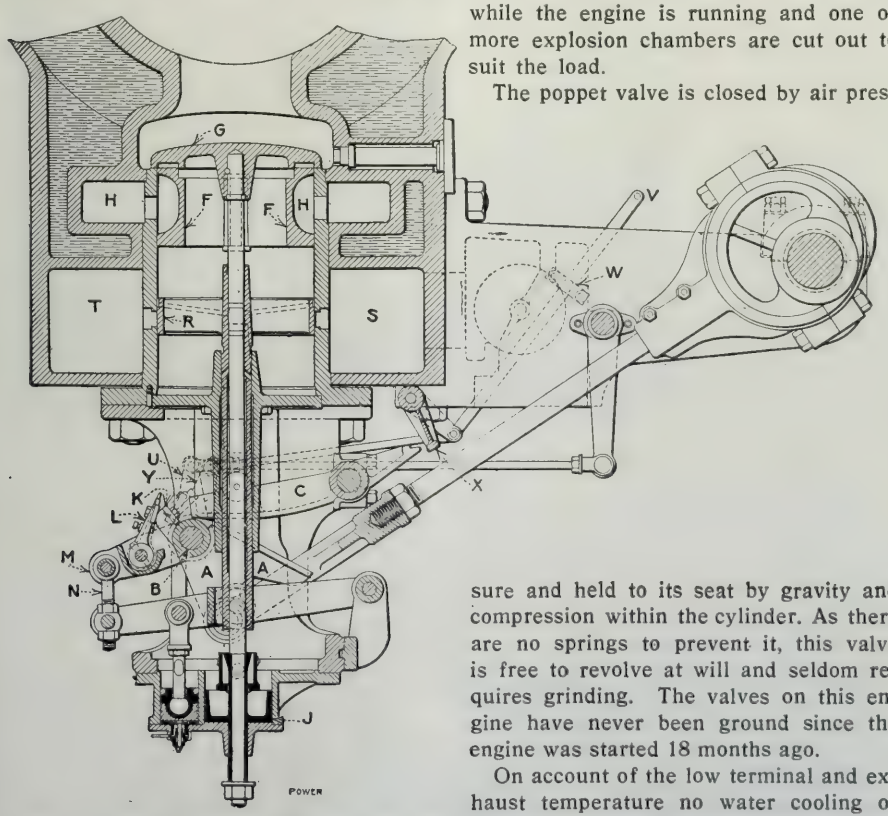


FIG. 4. VALVE GEAR AT BEGINNING OF POWER STROKE

and allowing the valves to return to the normal position shown in Fig. 4.

By raising the handle *V* it unhooks at *W*, allowing the pawl *X* to engage the cam *C*, holding the exhaust valve open and at the same time a secondary knock-off lever *Y* engages the hook *L* and prevents the gas and air valve from opening. In this way an igniter may be put in while the engine is running and one or more explosion chambers are cut out to suit the load.

The poppet valve is closed by air pressure and held to its seat by gravity and compression within the cylinder. As there are no springs to prevent it, this valve is free to revolve at will and seldom requires grinding. The valves on this engine have never been ground since the engine was started 18 months ago.

On account of the low terminal and exhaust temperature no water cooling of the poppet valves is necessary. Valves as large as 45 in. in circumference are used without water cooling with complete satisfaction. By removing six nuts and disconnecting the eccentric and governor rods the valve cage and valves may be removed from the cylinder.

Fig. 5 shows indicator diagrams taken from explosion chamber No. 4 when Nos. 1 and 3 were cut out, also when all the explosion chambers were working.

This engine was guaranteed to develop

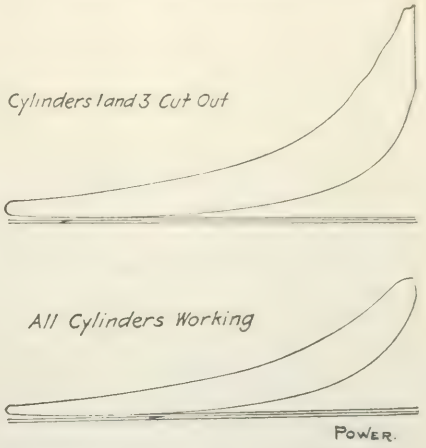


FIG. 5. DIAGRAMS FOR CYLINDER NO. 4 WITH FULL LOAD AND WITH PART LOAD

a brake horsepower on 8750 B.t.u. per hour and was designed by C. E. Sargent, chief engineer of the American Rotary Valve Co., of Chicago, and also Anderson, Ind.

The Chapman Gas Producer

A new design of bituminous gas producer for furnace work has recently been put out by the Chapman Engineering Co., of Mt. Vernon, Ohio, although experimental work on this producer has been in progress for the past six years and several have been in actual service for nearly two years. The new producer possesses several distinctive features, chief among which are the automatic feed, the manner of agitating the fuel bed and the method of removing the ashes.

Referring to the sectional view, Fig. 2, it will be seen that the top and bottom are stationary and the shell movable. The latter is divided through the hot zone into an upper and a lower section, the sections revolving, by means of the shaft and gears at the left, in the same direction but at different speeds.

Each section of the wall carries with it that portion of the fuel bed which it surrounds. Thus, without employing any agitating arms, agitation is produced through the hot zone where it is most required.

The fuel chamber has a capacity of 1000 lb. of coal—enough for half an hour. It is stationary, has no bottom and the lower edge is water-cooled. As the revolving bed of fire comes under this chamber a fresh supply of fuel is spread over the surface, thus constantly maintaining the level of the fire bed up to the bottom of the fuel chamber. It is claimed that this arrangement effectively handles fine and coarse coal simultaneously with-

out separating the latter from the former as is so often the case where a spout is employed.

The lower edge of the fuel chamber is beveled so as to produce a down-stroking effect upon the fire bed as it passes under, thus "rubbing in" the fresh fuel. Furthermore, the makers claim that the rotation of the fire bed tends to prevent the formation of blow-holes and clinkers.

The ashes are first ground between the sides of the stationary corrugated air box and the revolving corrugated wall of the lower section. After the ashes are crushed they are forced up to the top of the ash pan by three adjustable ash plows. These are kept continually in operation. As fast as the ashes are lifted to the surface of the water in the ash pan, they are automatically scooped up and carried to the discharge point. The fact that the ashes are removed upward instead of downward makes it possible to save several feet in the height required for the gas house and foundation.

Referring to details, the supporting rollers run on high carbon-steel pins, and are provided with brass bushings. Separate rollers take the side thrust and all the rollers are mounted in pairs set in equalizing yokes. The bearings are provided with large oil wells filled with waste after the manner of railroad cars. The faster gears run in oil.

The station cost per kilowatt-hour at the switchboard of only 5/8c. at the new plant of Bamberger & Co.'s department store, Newark, N. J., speaks well for the equipment and management.

CORRESPONDENCE

Heat Value of Liquid Fuels

An article in the Sept. 17 issue entitled "The Heat Value of Liquid Fuels," by A. A. Potter, gives results of a number of determinations of thermal values. The writer has had occasion to make a number of similar determinations on 60 deg. Bé. gravity gasoline and on fuel oils, using a Mahler bomb calorimeter. The samples of liquid fuel were placed in gelatine capsules for convenience in handling; otherwise the procedure was the same as when testing solid fuels. The heat value of the capsules and the calorimeter constant were accurately determined beforehand and due allowance was made therefor. Great care has to be taken to prevent leakage or evaporation of the liquid fuel after it has been weighed. The heat from the fine steel wire used in the ignition must also be considered.

By the Mahler bomb method the writer found that 60 deg. gravity gasoline averaged 20,100 B.t.u. per lb. instead of 19,411 B.t.u. as determined by Mr. Potter. This applies to the Eastern gasolines. A few samples of gasoline from California crude oil averaged about 19,800 B.t.u. per lb. In gravity and appearance the California gasoline was identical with the Eastern product but had a different odor. On a comparative test of 100-gal. lots in a gasoline motor car there was a marked difference in favor of the latter. When all weights and calorimetric readings are taken with great care and precision, very little variation is found in the resulting thermal values.

Samples of California crude oil of 18 to 24 deg. gravity which have been freed from moisture show a thermal value averaging approximately 18,700 B.t.u. per lb. A variation of nearly 1000 B.t.u. per lb. is found in samples of crude oils from different fields. On a comparative test of Wyoming coal and liquid fuel made a few years ago, the writer obtained an evaporation of 3780 lb. of water per barrel of 18 deg. gravity oil (336 lb. per bbl.) against an evaporation of 12,460 lb. of water per ton of run-of-mine coal. The thermal value of the coal averaged slightly over 12,000 B.t.u. per lb., which makes 3.3 bbl. of oil equivalent to one ton of coal; ordinarily 4 bbl. are taken as the equivalent of one ton. In burning liquid fuels much less draft is required than with solid fuels; otherwise the efficiency will be seriously impaired by an excess of air. It was found that a draft in the stack not exceeding 0.15 in. of water gave best results on a boiler of 100 hp. capacity using liquid fuel. A slight variation in the pressure of the atomizing medium will often produce a radical difference in the efficiency of combustion.

Reno, Nev. J. C. SCRUGHAM.

Attention has been called to the fact that in traction power houses, where for about three hours in the morning and in the evening it is necessary to develop twice as much power as during the rest of the day, plants which are equipped to burn oil have no difficulty in developing double the normal rate of temperature for each boiler regardless of draft conditions.—*Electric Railway Journal.*

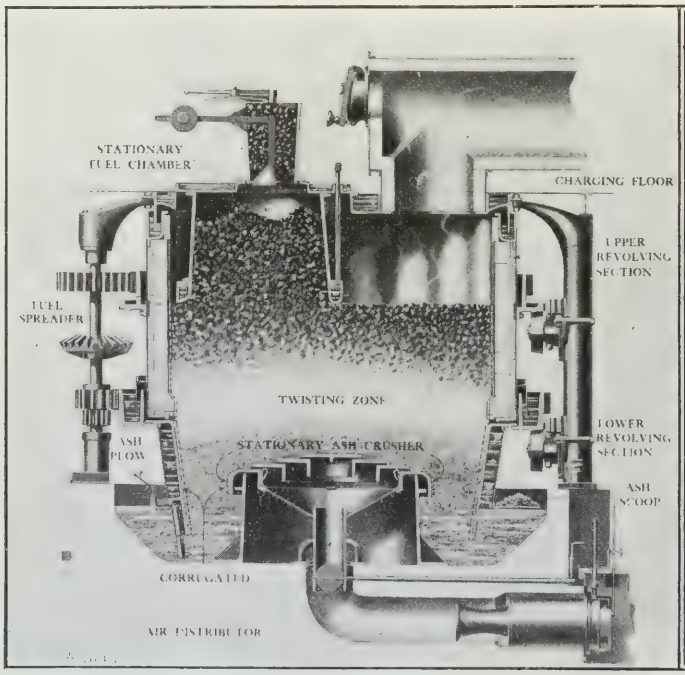


FIG. 1. SECTION THROUGH CHAPMAN PRODUCER

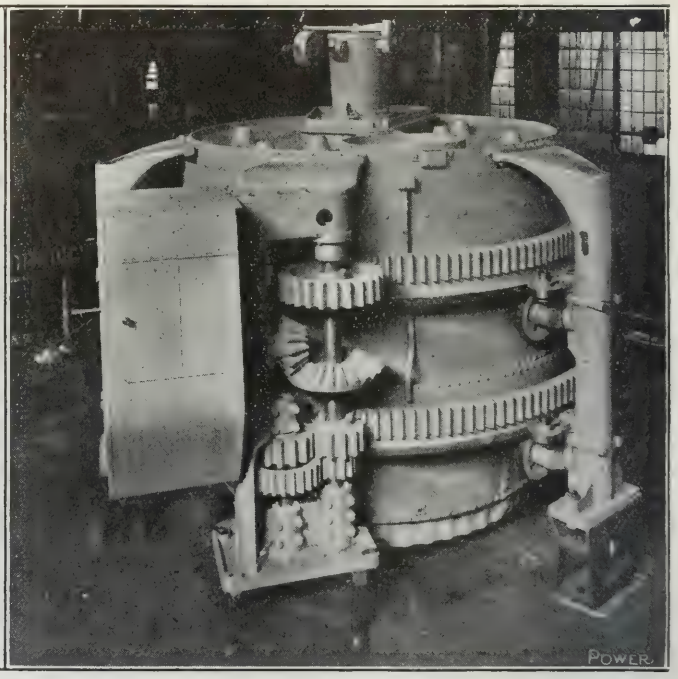


FIG. 2. EXTERIOR OF CHAPMAN PRODUCER

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Refrigeration by Direct Expansion and Brine Circulation

By John Creen

Among the different ways in which refrigeration is communicated by pipe coils, the direct-expansion system is more generally employed, being the most economical to construct as well as to operate. The ammonia or other refrigerating medium in the coils expands directly in the rooms to be cooled. When the amount of refrigeration required for a room in a given time is known, the amount of surface or the size and length of expansion coils required may be calculated on the basis, that about 10 B.t.u. will pass through every square foot of coil surface, per hour, for every degree difference between the temperature of the room and that of the ammonia within the expansion coils.

If R is the amount of refrigeration in tons required for a room in 24 hr. and t and t_1 the temperatures in the room and within the ammonia coils respectively, the number of square feet of pipe surface will be

$$S = \frac{R \times 284{,}000}{24 \times 10\,(t - t_1)}$$

The result in square feet may be readily converted into linear feet of pipe. Each 1.6 linear foot of 2-in. pipe has 1 sq.ft. of surface and for other sizes, tables of pipe dimensions may be consulted.

In practice, conditions are variable, owing to the difference in amount of pipe surface at different times, brought about by atmospheric conditions, by deposits from the water, as in the condenser, by the difference in the insulation of the rooms to be chilled, etc. For these and other reasons the above formula is subject to modification to meet special and individual cases.

The size of pipe usually employed for piping rooms varies from 1 to 2 in., and the length required varies according to circumstances, more especially with the temperature or the pressure of the expanding ammonia and the temperature at which the rooms are to be held. If a room is to be kept at 34 deg. and the temperature of the expanding ammonia is 10 deg., it will take only half as much pipe to convey a certain amount of refrigeration as if the temperature of the expanding ammonia were 22 deg. In the latter case, however, the machine works under far more economical conditions and using the larger amount of pipe so as to work with a higher back pressure is advisable.

In allowing a difference of 8 to 15 deg. between the temperatures inside and outside of the pipes it is variously assumed that in direct expansion 1 sq.ft. of pipe surface will convey 2000 to 4000 B.t.u. in 24 hr. This nearly agrees with the transmission of heat at the rate of 10 B.t.u. per hr. per sq.ft. of surface for each degree Fahrenheit difference between temperatures inside and outside of the pipe. With brine circulation, the brine with the same back pressure in the ammonia coils has a much higher temperature than the ammonia so that from 1½ to two times as much pipe is used.

If the piping is calculated on the above basis, the result will generally fall short of the piping required after the practical "rules of thumb" given in the next paragraph. These rules are founded on a very liberal basis calculated to cover unfavorable cases, as regards insulation, size of rooms, etc., it being understood that any possible surplus in piping will tend to increase the efficiency of the machine.

About one running foot of 2-in. pipe (direct expansion) will take care of 10 cu.ft. of space in rooms which are to be kept below freezing down to a temperature of 10 deg. F. On the same basis one running foot of 2-in. pipe will take care of 40 cu.ft. of space in rooms to be kept at or above the freezing point, 32 deg. F. or thereabouts. About one running foot of 2-in. pipe will take care of 60 cu.ft. of space in rooms to be kept at 50 deg. F. and above.

These rules are intended to cover rooms of 50,000 cu.ft. capacity and less, poorly insulated and operated with small differences in temperature. On a still more liberal basis it is frequently assumed that one ton refrigerating capacity will take care of 4500 cu.ft. of cold-storage capacity to be held at 32 to 35 deg. F. and that from 260 to 300 ft. of 1¼-in. pipe should be used to distribute one ton of refrigeration expediently and satisfactorily. One running foot of 2-in. pipe is equal to 1.44 ft. of 1¼-in. pipe and 1.8 ft. of 1-in. pipe as regards surface.

Brine Circulation

Formerly it was considered risky to use the direct-expansion system in storage rooms as possible leakage might cause great damage. On this account the brine system was very generally used. In this system the expansion coils are in a separate vessel or tank containing salt brine, which is cooled to the desired temperature. The brine is then pumped through pipes in the rooms to be refrigerated. In ice making the cells or cans containing the water to be frozen are suspended in the brine tank.

For general refrigeration from 120 to 150 running feet of 1¼-in. pipe in the brine tank is allowed per ton of refrigerating capacity in 24 hr., and for ice making 250 to 300 running feet of 1¼-in. pipe.

As the brine does not circulate as fast as the ammonia vapor, the surface of the brine coils in the storage rooms must be greater. In round figures it is generally assumed that the area of pipe surface should be from 1½ to two times as great as is required in direct expansion. The size of the brine pump should be estimated on the basis, that the brine should not travel faster than 60 ft. per min.

Sodium chloride or common salt has been commonly used for brine, but of late years calcium chloride is also largely used. It is convenient in preparing the brine to use a water-tight box, 4x8 ft., with a perforated false bottom and compartment at the end, with an overflow pipe for the brine to pass off through a strainer. In making sodium-chloride brine, the salt is spread on the bottom and the water fed in below as fast as a solution of the proper strength will form.

For refrigerating purposes brine should contain from 1 to 2 lb. of salt per gallon, or in general, sufficient salt to prevent its freezing at the lowest temperature in the freezing tank. If the temperature in the freezing tank does not go below 15 deg. F., 15 per cent. of salt is enough; if the temperature of freezing does not go below 20 deg. F. a brine containing only 10 per cent. of salt would be sufficient. Using stronger solutions of salt than these is unnecessary, for the specific heat gets smaller as the concentration of the brine increases and consequently the stronger the brine, the less heat a given amount of brine can convey between certain definite temperatures.

The clogging of pipes by depositing salt is another reason against the use of too strong brine. This danger, however, is not so great as that of having the solution too thin, for while it may be concentrated enough not to freeze in

the brine tank, it may be too weak to withstand the temperature existing in the expansion coil, so that a layer of ice will form around the latter and interfere with the prompt absorption of heat from the brine. For this reason the surface of the expansion coils in the brine tank should be inspected from time to time to see if any ice has formed.

Many prefer to use calcium chloride for the brine. It costs more, but keeps the pipes cleaner, causing less wear and a better conduction of heat. The freezing point can be depressed several degrees lower; hence it must be used in extreme cases. Adding a small quantity of soda to common salt brine will make it as noncorrosive as calcium chloride.

An advantage frequently cited for brine circulation is that comparatively large quantities of refrigerated brine are made and stored ahead, a supply which can be drawn on if the machinery should have to be stopped. In case of a prolonged stoppage in a small plant refrigerating brine made by dissolving ice and salt together can be circulated through the brine pipes. This, of course, is impracticable with direct expansion. It is further claimed that in small plants the general machinery may be stopped at night and only the brine pump kept going to distribute the surplus refrigeration which has been accumulated in the brine during the day's run.

Refrigerating Plant in Government Printing Office

By W. R. Metz

The Government Printing Office has a floor area of about 13½ acres divided up into the new building, eight stories high which is the plant proper, the old building four to five stories high used for storage and sorting purposes, and the annex seven stories high used by the superintendent of documents. All of the buildings are equipped with drinking-water fountains, and cold drinking water is constantly circulated.

When the new building was erected a drinking-water plant was installed, its capacity being 4000 gal. in 24 hr., based on entrance water at 70 deg. F. cooled to about 40 deg. F. This plant was very efficient, but during the past two years the daily consumption of water was so great that a larger plant had to be installed.

The Government Printing Office employs approximately 4000 people, and during the severe hot spell in July and August, the daily consumption rose to 6000 gal. per 24 hr. This was partly due to the fact that the entrance water often reached 85 deg. F., so that it was impossible to cool the water to less than about 50 or 60 deg. F. It is interesting to notice that when the water was cooled to about 50 deg. F., 6000 gal. were consumed, but when it was cooled to 38 deg. F., the consumption dropped to about 4500 gal.

Specifications for the new plant were issued under date of Feb. 9, 1912, calling for a plant of 20-ton refrigerating capacity per 24 hr. in the cooling tank, with 115 lb. pressure at the throttle valve and inlet water calculated from a temperature of 86 deg. In addition the plant must be able to produce one ton of clear merchantable ice every 24 hr.

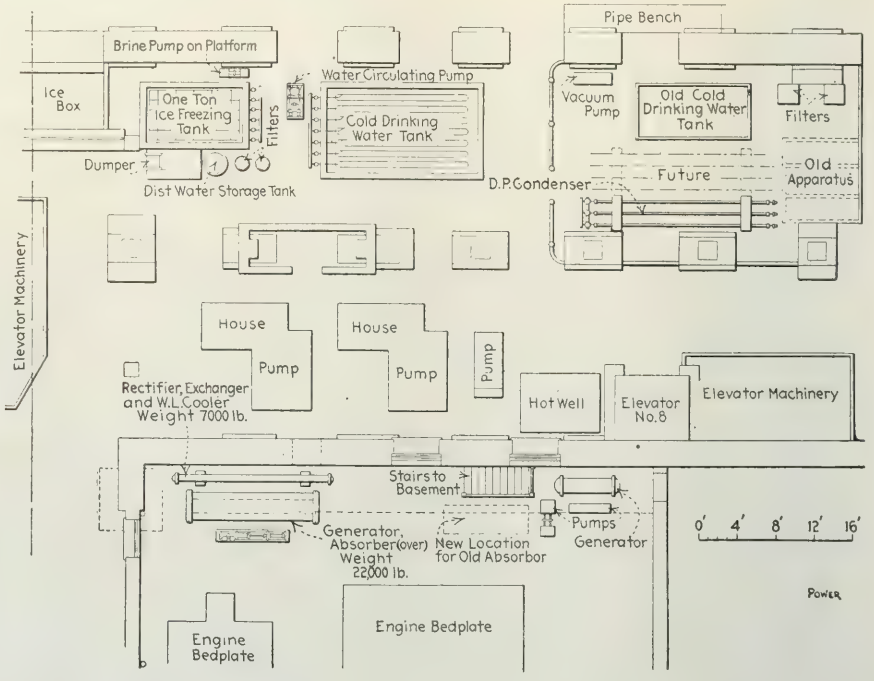
An absorption system was eventually selected. The absorber is of the horizontal type, the cooling surface consists of 2-in. No. 12 gage charcoal-iron tubes expanded into the heads. The ends are baffled for passing water 12 times through the absorber, which is 28½ in. in diameter by 14 ft. long, with 498 sq.ft. of surface.

The exchanger is of the multi-tube type, consisting of outer 8-in. pipe welded to forged-steel heads, into which are expanded seven 2-in. No. 11 gage charcoal-iron tubes, the surface area being 117 sq.ft. The weak liquor cooler has 117 sq.ft. of surface and the rectifier 58 sq.ft. The condenser consists of three stands of double pipe 2½ and 1½ in. in diameter and 20 ft. long with the necessary valves and headers for flushing connections.

The generator is horizontal and has an evaporating surface consisting of 1½-in. redrawn extra-heavy pipe with steel return bends, connections to headers being made outside of the generator. The generator cylinder is made of charcoal iron to resist the action of boiling ammonia.

For circulating the water a 7½ and 5 by 6-in. duplex brass-lined pump is provided, while the ammonia pump is 10 and 4 by 10 in. The cold-water steel storage tank, having a capacity of 6000 gal., is insulated with cork on the bottom and sides, and sheathed with 1-in. tongue-and-groove pine boards. The top of the tank is covered with two sections of tongue-and-groove flooring lined between with waterproof paper. This tank is fitted with 4500 lin.ft. of 1¼-in. galvanized-iron pipe, extra heavy.

For ice making a freezing tank made of ¼-in. steel and accommodating twenty 200-lb. ice cans, 10x20x36-in., is provided. The cans are made of No. 16 galvanized steel. The expansion coils are made of 1¼-in. extra-heavy, wrought-iron pipe. The plant is equipped with the necessary distilling and filtering apparatus, thawing apparatus, automatic can filler and geared truck hand hoist. For its operation three men are employed, one on each of the 8-hr. shifts, and one additional man for about 2 hr. each morning to pull the ice.



Layout of Plant for Government Printing Office

A Large Steam Turbine

One of the largest steam turbines built in one casing was recently delivered to the Rheinisch-Westfälischen power station at Essen, Germany, by its makers, Escher, Wyss & Co., of Zurich.

This turbine is designed to develop 22,500 hp. under normal continuous working at 1000 r.p.m. The initial steam pressure is 150 lb. per sq.in. above atmosphere and the steam temperature 572 deg. F. at the throttle, exhausting to a vacuum of 27½ in. It is claimed that this turbine can develop 28,000 hp. under constant working and up to 30,000 hp. for short periods.

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POWER

Scale in Turbines

On another page of this issue an article on the removal of scale from turbine blading indicates that it is possible for scale-forming matter to be carried over with the steam. With the proper degree of superheat and the supply line equipped with a separator and strainer this condition is surprising. It is not difficult to see how it could happen in a turbine using saturated steam or steam slightly superheated, and more especially when the character of the feed water is such as to cause frequent priming. We have heard of a case where cold water was turned into the line at a point just preceding the turbine nozzle to reduce the superheat, and with such practice it would be surprising if serious scaling did not occur. It is quite common for oil to work its way into the turbine and with the rust on the metal form in the course of time a gummy deposit which may be scraped off with a knife or cut from the blading by injecting kerosene into the intake as suggested by the article referred to.

It is difficult to account for this hard scale which must be chipped off with a chisel and comes over in sufficient quantity to eventually choke up the buckets. Ordinarily it should be precipitated in the boiler, but scale is common in the superheating tubes, and if scale-forming matter can be carried in suspension thus far, why not to the turbine?

That the separator will not always remove suspended matter, particularly with the tremendous velocities now common in steam mains, is made evident by a case in point. In a number of blowing engines, which had been giving good service, a serious cutting of the cylinders suddenly developed. The trouble was first laid to the oil but its source was soon proved elsewhere by a test of the quality and cleanliness of the lubricant. Eventually it was remembered that the trouble had started shortly after using a mechanical cleaner in the boilers. The fine particles of scale had not been removed by a thorough washing of the boilers, and had come in considerable quantity past the separators and into the cylinders of the engines. If this were possible in an engine it might readily happen in a turbine as the velocity of steam supply is usually fifty to one hundred per cent. greater.

We would be glad to have the experience and opinions of our readers on this question of scale. There are in service in this country a large number of turbines operating under different conditions of steam supply and superheat and furnished with steam from feed water of varied composition. Experience should not be lacking and what we want are the results with the specific conditions under which they were obtained. Let us have your information on the subject for the benefit of all concerned.

To Contributors

Not infrequently, after a contribution received by POWER has been edited, set up, illustrated, paid for and is about to be published, the same material, by the same author, appears word for word in another paper. Sometimes, also, after we have published a purchased article, one of our contemporaries produces the same article, doubtless because it likewise was imposed upon.

One would naturally suppose that an author, having sold a manuscript to one publisher, would understand that the subject matter of that manuscript, or at least the presentation of it therein contained, became the property of the publisher who had accepted and paid for it, and that the author had no more right to sell it over again than he would have to sell his horse or any other piece of property to a number of separate purchasers. It would seem as though a man whose ideas were so good that a publisher would want to buy them would have sense enough to know that after he had sold a thing it did not belong to him to sell again. Yet over and over, we find contributors practicing this duplicity, and, when called to account for it, in-

genuously explaining that they knew no better.

The publication of a duplicated manuscript of this kind puts the publisher of the later presentation in a very awkward position. Those who have already seen it naturally conclude he copied it without crediting its source; a rare but still occasional breach of journalistic ethics. Even if the duplication is discovered before the second publication, the publisher has lost a considerable amount expended in editing, composition, making of engravings, etc.

Another trick of some contributors which leaves the publisher not quite so defenseless but often catches him napping, is to rehash matter taken from one paper and submit it as original material to another.

We are glad to pay for new knowledge, new ideas, or for a more clever presentation of old facts or known methods, but we will not knowingly consider matter which has been sold to other publications. If it is accepted, paid for and used by POWER, we want it understood that POWER is the sole licensee to its use.

The Situation in Flanged Fittings

In our issue of Aug. 13 a comparison was made of the two standards for flanged fittings and editorially we suggested a joint committee to settle all differences. Up to the present, however, no progress has been made in this direction.

To determine the extent to which these standards had been adopted by manufacturers of fittings, valves and steam specialties, about seventy firms were circularized with the request that they indicate their position on the standards. Of the replies received thirty-seven gave information pertinent to the question.

One firm stated that it would furnish either standard as ordered and at the same price; five were undecided; seven advised that the "U. S. 1912 Standard" had been adopted, and twenty-four reported that they would follow the "Manufacturers' Standard." In the majority group are included the largest makers of fittings in the country, but in every case the replies from these firms made it plain that the "U. S. 1912 Standard" would be furnished, though at a special price.

We sincerely regret this setting up of rival standards, when the purpose of the movement which brought them into being was to develop a single standard for the benefit of user and manufacturer alike. The confusion, instead of having been lessened, is now greatly increased, and may become intolerable unless one or the other standard gives way or a compromise is effected.

Probably the manufacturers intend the additional charge for "U. S. 1912 Standard" fittings as a handicap which will automatically turn the tide in their favor, but the other standard is of too great importance to be shelved in this manner. The engineering merits of the two standards should be the real issue, and in this connection greater safety and eventually greater convenience to the user are on the side of the "U. S. 1912 Standard."

After all the user should have a good knowledge of what is wanted. The fittings installed in his plant are under daily inspection and in the course of time the defects become evident. The committees of the engineering societies who are responsible for their schedule included some of the most prominent engineers in the country and some of them the largest users of fittings. These men, drawing upon their extended experience and all the data that it was possible to get, formulated a schedule with safety and convenience as the chief objectives. They had the consumer in mind, for it is really he who foots the bills and uses the equipment. Logically they should receive his support, and if they do the "U. S. 1912 Standard" will be generally adopted.

If this standard is specified by users of flanged fittings and flanged steam specialties the manufacturers will be compelled to change their attitude. The law of supply and demand is as decisive here as for any other commodity.

There is no denying that the manufacturers should have some voice in the formation of the schedule, but they had their opportunity and persisted in viewing the question from a monetary rather than an engineering standpoint. The schedules are not so far apart that a compromise could not be effected, but as this solution now appears to be altogether unlikely we would strongly urge the user to adopt and specify the "U. S. 1912 Standard."

It is his duty to stand behind the engineering committees which have worked in his behalf. In ordering new equipment he should specify their standard and reduce the present confusion to one schedule which appears good enough to last as long as present materials and pressures are employed.

Germany and the Standard Oil Co.

According to the latest press advices the German Government is now taking an active hand in a movement to prevent monopolistic control of its oil supply by the Standard Oil Co. The proposed method of attacking this problem appears to lie in stimulated competition to be brought about by the creation of a government monopoly. By this it is not meant that the German Government proposes to enter actually into the oil business; but rather it is to grant a license to a group of private individuals to form a monopoly under strict government supervision, including the regulation of prices, etc. It is understood that this license will carry with it the right to buy out independent companies by condemnation proceedings, if necessary.

To Germany an adequate oil supply at nominal prices is of vital economic importance as a very large part of the power generated in that country is produced from this source. Profiting, no doubt, by the experiences of this country in attempting to handle the same problem, Germany appears determined to take action while it is yet time and before the monopoly has become too firmly rooted. In this she seems to realize that the only way of fighting monopoly is for the government to assume the right to keep the price of a commodity, necessary to the welfare of its people, at a price which will pay a fair profit upon the cost of production rather than to let private operators fix it at "whatever the traffic will bear."

It may interest those authorities who contend that coal deteriorates rapidly when stored under water to know that coal which has been under water for fourteen years in Havana Harbor has been analyzed by the Bureau of Mines and found to have a heat value of 15,458 B.t.u. It was sunk with the old battleship "Maine."

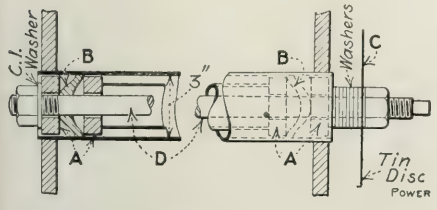
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Metallic Tube Stopper

An effective tube stopper and one that I have frequently used with success is shown herewith. When a tube bursts the hole is not often more than ⅜ in. in diameter. Occasionally though when a tube collapses the stopper cannot be inserted and then I have found the cast-iron washers and jointing to be the quickest method of repair. With the stopper I have often stopped a leak without taking the boiler off the line.

It is quite common to put in stoppers



DETAILS OF METALLIC TUBE STOPPER

while the boiler is under 180 lb. pressure. The washers *A* on each side of the white-metal disks *B* are just rough castings. The bore can be $\frac{1}{16}$ in. larger than the diameter of the rod *D*. A tin disk *C* is put on the front end which saves one from getting scalded, as the water rushes out 3 or 4 ft. from the tube if it is a bad burst.

When the stopper is in place a tap wrench is put on the square end and the nut is screwed up with a spanner. This spreads out the white-metal disks into a flat washer and they bind on the rod and around the tube. These washers will keep tight for months and can easily be drawn out by slackening back the nut when there is time to put a new tube in. It is a great advantage to be able to make this repair quickly.

J. A. CAMPBELL.

Providence, R. I.

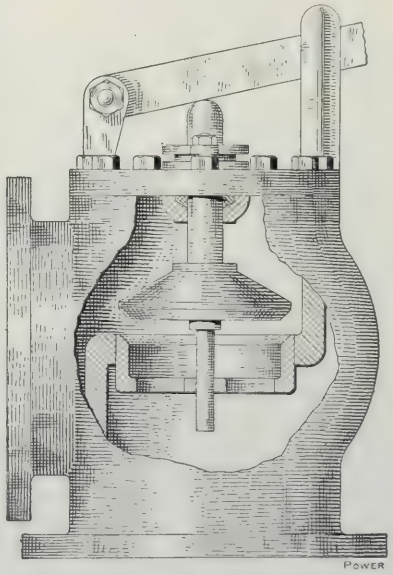
An Unsafe Safety Valve

An instance illustrating the importance of the care of safety valves came to my observation recently. Some oil got into the boilers, and to offset its evil effect, the engineer used soda very freely. This was doubtless part cause at least, for a condition brought about in the safety valves, which was indeed dangerous.

One boiler was made ready for the inspector and after having passed internal inspection was filled for a hydrostatic test. For some reason a delay of about a week occurred during which time the boiler remained full of cold water to the point of overflowing at the safety valve. After the test was made, and the boiler passed by the inspector, it was warmed up, and put into service. Some hours after this the engineer noticed a gage pressure of 115 lb. with the safety valve not blowing. As 100 lb. was allowable the pressure was at once reduced to the point where the valve should have opened.

Once operating, it continued to blow until a reduction of pressure to 70 lb. was reached.

Upon the first opportunity the boiler was again taken off the line and the valve examined, which showed that a gummy deposit had accumulated at and around the valve stem and seat. The stem was so coated as to make it almost impossible for it to slide in the guide which was about 3 in. long. This was evidently the cause for the valves sticking shut. Also the valve and seat were covered to the depth of about $\frac{1}{16}$ in. with



SHOWING PARTS OF SAFETY VALVE

the same deposit. This was doubtless why the valve failed to seat and close tightly.

Having told the inspector about the occurrence and that no trouble had been experienced before, he explained it thus: Steam will carry and deposit certain particles of matter which are the product of boiled soda and oil. This deposit had accumulated about the safety valves, but had not affected them as long as the boilers were hot, but had hardened when allowed to become cold.

EDWARD T. BINNS.

Philadelphia, Penn.

Considerations Before Reboring a Cylinder

Recently I was called upon to decide, whether an engine with a worn cylinder would stand reboring, or whether it would be advisable to order a new cylinder. As master mechanics and chief engineers have frequently to make the same decisions, the explanation of the way I proceeded will perhaps prove of interest. The original diameter of the cylinder was 36 in., but it was bored eight years ago to 36¾ in.

Indicator cards taken recently showed that quite a loss was being sustained through leakage of steam past the piston, and an examination of the cylinder showed that it was worn out of round, so that the engine was not working economically. To true up the cylinder would require boring it out ¾ in. larger, or to 37½ in. in diameter. The steam pressure was 150 lb., and the first question that came to my mind was that after another reboring, would the cylinder walls be strong enough to stand the pressure. With an air drill, I bored the cylinder in three places, using a ¼-in. drill first on the bottom, and then on each side, being sure that I did not strike any ribs.

I found that the thinnest place after truing up the cylinder would be ¾ in. I then started to figure, by using the following formula:

$$T = \frac{PD}{2S} = \frac{150 \times 37\frac{1}{2}}{2 \times 2000} = 1.4 \text{ in.}$$

in which

D = Diameter of cylinder in inches;
P = Pressure in pounds per sq.in.;
S = 2000 lb.; safe working strength of cast iron;
T = Thickness of walls of cylinder in inches.

A safety factor of 10 and 20,000 lb. as the ultimate tensile strength of cast iron were used.

Substituting in the formula I found that with a pressure of 150 lb. it would require a wall thickness of 1.4 in., while after the cylinder was bored out I would have only a ⅞-in. wall, which was good for about 93 lb. pressure. As the bore would also come close to the cylinder-head studs, and the valve chambers needed reboring, also as the cylinder was

slightly cracked in the ports, I decided that the old cylinder could not be rebored and be safe, so a new cylinder was ordered immediately.

A. RAUCH.

Pittsburgh, Penn.

Piping Details and Drafting Problem

My attention was called to this short-cut method described below, when I expressed surprise at the time lost by some draftsmen in laying out power-plant piping.

Given three views of two parallel pipes, *A* and *C*, connected by a third pipe, *B*. To determine the true angle of the fittings connecting *A* and *B* and *B* and *C*. When the pipes *A* and C are in the same plane as their projection, the projected angle will be the true angle. Therefore (see Fig. 2), when line *ab* is revolved to the new position *ab'*, the point *b* will be on the line *C*, where *b'* meets the line *C* when *ab'* is revolved.

These problems may seem very elemental as they are, but they take less time than the lengthier process as described in some books on descriptive geometry.

H. W. TERRY.

Chicago, Ill.

Silica in Feed Water

Frequently it is stated that water with less than ten grains of hard matter to the gallon is considered good for steam-generating purposes, but this is very misleading in some cases.

It depends altogether upon what the nature of the hard matter is. I have used water with only 4½ grains to the gallon that proved very bad for boilers, especially in forming scale, but it contained 1.65 grains of silica to the gallon, which is one of the worst minerals we have to contend with. It does not make a very thick scale, but it is so hard that it does not have to get very thick to be exceedingly dangerous, especially when it gathers around the end of the tubes at the back of the boiler. Scale will form the heaviest near the tube ends where they go through the head and it does not take long for it to bake hard enough to allow the tube end to be badly burned. This will cause more leaky tubes than anything else, for as soon as the tubes burn, the scale falls off and the water strikes the tube and shrinks it; then it starts leaking.

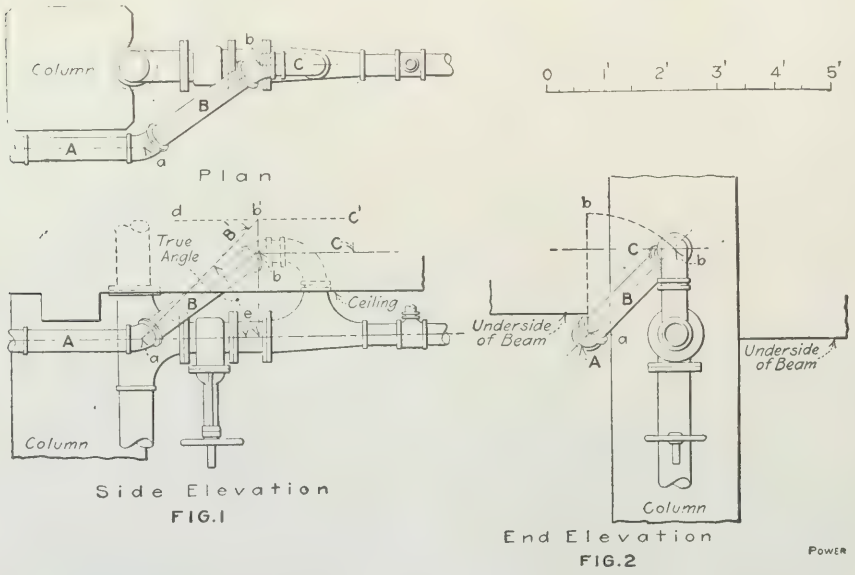
When I took charge of the plant this water was used. The manager said that he had not been able to get an engineer to stay more than a month, and this was evidently true from the list of names covering a period of two years. If an engineer did not quit he was discharged and if he was not discharged he quit anyhow. When the boilers began leaking in the daytime the night man blamed the day man for letting the water get low and burning the boilers, and when they began leaking at night, the day man would say the same thing about the night man.

After I had been there a few days and explained to the manager how the scale formed, he saw that the water was more to blame than the men. We then decided to try some boiler compound, which we did without getting good results.

Relative to compounds I would state that I regard as useless those that separate the foreign matter from the water in the boiler, as it throws down the impurities on the boilers, and the trouble continues just the same. The water should be treated before going into the boilers; not after entering them. The best thing we used in this case was coal oil, half a gallon every 24 hr., per 100 hp., feeding it through the suction pipe of one pump, using a sight-feed drip valve to get a uniform feed. I do not contend that this will apply to all cases, but we got good results. One must experiment and watch results to find the best remedy.

W. A. LEE.

Sorrento, La.



SHOWING MANNER OF LAYING OUT PIPING DETAIL

side elevation will give the true angles. The line *ab* in revolving about the point *a* will generate a cone. Therefore, in Fig. 1, *a* will remain fixed and *b* will always be on a perpendicular through its original position. Therefore, *b'* is the new position of *b* and the angle *eab'* is the true angle of the fittings.

To calculate this angle we either know or can determine the length of *B* in Fig. 2. We also know or can determine the length of *ae*, Fig. 1.

Tangent of the angle $eab' = \frac{b'c}{ae} = \frac{B}{ae}$.

Angle $db'a = eab'$

because *A* and *C* are parallel.

We also can start with the assumption that *A* and *B* are connected by a 45-deg. fitting and that *ABC* is the elevation. Fig. 1, with $eab' = 45$ deg., $AB'C'$ is the true projection or is the side elevation of the revolved position of *ab*. In Fig. 2, the

Correcting Trouble of Pilot Piston

While I was on duty on a coastwise steamer, we fitted a new steam chest to the independent vacuum pump, but when we turned steam on we found that the pilot piston struck the heads of its chamber, instead of cushioning as it should. Upon taking the heads of the pilot-piston chamber off we discovered that the small check valves in the steam ports did not seat and permitted the steam to exhaust through the admission port, which left no steam in the chamber for the pilot piston to cushion on. After repairing the check valves (in this case they were simply brass flaps over the admission port) the pump ran noiselessly.

ALFORD H. SMITH.

Rogers, Ark.

Sand Cuts Pump Rods and Packing

While working in a certain plant, we experienced much trouble from the cutting out of the packing in our boiler-feed pumps, the plungers of which were outside packed. We tried flax, canvas and a few well known pump packings, but it got so they would cut out in a few days. Determined to find the cause, I removed the air chamber and cover plate from the water end and discovered a large quantity of sand between the valve decks, which sand had been drawn from the surge tank.

To offset this trouble from sand, the suction pipe was extended up through the bottom of the tank for about 1 ft. Every day the sand was blown out from a blow-down pipe in the bottom of the tank, at one end. No further trouble was experienced with the packing cutting out.

C. E. SKELLEY.

Chicago, Ill.

Dry Air Pump as Air Compressor

The use of compressed air in most modern power houses for cleaning has become almost a necessity. The wiring and contacts on a remote control switchboard cannot be effectively rid of accumulations of dust and dirt by any other means, and to attempt to wipe them off, while in operation, would not only be futile, but dangerous. The hand bellows, besides not giving sufficient blast, is an unhandy, not to say impossible thing with which to clean circuit-breakers, oil switches, and other adjuncts of a modern board, which, in plants running 24 hr., must be cleaned while in operation.

Compressed air is not always available at power houses, for consulting engineers, as a rule, do not yet appreciate that a small air compressor is a very desirable addition to the power-plant equipment.

In one plant, where the dust problem is serious, the engineer obtained air by temporarily converting the dry-vacuum

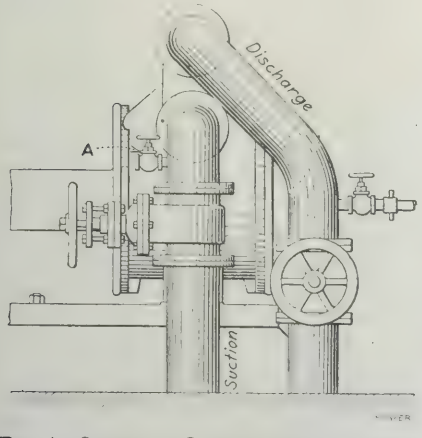


FIG. 1. SHOWING CONNECTIONS FOR CONVERTING AIR PUMP INTO AIR COMPRESSOR

pump into an air compressor. Fig. 1 shows how simply this was done. Both the suction and discharge pipes have gate valves in the positions shown. Above the valves, the pipe was drilled and tapped for 1¼-in. connections, putting a valve on each as shown.

When it is desired to use it as an air compressor, both large valves are closed, and the air is taken in at *A*, and the hose is connected at the small valve on the discharge line. This gives us a blast of dry air sufficiently strong to dislodge any dirt that may collect. As there are duplicate units, one machine is usually available, although recently, while both were in operation, the pump was cut off from the condenser for a few minutes and used for cleaning, the vacuum meanwhile falling, slowly, from 28 in. to 10

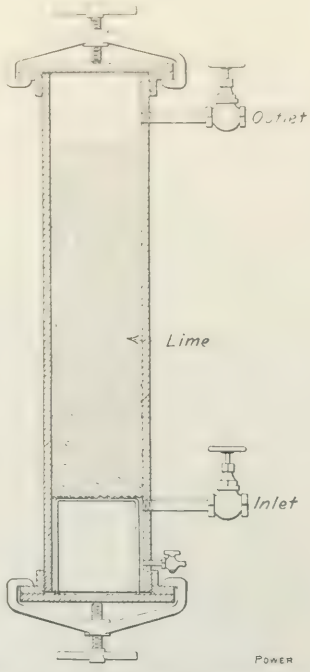


FIG. 2. DRYER FOR ABSORBING MOISTURE FROM AIR

in., when the blowing was stopped, the large valves were opened, and the vacuum again raised to normal. The valve *A* is also used as a vacuum breaker when the main units are operating condensing.

When using compressed air for cleaning electrical apparatus, one should be sure the air is dry, and that no slugs of water can be picked up anywhere in the line and carried along. Such a happening might result in serious damage. When the air is taken from a general line supplying air for industrial purposes, an air dryer or moisture collector should always be used. A form of dryer which is easily constructed, and will give good results, is shown in Fig. 2. A piece of large diameter wrought-iron pipe, preferably 12 in., and about 4 to 6 ft. long, is drilled and tapped at each end, as shown, for ¾-in. pipe (which size will give ample blast for any purpose); a bolted flange or a clamp may be used to close the ends of the 12-in. pipe. At the lower end of the pipe, a piece of heavy screen should be placed about 6 in. from the bottom. The pipe should be filled to within a foot of the top with lime, and the flange clamped on tight. This will absorb all the moisture and leave the air dry.

In blowing out apparatus, it is well to remember that damage can be done by the careless use of an air blast; insulation can be loosened, and the wires in small resistance coils might easily be disarranged and possibly short circuited resulting in much trouble.

Finally, the hose used should not be wire-wound, or terminate in a metal nozzle or coupling.

S. M. QUINN.

Hudson, N. Y.

Queer Feat of Cylinder Head

Some years ago I was employed on a side-wheel steamboat of the beam-engine type. We had been taken off our run to make some repairs, of which the most important was reboring the cylinder. It was 63 in. in diameter and 11-ft. long.

We had opened and washed the boilers, and the drips to the cylinder were also opened. Lifting the cylinder head was done by putting the crank on the lower center and then lashing the wheels so the engine could not turn either way. By using large tackles from the crosshead to the cylinder head the latter was lifted. We had started to lift the head and before we had much more than got a strain on the tackles, the cylinder head jumped up about 3 in. and then dropped back onto the cylinder. There was a flat rubber gasket on the cylinder head.

Those who have had experiences on engines like this will remember that the cylinder head is very heavy and not easy to lift. I would like to have readers give their opinion as to why the head jumped off in this manner.

CLARENCE R. DAVIS.

East Boston, Mass.

Boiler Explosion in New Zealand

I notice in POWER many accounts of boiler explosions. Boiler explosions here in New Zealand are practically unknown. In this country every boiler must be examined by an inspector once a year. If the boiler is passed, he gives the owner a certificate, which shows the amount of pressure the boiler is allowed to carry. This certificate must be placed in the boiler room. Every boiler over 15 hp. and also engine with cylinders of 144 circ.in. up to 200 circ.in. must be in charge of a second-class certified engineer. Every engine with a cylinder of 200 circ.in. or over, with boiler attached must be in charge of a first-class certified engineer.

Any person found tampering with the safety valve to make the boiler carry more pressure than allowed by the certificate, is liable to lose his certificate or to a heavy fine. The inspectors are appointed by the government.

FREDERICK W. KIRBY.

Auckland, N. Z.

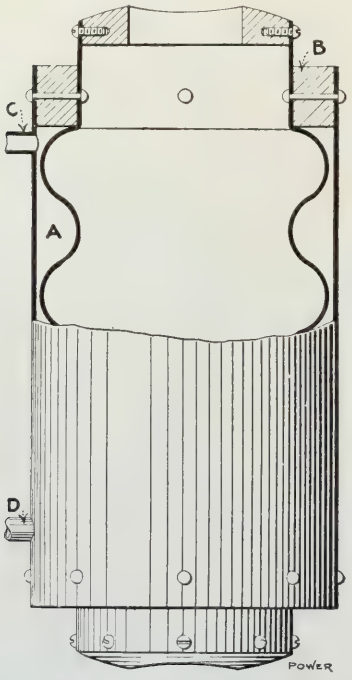
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Design of High-Pressure Tank

On re-reading the issue of Apr. 9, I was interested in the article "Exploding Tank Wrecks Building," page 506. It is a good description of what a faulty rendering tank can do in the way of destruction. I have made, patented and used for many months a tank unique for its simplicity of design and ability to withstand hard usage.

This rendering tank consists of an outer and inner shell, the latter being corrugated as shown; the steam space is at *A*. Between the shells at each end is placed a ring *B* to which both shells



SHOWING SECTION OF NEW DESIGN RENDERING TANK

are riveted. The steam inlet is shown at *C* and the outlet at *D*. The inner shell has a manhole at each end.

The corrugated shell is first made by welding the edges instead of riveting them, which gives full efficiency to the joint. Then the corrugations are rolled at white heat, and the strength exceeds many times the pressure required for rendering. The surface is free from such nuisance as stay-bolts, seams, or places where the animal acid can attack.

The usual size jacketed tank is about 9 ft. high by 30 in. internal diameter and has stay-bolts every 6 in. of inside surface, or about 225 bolts in all, any one of which may begin to leak after three years' steady use. This is due to cold heading of the stays.

H. F. BREHM.

Waterloo, N. Y.

Graphite in Boilers

I read with much interest your editorial in the issue of Sept. 17 entitled "Graphite in Boilers," as yours is the first real mention that has come to my notice on this subject.

We have been using graphite in our boilers for the past 18 months with very gratifying results. We started its use by putting about two quarts of the commercial flake graphite into each boiler when it was cleaned. While this was successful in loosening a small amount of scale, it was rather slow. Consequently, about Nov. 1, 1911, we commenced using a regular boiler graphite.

This finely powdered graphite was introduced into the boiler water in the following manner: We tapped the boiler feed line on the pump side, screwed in a short length of ¾-in. pipe fitted with a globe valve and a large (one pint) oil cup. The cup is filled with graphite, the valve opened and the pump does the rest. As soon as all the graphite has been drawn into the line, the valve is again closed.

We have four 250-hp. water-tube boilers operating at 125 lb. pressure. Three of them are in constant operation with the fourth for reserve. These boilers were second-hand when installed by us and contained about ⅛ in. of hard scale. Immediately after cleaning each boiler, we throw into it about two quarts, or, say 3 lb. of graphite. Then we daily introduce with the feed water about one pint of graphite to each 100 hp. The boilers are blown down twice daily, the morning shift blowing from the bottom and the afternoon shift blowing from the top.

At present almost all the old scale has come off, while that remaining is loose and in a short time will all come off and the boilers will be perfectly clean. After the boiler has become clean the graphite circulates with the water against the metal to which it imparts a graphite finish or polish.

The action of graphite in neutralizing scale-forming contents, or in loosening old scale is, I consider, purely a mechanical one. It prevents the scale-forming content of the water from adhering to the metal by mixing with it and forming a sludge which drops to the bottom of the boiler and is easily washed out. As to its action on the old scale, the finely powdered graphite works its way into all the small cracks and interstices of the scale, weakens it, and destroys its adhesion to the metal.

While the action of the graphite is, perhaps, somewhat slower than that of chemical compounds, yet this disadvantage is more than counteracted in its following advantages:

Its action being a mechanical one it can be used successfully with any feed water, i.e., it is not necessary to have a compound made up for each kind of feed water. Being insoluble it cannot pass out with the steam and thereby unfit it for certain industrial purposes, as for instance, in breweries, laundries, ice plants, etc., as is frequently the case where compounds are used. It acts mechanically and not chemically and can in no way injure the boiler. It is also cheap, costing only about 3c. per 100 hp. per day.

J. M. BURNS.

Dover, N. J.

While working as boiler washer in a plant containing six 250-hp. water-tube boilers, we washed out and removed all the scale from the tubes and drums six weeks. After the tubes and drums had been cleaned and the tube caps replaced, about 5 lb. of flake graphite was placed in each drum, the manhole plates replaced, the boiler filled, tested and put in service.

When a boiler was opened for cleaning and the cleaner run through the tubes, large pieces of scale would come off, leaving the metal clean. Upon examining pieces of scale, the flakes of graphite could be plainly seen on the under side. This prevented it adhering tightly to the metal. The same conditions prevailed in the drums.

I cannot speak on graphite scale-preventing qualities, but I do know that it makes much easier the removal of scale already formed.

FRED L. WAGNER.

Chicago, Ill.

We have a plant of four water-tube boilers; one 200 hp. and three 250 hp. each. We were using lake water which makes

a hard scale, and tried numerous compounds with various degrees of success; finally we tried graphite compound. It prevented the scale getting hard, loosened the old scale and left the boilers perfectly clean.

But our water bill was excessive so the firm laid a line to a small stream near the plant and pumped the water into a storage tank. Then our troubles began; the water was surface water and the greater part of it came from a large brick yard.

The boilers began to scale heavily. We increased the graphite and it had no effect unless to make the scale harder, for it was the hardest scale I ever saw; it would take from three to five days to clean a boiler with a turbine cleaner.

Then we began to lose tubes—as many as 15 tubes in one month. This was too expensive, so we had a sample of the water analyzed and a compound made to counteract the scale-forming properties. It is giving excellent results as we have not lost one tube since we began using the compound. But I have tried the same compound in other plants with poor results. So I find it is with graphite as with other compounds, what will give good results in one plant may be detrimental to another.

WILLIAM J. AU.

Cleveland, Ohio.

Zinc as Corrosion Preventive

Mr. Stafford in the Oct. 1 issue asks how zinc slabs should be used to prevent corrosion.

While employed as a boiler inspector I made a trip through Oklahoma in the place of the district inspector. At one plant I met with a very enthusiastic reception. The previous inspector had advised the use of zinc slabs suspended in the boiler to counteract the pitting effect of sulphuric acid in the feed water which was partly mine water. The engineer at the plant religously followed the inspector's advice as to how the slabs should be suspended from the through rods below the tubes (the boilers were of the horizontal tubular type), however, he neglected to comply with the additional instructions to place a pan made of sheet iron underneath the suspended zinc.

The plan worked well; the acid attacked the zinc with a vengeance, the result being that the zinc disintegrated and fell to the bottom of the boiler in a pile and soon after there was a bag in the fire sheet, which came down like an elongated cocoanut, necessitating a patch about 12x20 in., right where a patch should not be. Naturally, the superintendent felt like welcoming boiler inspectors in the true Western fashion of welcoming horse thieves.

In Mr. Stafford's case he proposes to treat a marine boiler in which the danger of burning a sheet would be practically nil, provided he does not suspend the zinc directly over the furnace. The corrosion in his case is most likely caused by the returns from the heating system, which become so purified by continued condensation that they have a corrosive action.

A. C. TERLENE.

Chattanooga, Tenn.

Relative to Mr. Stafford's inquiry concerning the placing of zinc slabs in boilers to prevent pitting, it does not make so much difference how the zinc is suspended so long as it does its work—whether hung from a hook or otherwise I have found the best way, for general convenience in cleaning out and making renewals, to be to construct a pan of ⅛-in. iron, any size to suit the conditions and shape of the braces, and suspend it from the braces so it will clear the top row of tubes by 2 or 3 in. Turn up the edges of the pan not less than 2 in. and drill a number of small holes in the bottom. In a few weeks, depending on the acidity of the water, the zinc will have pulverized and become like gravel—depending on the amount of dross in it.

If the back end of tubes and shell show most pitting, hang the basket or pan at that end; if the front end is most affected, put it there.

H. A. WYCKOFF.

Brooklyn, N. Y.

The following is the method in which zinc is used to prevent pitting of the boilers aboard Uncle Sam's battleships:

An iron box is made of ¼-in. material, 12 in. wide by 40 in. long, and 13 in. deep. Large holes are cut in the sides and bottom, and the top is left open. This is suspended from the shell of the steam drum and comes just below the water line. In this box are placed 18 slabs of zinc, each 12x12x½ in.—set six in a row. The zinc weighs 446 lb. per cu.ft., or about 18.5 lb. to each slab, making a total of about 333 lb. in each boiler. These slabs are set on edge in the box with holes drilled in them and through the box, through which ½-in. rods are run with large nuts or washers between the slabs to hold them in place and about 1¼ in. apart. The rods have a split pin through the end to keep them in place. This arrangement exposes the greatest surface of zinc to the action of the water which passes through the holes in the box and around the plates of zinc. The action of the water on these slabs causes them to waste away, but when they are taken out there is a coating of scale on them which often entirely fills up the space between the slabs. This scale is easily cracked off when dry, and leaves the slabs about one-quarter or one-half their original thickness. These slabs, if too thin, are not used again.

The water used in the boilers of these ships, although coming from the evaporators, is more or less salty and contains chemicals that cause galvanic action and pitting. When zinc is used the galvanic action takes place on the zinc instead of on the boiler, as zinc has a stronger attraction for the acids than the iron.

In one battleship there are 16 water-tube boilers with a total of 29,000 hp., or 1812 hp. each. Each boiler has one box of these zinc plates. I should judge that 10 to 15 lb. of zinc slabs suspended in the drum just below the water line, or about 7 lb. per 100 boiler-hp. would be sufficient for a stationary boiler.

J. C. HAWKINS.

Hyattsville, Md.

I would advise Mr. Stafford that zinc will prevent incrustation as well as corrosion in boilers. The scale may acquire thickness and hardness, but can easily be removed from the plates. It is supposed that the zinc in connection with the iron plates keeps up a feeble galvanic action and that the hydrogen liberated at the surface of the plate by this action prevents the incrustation from adhering to it.

The zinc should be thoroughly distributed throughout the boiler in the form of slabs. About 1 sq.in. of zinc surface should be supplied for every 50 lb. of water.

FRANK BYUS.

Norborne, Mo.

Pitting is perhaps the worst kind of boiler corrosion. Pit holes range from the size of a pin head to the size of a dime or more and are conical or cup-shaped. The depth of these pits may be anything from a slight depression to a hole cut through the shell or tube. They are not found in any fixed place, although they are more commonly found on the tubes, furnaces and the combustion chambers of Scotch boilers than any other place. They are generally filled with a black pasty mass, which is chiefly iron oxide, lime salts and oily matter.

To account for the formation of these pits, various explanations have been suggested, most of them claiming electrochemical action as the cause. It is claimed that all substances are at a different electrical potential or pressure. This difference is found not only between substances of different kinds, but also between similar substances at different temperatures or in different physical conditions, as, for example, between two pieces of iron or steel, one of which has been hammered or worked more than the other. Due to this difference of potential there is a tendency to set up a flow of electric current from one to the other, and to change the two substances so as to bring them into electrical equilibrium or balance.

These chemical changes may be much accelerated or retarded by the medium in which the bodies are immersed. If they are in dry air, for instance, no activity takes place, and the difference of electrical conditions remains unchanged. If they are immersed in water, or especially in salt or slightly acid water, the operation will usually be greatly assisted by the medium.

In a boiler the continued electro-chemical action occurs for a variety of reasons. Parts of the boiler of widely differing temperatures or of different physical or chemical compositions may provide the elements which are at different electrical potential. Such a difference is likely to be found between iron and its oxides, especially the magnetic oxide or mill scale or between a particle of carbon in the steel and the surrounding metal, or between a place in the steel where the proportion of carbon is much greater than the average or in the surrounding metal, or between a bit of slag or other impurity in wrought iron and the adjacent metal.

What is needed is a substance which will be electro-positive to iron and readily attacked, so that the wasting away will be diverted from the iron to this protecting substance, and the operation will proceed at the expense of this substance instead of the boiler. Such a substance is zinc, and its use for this purpose is beneficial.

The zinc must be in actual metallic contact with the boiler. The best zinc to use is the rolled variety of close grain and uniform structure, free from impurities. I have found that the number of zinc slabs 12x8x⅝ in. required for the thorough preservation of boilers is about one for every 50 sq.ft. of tube surface, that is approximately equal to

$$\frac{N \times D \times L}{197}$$

where

N = Number of tubes plus the number of solid stays;

L = Length between the tube plates or sheets in feet;

D = External diameter of the tube in inches.

This method is used by the U. S. Government. Another rule often used is 12 lb. of zinc to every 25 indicated horsepower.

Care should be taken in placing the zincs so that there are enough near the water line and the combustion-chamber tops and about one-sixth of the total number should be placed in the steam space.

Upon examining the zincs in the boiler after they have been in use for some time, it will be noticed that the wear or action on the zinc is entirely confined to its surface.

C. E. ANDERSON.

Chicago, Ill.

Distribution of Power in the United States

The tables from the U. S. census report on manufacturing industries submitted by Mr. Tuck and published in the Oct. 15 issue are interesting, but it should be noted that they give only the distributed horsepower in the manufacturing plants and that two important branches, electric-railway power stations and central power plants, are not included.

POWER readers may be interested in more complete information on this subject, which goes a step further than the tables submitted by Mr. Tuck. My composite table necessitated going through many government reports and other sources of information and the main table gives 35 different items under each state.

Inasmuch as the census reports on central stations and electric railways were issued in 1902 and in 1907, while the census reports on manufacturing industries were issued in 1904 and 1909, it was found necessary to group the 1902 and 1904 reports and compare them with the 1907 and 1909. As most readers are interested mainly in the amount of power generated in steam-power plants, I only include the summary covering that branch, omitting plants using water power, gas engines and purchased electrical power.

This summary shows that in 1902 and 1904, 80 per cent. of the horsepower of the steam-power plants was installed in private plants in the manufacturing industries, while in 1907 and 1909, only 74 per cent. of the total amount of power was generated in these plants. This corroborates Mr. Tuck's statement that manufacturers are buying more power than formerly and that the central stations are becoming more important. Central-station activity is further apparent by considering the third column of Table 1, which shows that during five years central stations increased their power installations by 92 per cent. while manufacturing plants increased only 32.4 per cent.

Some of the information developed is extremely interesting and has shown where to look for new business. For instance, in the five years covered by the report, the table shows only a 15 per cent. increase in the total amount of steam power-plant apparatus installed in Massachusetts, while Wisconsin has a 129 per cent. increase. Many other surprises were brought out that show where in it was necessary to revise some of our selling plans in order to take advantage of natural tendencies.

While, as Mr. Tuck points out, there has been quite an increase in the gas-engine business in some parts of the country, this item is not of so much importance when it is reduced to its proper proportion, as will be made clear by referring to Table 2.

Table 2 shows that, while in the five years covered by the tables the steam horsepower has increased only 42.5 per cent. and the gas-engine horsepower 173 per cent., when figured out on the proportion that each of these items is of the total, as given in the last column, only 3.5 per cent. of the total horsepower in the United States is in gas-engine installations.

GEORGE H. GIBSON.

New York City.

TABLE 1.—SUMMARY OF STEAM POWER

	Hp 1902–04	Hp. 1907–09	Per cent. inclusive in 5 yr.	Per cent. of total 1902–4	Per cent. of total 1907–09
Manufacturing	10,813,044	14,202,142	31.4	80.2	74.0
Light and power central stations	1,369,155	2,627,450	92.0	10.2	13.7
Electric railway power stations	1,298,133	2,368,183	82.5	9.6	12.3
Totals	13,480,332	19,197,775	42.5	100.0	100.0

TABLE 2.—SUMMARY OF TOTAL POWER

	1902–04	1907–09	Per cent. increase in 5 yr.	Per cent. that increase is of total inc.	1902–04	Per cent. distribution 1907–09
Steam horsepower	13,480,332	19,197,775	42 5	77.7	84.6	82.5
Number of units	135,533	163,199	21	65.4	75.4	73.6
Gas-engine horsepower	303,529	826,246	173	7.1	1.9	3.5
Number of units	21,695	34,856	61	31.1	12.1	15.6
Water horsepower	2,135,505	3,263,832	53	15.2	13.5	14.0
Number of units	22,451	24,038	7	3.5	12.5	10.8
Total horsepower	15,919,366	23,287,853	46	100.0	100.0	100.0
Total Number of units	179,769	222,093	24	100.0	100.0	100.0

The first steel produced in this country was probably made in Connecticut in 1728 by Samuel Higley and Joseph Dewey, according to the U. S. Geological Survey.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Varying Voltage of Converters

What commercial methods are employed for varying the direct-current voltage of a rotary converter?

T. W.

Using (1) an alternating-current synchronous booster, (2) an alternating-current potential regulator, (3) a direct-current booster, (4) a split-pole converter, or (5) by automatic compounding.

Idler Pulleys

Are idler pulleys which run on the opposite side of the belt from the transmission pulleys, usually made with straight or crown face, and why?

W. M. F.

Idler pulleys regardless of the side of belt on which they are used are generally made crown faced to urge the belt to travel symmetrically with the idler. When a tightener-idler is used for increasing the arc of contact of a belt on a transmission pulley the face of the idler is generally less crowned than the crowning commonly given to faces of transmission pulleys.

Preventing Motor Runaway

What prevents a shunt-wound motor from running away when the load is suddenly taken off?

J. B.

The armature winding, in cutting the magnetic lines of the field, generates an electromotive force opposite to that which forces the current through the armature. As the motor tends to speed up, the counter electromotive force tends to increase and thus automatically prevents a rush of current through the armature. In this way the speed is kept down to a value very little in excess of that at full load.

Cracks in Boiler Side Walls

Can the side walls of ordinary return-tubular boiler settings be constructed so they will not crack, and if not, what precautions can be taken in their construction to minimize their cracking?

A. H.

As the linings of a boiler setting are subjected to high temperatures, cracks in the side walls resulting from expansion of the brickwork are practically unavoidable. Side walls which remain freest from through cracks are those which are carefully laid up with an air space and ample expansion space, filled with asbestos or mineral wool, between the boiler and the brickwork, except where supporting side-brackets rest on the walls.

Stop Diagram

What is a stop indicator diagram and why is one taken from a gas engine?

D. R.

A stop diagram is one in which the recording pencil of the indicator by stopping at the limit of its travel, is thereby prevented from indicating cylinder pressures which exceed the limit of pressure so recorded. Stop indicator diagrams are taken of gas engines by using a weak indicator spring to obtain a diagram which will more clearly show events during the exhaust and suction strokes in a four-stroke-cycle gas engine regardless of greater pressures which are beyond the range of measurement by the indicator spring thus used.

Formula for Shafting

What is the rule for estimating the safe number of horsepower transmitted by iron or steel shafting when properly lined up and supported?

A. F.

The safe-transmitting capacity of a $1\frac{15}{16}$- or nominal 2-in. diameter shaft, with the usual bending strains of pulleys, belts, etc., is 10 hp. at 100 r.p.m. The torsional resistances of shafts are as the cubes of their diameters, and the number of horsepower transmitted by any given size is directly in proportion to its speed of rotation. Hence for finding the power capable of being transmitted by any size shaft at any number of revolutions per minute: Multiply the cube of its diameter in inches by the revolutions per minute and by 10 and divide by the cube of 2, multiplied by 100; or expressed in a formula wherein

d = Diameter of shaft;
n = Number of revolutions per minute;

$$\frac{d^3 \times n \times 10}{2 \times 2 \times 2 \times 100} \text{ or } \frac{d^3 n}{80} = horsepower$$

As an example, the safe number or horsepower which a 3-in. shaft, running at 200 r.p.m., could transmit would be

$$\frac{(3 \times 3 \times 3) \times 200}{80} = 67.5\ hp.$$

Small Loss of Head

What pressure per square inch would be lost in overcoming pipe friction to pump water at the rate of 500 gal. per hr. through a 6-in. pipe, 2500 ft. long?

A. J. H.

The cross-sectional area of a 6-in. diameter pipe being 28.274 sq.in., and as 500 gal. would be $500 \times 231 = 115{,}500$ cu.in., the velocity of flow in the pipe would be

$$\frac{115{,}500}{28.274 \times 60 \times 60 \times 12} = 0.094 + ft.\ per\ sec.$$

or practically only about 0.1 ft. per sec. By Darcy's formula the loss of head from friction would be, in feet,

$$h = \left(0.0199 - \frac{0.00167}{d}\right) \times \frac{l}{d} \times \frac{v^2}{2g}$$

in which d, l and v are respectively the diameter and length of pipe and the velocity of flow per second all given in feet, and g is the acceleration of gravity (equal to 32.162 ft. per sec.) Substituting in the formula the values

$l = 2500$ ft.; $d = 0.5$ ft.;
$v = 0.1$ ft.; $2g = 64.324$;

$$h = \left(0.0199 - \frac{0.00167}{0.5}\right) \times \frac{2500}{0.5} \times \frac{0.1 \times 0.1}{64.324}$$

$$\text{or, } h = 0.02324 \times \frac{25}{32.162} = 0.018\ ft.$$

is the head lost in friction, which would amount to $0.018 \times 0.433 = 0.0078$ lb. per square inch, showing that in pumping through a 6-in. pipe, at the rate of flow stated, the loss of pressure from friction would be practically zero.

Raising Sheet Iron Stack

What is the best method of raising or lowering a sheet-iron smoke-stack 60 ft. high and 3 ft. in diameter?

L. C. F.

The best method is to use a block and fall and gin pole. It will generally be safer to erect or lower such a stack in two or more sections of length, for which purpose each section should have lashed to it a number of equally spaced staves for receiving a rope sling for suspending the weight of the section a little above its center of gravity. Suspended in this manner the position of a section is readily controlled and the staves will prevent crushing of the section at its point of suspension. An illustrated article describing the method of erecting a 90-ft. stack appeared in the Mar. 8, 1910, issue of POWER, pages 435-437.

Engineers' Study Course—III

Multiplying the numerator multiplies the fraction.

It is easy to see that five of the subdivisions, or $\frac{5}{12}$ of the circle in Fig. 1 is five times as much as one subdivision, or that

$$\frac{5}{12} = 5 \times \frac{1}{12}$$

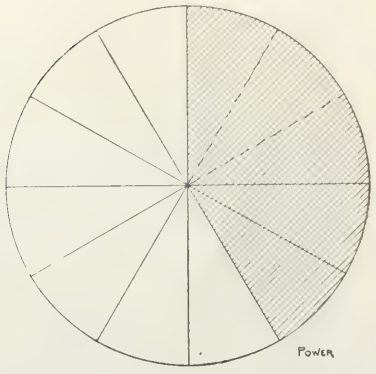


FIG. 1

EXAMPLE: The diameter *AB*, Fig. 3, is 24 in. Along this diameter are 12 holes, each ¾ of an inch in diameter. How much solid plate is there left on the section *AB*?

$$12 \times \tfrac{3}{4} = \frac{12 \times 3}{4} = \tfrac{36}{4} = 9 \text{ in.}$$

Applying what we have just learned about cancellation, we can do this still more easily, as follows:

$$\frac{\overset{3}{\cancel{12}} \times 3}{\cancel{4}} = 9$$

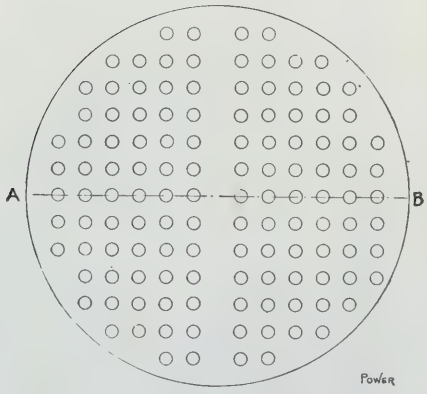


FIG. 3

If 9 in. are taken up by the tubes, there will be

$$24 - 9 = 15 \text{ in.}$$

of solid plate.

EXAMPLE: The area of the circular head, of which *AB*, Fig. 3, is the diameter, is $3\frac{14}{100}$ sq.ft. There are in it 125 tubes, each having a cross-sectional area of $\frac{11}{25}$ of a square inch. How many square inches of head remain?

The area occupied by the tubes is

$$125 \times \tfrac{11}{25} = \frac{125 \times 11}{25} = \tfrac{1375}{25} = 55 \text{ sq.in.}$$

The area of the head is $3\frac{14}{100}$ sq.ft. The difference between $3\frac{14}{100}$ sq.ft. and 55 sq.in. will be the area of the head.

But one cannot subtract square inches from square feet directly.

$$88 \text{ sq.ft.} - 16 \text{ sq. in.} = 72 \text{ what?}$$

Before quantities can be added or subtracted they must be reduced to the same terms. Feet must be subtracted from feet, inches from inches, etc. You cannot take six barrels of pears from 12 barrels of apples and have six barrels of something left, but you can take six barrels of fruit from 12 barrels of fruit and have six barrels of fruit left, because you have reduced it to fruit, a term common to both kinds of article involved.

In 1 sq.ft. there are 144 sq.in.

$$3\tfrac{14}{100} \text{ sq.ft.} = \tfrac{314}{100} \text{ sq.ft.}$$

In $3\frac{14}{100}$ sq.ft. there are

$$144 \times 3\tfrac{14}{100} = 144 \times \tfrac{314}{100} = \frac{144 \times 314}{100}$$

$$= 452\tfrac{16}{100} \text{ sq.in.}$$

From this we can now take the 55 sq.in. occupied by the tubes, leaving

$$452\tfrac{16}{100} - 55 = 397\tfrac{16}{100} \text{ sq in.}$$

in the rest of the head.

Get that idea clearly that things must be reduced to the same *denomination*, before you can add or subtract them. We shall need to use it again in the next lesson.

Dividing the denominator multiplies the fraction.

Notice that? Dividing the denominator does the same thing to a fraction that multiplying its numerator does.

The value of a fraction is the quotient found by dividing the numerator by the denominator. The smaller the divisor or denominator the greater the quotient. If the divisor or denominator is only one-half as great, the quotient or value of the fraction will be twice as large. If the denominator is only one-third as great, the value will be three times as large, etc.

$$\tfrac{1}{8} = 2 \text{ but } \tfrac{1}{4} = 4 \text{ (twice as much)}$$

$$\tfrac{6}{15} = 4 \text{ but } \tfrac{6}{5} = 12 \text{ (3 times as much)}$$

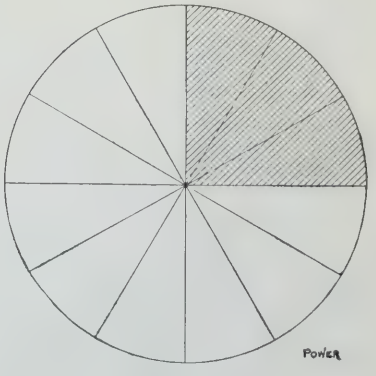


FIG. 4

In Fig. 4 the circle is divided by the horizontal and vertical lines into four equal parts. Each of these is divided into three equal parts. If we write $\frac{3}{12}$, it will denote only the shaded part of the circle, i.e., three of the smaller divisions. (These *smaller* divisions are denoted by the *larger* number 12, because the *more* portions we divide the circle into the *less* each portion becomes.) But if we write ¾ it will denote the larger unshaded part of the circle three times as large as the shaded portion. The numerator is the same in both cases, but in one case, we have three of the smaller divisions, $\frac{3}{12}$ths, and in the latter case three of the larger divisions, ¼ths.

Dividing the numerator divides the fraction.

One-half of eight apples is four apples. It is equally easy to see that one-half of eight "ninths" or $\frac{8}{9}$ is four "ninths" or $\frac{4}{9}$.

Multiplying the denominator divides the fraction.

Applying the same reasoning that we used to show that dividing the denominator multiplied the fraction, it is easy to see that multiplying the denominator divides the fraction. A look at your rule will show that $\frac{3}{16}$ is only one-half as much as ⅜. By multiplying the denominator by 2 you have divided the value of the fraction by 2.

In some examples previously given we multiplied fractions by integers or whole numbers. Suppose we want to find what three-quarters, or ¾ of 12 is.

One-quarter of 12 is $\frac{12}{4} = 3$, then three-quarters of 12 are $3 \times 3 = 9$, or doing the whole thing together

$$\frac{3 \times 12}{4} = 9$$

We have *divided* the quantity which was to be multiplied, by the denominator of the fraction and *multiplied* by the numerator.

Now suppose we wanted to find ¾ of $\frac{9}{16}$, that is, to multiply $\frac{9}{16}$ by ¾. We simply do the same thing. We *divide* $\frac{9}{16}$, the quantity to be multiplied, by the denominator of the fraction by which we want to multiply it, by multiplying its denominator by the denominator of the multiplier, and then we multiply it by multiplying its numerator by the numerator of the multiplier.

$$\tfrac{9}{16} \times \tfrac{3}{4} = \frac{9 \times 3}{16 \times 4} = \tfrac{27}{64}$$

The rule for multiplying fractions turns out to be very simple. I could have

given it to you without so much explanation, but I want you to understand the reason for it, as these reasons will be of constant use in understanding what is coming.

To Multiply Fractions

Rule: *Multiply all the numerators together for the new numerator.*

Multiply all the denominators together for a new denominator.

Example: An engine developing 24⅔ hp. takes 5⅙ lb. of coal per hour per horsepower. How much coal would it take to run it three-quarters of an hour?

$$24\tfrac{2}{3}\ hp. = \frac{3 \times 24 + 2}{3} = \tfrac{74}{3}\ hp.$$

$$5\tfrac{1}{6}\ lb. = \frac{6 \times 5 + 1}{6} = \tfrac{31}{6}\ lb.$$

$$\frac{74}{3} \times \frac{31}{6} \times \frac{3}{4} = \frac{74 \times 31 \times 3}{3 \times 6 \times 4} = \frac{1147}{12} = 95\tfrac{7}{12}$$

Suppose we wanted to *divide* 12 by ¾.

Division is just the opposite of multiplication. When we *multiplied* 12 by ¾ we divided it by 4 (the denominator) and multiplied by 3 (the numerator).

To *divide* it by ¾ we multiply it by the denominator and divide by the numerator.

This simply means that you tip the divisor fraction upside down, and multiply the quantity to be divided by its denominator, and divide it by its numerator.

$$12 \div \tfrac{3}{4} = 12 \times \tfrac{4}{3} = \frac{12 \times 4}{3} = 16$$

$$\tfrac{11}{16} \div \tfrac{3}{8} = \tfrac{11}{16} \times \tfrac{8}{3} = \tfrac{88}{48} = 1\tfrac{40}{48}$$

When the quantity to be divided is a whole number it may be treated as an improper fraction, the denominator of which is 1, so that we could say

$$\tfrac{12}{1} \div \tfrac{3}{4} = \tfrac{12}{1} \times \tfrac{4}{3} = 16$$

To Divide by a Fraction

Rule: *Invert the divisor and proceed as in multiplication.*

Example: A stuffing-box is 4½ in. deep. How many rings of packing, each ¾ in. wide, will it take to fill it?

$$4\tfrac{1}{2}\ in. = \frac{2 \times 4 + 1}{2} = \tfrac{9}{2}\ in.$$

$$\tfrac{9}{2} \div \tfrac{3}{4} = \tfrac{9}{2} \times \tfrac{4}{3} = \tfrac{36}{6} = 6\ rings$$

I told you that a simpler way to indicate division than by the use of the division sign ÷ is to write the dividend above the line and the divisor below. This applies to fractions as well as to whole numbers.

Instead of $\frac{9}{2} \div \frac{3}{4}$, we may write $\frac{\frac{9}{2}}{\frac{3}{4}}$.

When you have a fraction above the line drop its denominator below the line with a multiplication sign. This will simplify matters and will not change the value of the expression

$$\tfrac{3}{4} \div 2 = \tfrac{3}{8} \qquad \frac{\frac{3}{4}}{2} = \frac{3}{2 \times 4} = \tfrac{3}{8}$$

When you have a fraction below the line, change its denominator to a multiplier above the line. For instance, in the above example about packing we could write

$$\frac{\frac{9}{2}}{\frac{3}{4}} = \frac{9 \times 4}{2 \times 3} = \frac{36}{6} = 6$$

This amounts to the same thing as the rule given but is simpler. Remember, that it is always the denominator whose position is changed and that it becomes a multiplier in both cases.

Let us try one final example involving both multiplication and division.

Example: A boiler evaporates 9¼ lb. of water per pound of coal and the engine uses 25⅛ lb. of steam per hour per horsepower. How many pounds of coal will it take to run the engine for a week of 5½ days of 9¾ hours each with a load of 246⅔ hp.?

The engine will take

25⅛ × 246⅔ lb. of steam per hour

25⅛ × 246⅔ × 9¾ lb. of steam per day

25⅛ × 246⅔ × 9¾ × 5½ lb. of steam per week

If 1 lb. of coal makes 9¼ lb. of steam, it will take as many pounds of coal to run the engine a week as 9¼ is contained times in the above product; so that the whole problem reduces to

$$\frac{25\tfrac{1}{8} \times 246\tfrac{2}{3} \times 9\tfrac{3}{4} \times 5\tfrac{1}{2}}{9\tfrac{1}{4}}$$

Reduce these mixed numbers all to improper fractions

$$25\tfrac{1}{8} = \frac{8 \times 25 + 1}{8} = \tfrac{201}{8}$$

$$246\tfrac{2}{3} = \frac{3 \times 246 + 2}{3} = \tfrac{740}{3}$$

$$9\tfrac{3}{4} = \frac{4 \times 9 + 3}{4} = \tfrac{39}{4}$$

$$5\tfrac{1}{2} = \frac{2 \times 5 + 1}{2} = \tfrac{11}{2}$$

$$9\tfrac{1}{4} = \frac{4 \times 9 + 1}{4} = \tfrac{37}{4}$$

Substituting these figures in the above statement of the problem, we get

$$\frac{\frac{201}{8} \times \frac{740}{3} \times \frac{39}{4} \times \frac{11}{2}}{\frac{37}{4}}$$

Dropping the *denominators* which are above the line to positions under it, and transferring that *denominator* which is below the line above it we get

$$\frac{201 \times 740 \times 39 \times 11 \times 4}{8 \times 3 \times 4 \times 2 \times 37} = 67 \times 20 \times 13 \times 2 = 34{,}840\ lb.$$

Answers to Last Week's Questions

1. $\frac{96}{144} = \frac{8}{12} = \frac{2}{3}$.
2. $\frac{2301}{1728} = \frac{77}{576}$.
3. $\frac{307}{2156} = \frac{1}{8}$.
4. $\frac{(64 - 12) \times 40}{2240} = \frac{52 \times 40}{2240} = 1\frac{3}{14}$.
5. 1.
6. $\frac{1}{3}$.
7. $\frac{1}{12} = \frac{2}{24}$.
8. $\frac{720}{1440} = \frac{1}{2}$, $\frac{699}{1398} = \frac{1}{2}$, $\frac{5439}{10878} = \frac{1}{2}$.
9. $\frac{1}{2}$.
10. $\frac{1}{2}$.
11. $\frac{77}{154} = \frac{1}{2}$.
12. $\frac{1365}{2730} = \frac{273}{546} = \frac{91}{182} = \frac{1}{2}$.
13. $\frac{275 \times 90}{33{,}000} = \frac{24750}{33000} = \frac{825}{1100} = \frac{75}{100} = \frac{3}{4}$.

Examples for Practice

1. What is the weight of 2712-in. of 4-in. wrought-iron pipe if it weighs $10^{33}/_{50}$ lb. per ft.?

2. A 225-hp. engine consumes 3⅝ lb. of steam per hr. per hp. How many pounds will it consume if run $2^{4}/_{5}$ days of 10 hr. each?

3. A boiler grate contains $56^{23}/_{29}$ sq.ft. of surface. If $8^{4}/_{9}$ lb. of coal can be burned per square foot in 1 hr., how many pounds can be burned in 9⅞ hr.?

(4) An engine that has been running $2^{2}/_{5}$ hr. has been running how many times longer than one that has been running $\frac{6}{10}$ hr.?

(5) How many tubes does a boiler contain if its total heating surface is 2133⅓ sq.ft. and each tube has $^{64}/_{3}$ sq.ft. of surface?

(6) How many washers each $^{3}/_{64}$ in. thick can be strung on a bolt that is $^{9}/_{8}$ in. long from inside of the head to inside of the nut?

American Coal Abroad

The value of the U. S. coal sent to foreign countries in the last fiscal year was $52,500,000 against $21,000,000 in 1902 and $8,500,000 in 1892, having thus increased over 500 per cent. in the last 20 years and 150 per cent. in the last decade. Even these large figures of coal sent to foreign countries in the fiscal year 1912 do not include the value of that passing out of the country in the form of "bunker" or fuel coal laden on vessels engaged in the foreign trade, which aggregated nearly $23,000,000 in value, making a total of over $75,000,000.

The quantity sent to foreign countries in 1912 was, according to figures compiled by the Department of Commerce and Labor, 17,500,000 tons. The quantity exported in 1912 is over six times as much as in 1892.

Coke exports also show a decided growth, the value in 1892 having been $112,000 and in 1912 $3,000,000.

The fact that the coal sent to foreign countries has increased 150 per cent. both in quantity and value, during the last 10 years, and that the total value of exports to foreign countries plus the value of that leaving the country as bunker coal now aggregates over $75,000,000 suggests that the total value of the coal passing out of the United States in a single year will soon reach the $100,000,000 line.

Steam Engine Breakdown Lessons

By H. S. Knowlton

A number of steam engine failures; an analysis of the causes responsible for them; and recommendations showing how they could have been prevented by a little forethought.

The following notes, condensed from the records of a large accident-insurance company, review the more important facts bearing upon various steam-engine failures occurring during the past year, and suggest in a number of cases, defects in design or operation which should be avoided in future work.

Water in the Cylinder

The bottom of the low-pressure cylinder of a vertical compound engine was fractured, probably through the entrance of water. As shown in Fig. 1, the cylinder exhausted through a 6.5-ft. length of pipe into a jet condenser. After an inspection of the air pump the engineer opened the injection valve *C*, and then the engine

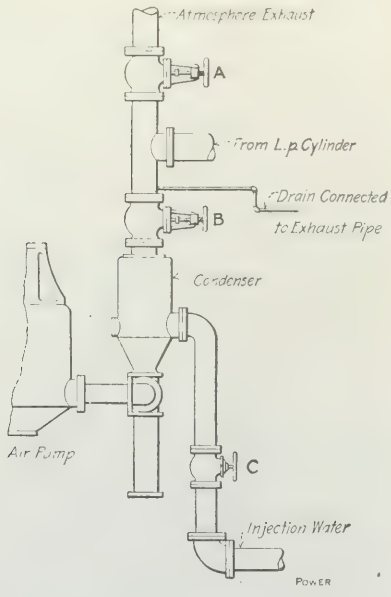


Fig. 1

stop valve, slowly starting the engine. He then opened the valve *B* and shut the atmospheric valve *A*, and at once the low-pressure cylinder bottom was fractured circumferentially at the stuffing-box and at the inner edge of the flange. The latter was also cracked radially through 17 of the 18 bolt-holes, and the center boss of the piston was torn out.

A careful examination of the broken parts failed to disclose any indication of the presence of any foreign substance between the piston and the cylinder bottom, or any abrasion of the cylinder bore. It was difficult to learn how the water reached the low-pressure cylinder, but it was reasonably certain that it did not come through the main steam lines, since these were carefully designed, falling to a separator near the engine, which had a glass water gage and trap drain. It was thought that the water might have come from the atmospheric exhaust pipe or from the condenser. In the former case, as the engine had been running noncondensing for a considerable period, water must have accumulated in the lower part of the exhaust pipe above the valve *B*, and may have risen to overflow into the valve chest.

As long as the engine ran slowly the water trapped in the cylinder at the beginning of the compression would have been forced through the drain *E* into the exhaust pipe and back into the steam chest, but when the speed increased, the drain no longer was sufficient to handle the water.

A drain just above the valve *B*, to be opened when working noncondensing, would have prevented an accident of this kind. It also seemed possible that the trouble may have been due to the improper handling of the valves by which water from the injection pipe rushed into the condenser and upward into the low-pressure cylinder.

Turbine Accident

An accident to a turbine emphasized the importance of automatic drainage of steam piping. In this case the turbine was of the exhaust type, connected to the low-pressure cylinders of an engine plant of about 1000-hp. rating, with a supplementary steam connection through which

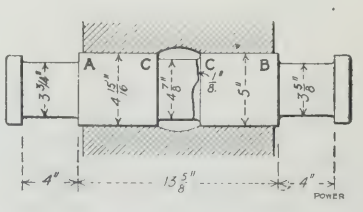


Fig. 2

boiler steam reduced in pressure could be supplied when the load upon the engines was too small to furnish the turbine with the necessary quantity of exhaust. The turbine ran at 3000 r.p.m., and exhausted into a surface condenser with the circulating and air pumps driven by a small compound engine. When operating with most of the equipment shut down, the 4½-in. blades of the thirteenth runner were all stripped, and many other blades were bent. The corresponding rows of fixed blades were also damaged, and several of the condenser tubes immediately below the steam inlet were considerably damaged.

The most probable cause of the accident appeared to be entrained water coming from the steam pipe. This pipe was 10-in. in diameter and 5 ft. long, and being uncovered served to condense much of the steam. It was undrained except by a tap, which was never opened when the turbine was working. There appeared to be no reason why water could not readily accumulate in this pipe and back up at the end next the turbine until carried forward in a mass by the steam current. After the accident the pipe was replaced

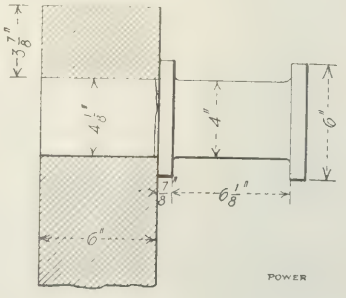


Fig. 3

by another connected with a well drained automatically by a trap.

Beam Engine Failure

A failure in a beam-engine gudgeon occurred recently, illustrating the importance of looking after the wear and strength of all moving parts when modernizing an old prime mover. The engine had been rebuilt sufficiently to permit raising the boiler pressure to 130 lb., and the load upon the piston was increased to 50,000 lb. Unfortunately, the gudgeon in the beam was not replaced, although attention was drawn to it by rust working out between the gudgeon and the boss. It was found to be loose and when taken out, was seen to be broken, as shown in the illustration.

Measurements showed that both the gudgeon and the inside surface of the boss holding it were worn about $^{1}/_{64}$ in. at *A* and *B*, Fig. 2, so that the gudgeon, when free from the strain was gripped only by the beam at *CC*, and could receive no support from the outer parts of the boss unless strained about $^{1}/_{64}$ in. The gudgeon was thus in the position of a beam supported at the journals and loaded at two points each about 1⅝ in. from the center. Under such conditions, the load of 50,000 lb. upon the piston would at times produce a stress of about 16,000 lb. per sq.in. at the top and bottom surface of the gudgeon.

Crankpin Failure

A crankpin failure resulted from the use of a poor and rather antiquated design. The engine was of the horizontal tandem-compound type running at 74 r.p.m. under 30 lb. steam pressure. The pin was of wrought iron of the dimensions shown in Fig. 3 without a fillet at the back of the collar, and it was shrunk into a wrought-iron crank. The strain produced by the grip of the crank, combined with the abrupt change of sectional area, started a crack close behind the collar which gradually extended until the pin severed more than half through and gave way. The load upon the pin was about 20,000 lb., giving a bending stress of about 1500 lb. per sq.in.

It was estimated by the engineer of the accident-insurance company that the

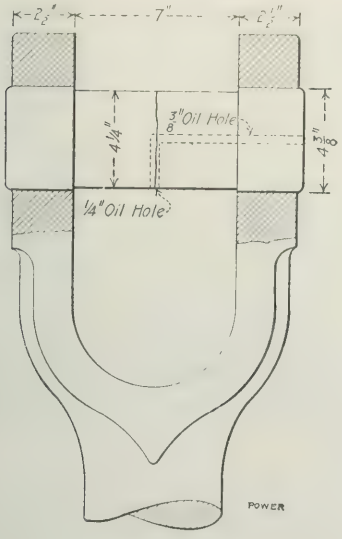


Fig. 4

shearing stress due to the shrinkage of the crank was probably sufficient to compress the shaft where the crank gripped it, and thus tear it away from the collar. The point was emphasized that where shrinkage is employed, the diameter of that part of the internal piece upon which the external part is to bear should be slightly larger than the diameter of the adjacent parts, and that the part of enlarged diameter should be entirely covered by the external piece.

Fracture of Crosshead Pin

An accident to a crosshead occurred in a 21x48-in. vertical engine running at 76 r.p.m. The pin coupling the piston-rod crosshead to the connecting-rod was of steel, and was fixed in the forked end of the latter, as shown by Fig. 4. The fracture lay entirely in a plane at right angles to the axis of the pin and had apparently started at the oil hole and worked its way through. It appeared that if the ¼-in. hole had coincided with a horizontal instead of a vertical diameter, and cut through the least instead of the most highly stressed fibers of the pin, it might have done no harm, and probably would have lubricated the surface fully as well.

Connecting-rod Failures

Cross-bending stress produced by the centrifugal tendency of the connecting-rod and cap appeared to be responsible for the failure of a 7½x6-in. vertical,

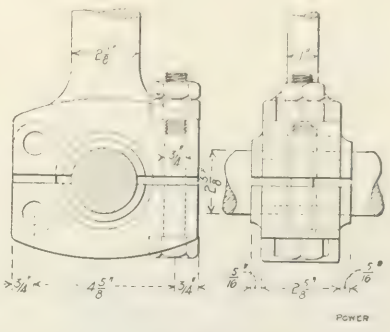


Fig. 5

single-acting, noncondensing engine running at 465 r.p.m. The connecting-rods were 15 in. long and weighed about 40 lb., the larger ends being made as shown in Fig. 5. The caps were held by a link hinge on one side of the bearing and a ¾-in. bolt on the other. The bolt in one of the rods broke at the bottom of the first thread, and the failure of the rod end followed. The bending stress upon the bolt was estimated at 6100 lb. per sq.in., and it was believed that the engine had made about 200,000,000 revolutions since the bolt was new. The significant point was the magnitude of the cross-bending stresses to which the connecting-rod bolts of rapidly running engines may be subjected.

A connecting-rod fracture occurred in the operation of a horizontal, noncondensing engine running at 92 r.p.m., under a steam pressure of 40 lb. The strap at the crosshead end of the connecting-rod broke through the bend, as shown in Fig. 6, resulting in a broken cylinder

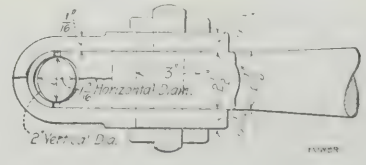


Fig. 6

cover and bedplate and a bent connecting-rod. An examination showed that the crosshead pin was oval, its vertical and horizontal diameters measuring 2 in. and 1⅞ in. respectively, and it was found that new brasses had been fitted about seven weeks before the breakdown.

From measurements of these after the breakdown it appeared that they had been bored to the horizontal diameter of the pin, and when first put in must have touched the pin only on the top and bottom. Thus fitted, they became powerful wedges, and under the action of the piston a fracture of the strap naturally followed. This fracture was not new, but it was uncertain whether it began before or after the new brasses had been placed in service. The pin was loose in the crosshead, and revolved in it freely instead of in the brasses; otherwise overheating would have been experienced with the brasses wedged at the top and bottom between the pin and the strap.

Breakage of Air Pump Crosshead

Fig. 7 shows the details of an air-pump crosshead connection which failed on account of the wear on the journals to which the links from connecting bell-crank levers were coupled, the air pump being driven from the piston-rod crosshead by links and bell cranks. One of the air-pump crosshead arms broke close to the central socket into which the rod was cottered. The original diameter of the journals had been 2 in., but when the crosshead failed, they measured 1¼ in. vertically and 1⅝ in. horizontally, as

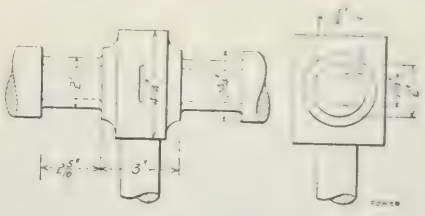


Fig. 7

indicated in the illustration. Not only was the crosshead weak, but the stress upon it was greater than under ordinary conditions for the reason that the overflow pipe from the hotwell was not provided with a continuous drop to its outlet, but dipped downward to clear a thoroughfare, later rising again and permitting the accumulation of a column of water which had to be accelerated at every stroke of the air pump. Furthermore, the space for the cooling water was so limited that the temperature of injection was high, necessitating the pumping of an unusually large volume of water for this service.

This point had been emphasized a few years before, when the engine was insured, and it had been suggested that the engine should discharge into a tank, a centrifugal pump being employed to lift the water into a pond or cooling tower. As the journals were never stripped when the insurance inspector visited the plant, there was no opportunity to discover their condition, and knocking had been prevented by packing the brasses with leather, the owners being unwilling to purchase a new crosshead.

"It'll be a cold day when you get me to work again!" remarked the radiator to the reducing valve.

Fuess Steam Meter

Steam which meets obstruction in a pipe line will suffer a certain loss in pressure. If the obstruction is permanent and not changeable, a certain loss in pressure will correspond with a certain velocity of steam. It is therefore possible to measure the velocity of steam by means of reducing the pressure if the relationship between these two factors is known. In the steam meter made by R. Fuess, of Germany, who is represented in this country by H. Bacharach, of Pittsburgh, this physical process was adopted.

The reduction of the inside pipe section is accomplished by a throttling disk or a plate, with an opening smaller in diameter than the pipe, located between two flanges. The steam at this point loses part of its pressure, but the shape of the throttling disk and the reduction in the pipe section is so calculated that the loss in pressure at the highest steam velocity will not exceed 1½ lb.

To measure the loss in pressure a mercury differential manometer is used. It consists of a straight glass tube, the lower end of which is connected with a receptacle of a large surface. By connecting the tube of the mercury manometer, Fig. 1, with the steam line in front of the throttling disk and the receptacle of the manometer with the steam line back of the disk by means of small tubes filled with water, the difference of pressure in front and back of the throttling disk

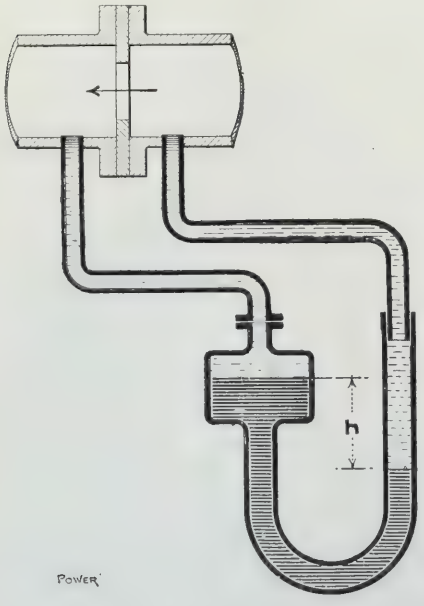


FIG. 1. ILLUSTRATING PRINCIPLE OF OPERATION

will effect a lowering of the mercury in the manometer tube. The difference in the height, h, gives the means to ascertain the loss in pressure.

Through research work it was found that the quantity of the steam G has the following relation to the loss in pressure h.

$$G = 151.2\ K q \sqrt{\frac{p \times h}{p_0 \times S}}$$

where

G = Quantity of steam;

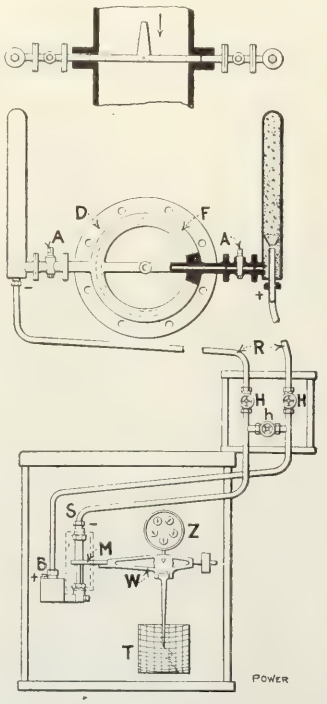


FIG. 2. SKETCH OF METER AND CONNECTIONS

K = Constant depending on the size of the opening in the throttling disk;

q = Cross-section of throttling disk;

p = Steam pressure in front of throttling disk;

p_0 = Steam pressure back of throttling disk;

$h = (p - p_0)$ = Difference in pressure measured at the mercury manometer;

S = Specific gravity.

The quotient $\frac{p}{p_0}$ given under the square root sign is always close to 1 on account of the difference between p and p_0 never exceeding 1½ lb. The specific gravity of the steam, while changeable with increase or decrease of pressure and, when using superheated steam, with the increase and decrease of temperature, will not change so much as to have a noticeable influence on the measuring of steam. Constant conditions of steam to secure regular and satisfactory operation may usually be expected in a certain plant, so that it is permissable to use for S a certain standard factor corresponding with the standard steam conditions and insert this factor in the above formula. Besides G and h all other values in the formula are unchangeable for a certain steam line. They can, therefore, be combined for this steam line to a constant K, thus obtaining the following simplified formula:

$$G = K \sqrt{h}$$

This constant K is sent with each steam meter and it is only necessary to extract the square root of the reading taken on the manometer board and multiply it by K, to obtain the amount of steam flowing through the line expressed in pounds per hour. The coefficient K holds good only for a certain condition of steam or at least for one which varies only in very narrow limits. In cases where the steam line has, at a certain time, a pressure of, say, 140 lb., and then of, say, 70 lb., a different constant K has to be used for each pressure. It is therefore possible to apply the meter to different steam lines by using a different constant for each one.

For a continuous record, the amount of steam flowing is recorded automatically on a chart. Wherever it is intended to find the total amount of steam used during a specified period a counter has to be used.

A sketch of the instrument as actually installed is shown in Fig. 2. A throttling disk F is inserted between two flanges of the steam line D. Across the disk is a bridge which has one opening in the center at the front and one at the rear. The two openings do not communicate with each other but are carried through the disk separately and connect to the cocks A and A. From each of these cocks a line R leads to the valves H

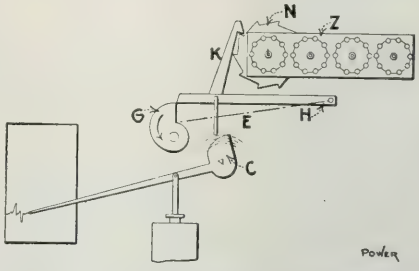


FIG. 3. MECHANISM OF COUNTER

which, in turn, are connected to the mercury manometer at B and S. The narrow leg of the manometer is made of glass and the wide one of iron. The line is filled with water which reaches down to the mercury in the manometer. A small cylinder floats on top of the mercury in the narrow leg and is embedded in glass to prevent corrosion. This small piece of iron is the core of a permanent magnet which is connected with the balance lever W. The latter carries a pen which gives the record on drum T.

The valves H shut off the meter while valve h equalizes the pressure in both legs and adjusts the recording pen to the zero position.

To avoid planimetering the daily charts, a counter Z is connected to the apparatus which integrates automatically the differ-

ent amounts recorded on the diagram. The construction and working of this apparatus, shown in Fig. 3, is as follows:

A small clock movement releases every minute another clock movement which again releases, through cam *G*, the lever *H*. The latter actuates, by means of the ratchet *K*, the counter *NZ*. The upper stop of lever *H* is fixed, the lower depends upon the position of the metal piece *C* which is connected to the balance lever *W*, Fig. 2. The shape of this piece depends upon the velocity of the steam

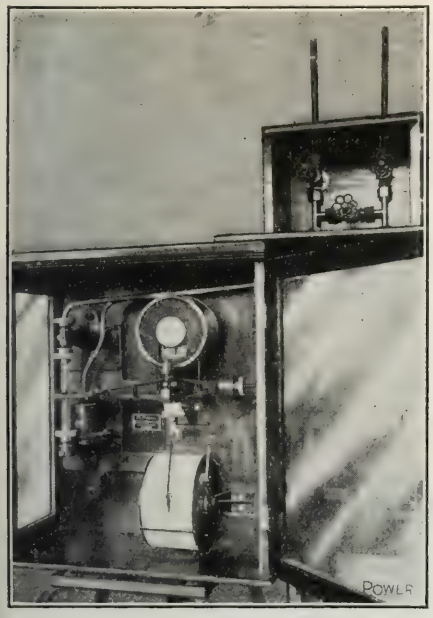


FIG. 4. A PHOTOGRAPH OF THE METER

and the area of the main line. High velocities cause large up and down movements of *E* corresponding to a large stroke of the counter; a low velocity, on the other hand, causes small up and down movements and a corresponding smaller record of the counter. This makes it possible for the reading of the counter to be taken in cubic feet as in a gas meter.

The correctness of the record depends, in the first place, upon the constant weight of the steam to be measured. But since this weight varies with varying temperature and pressure, the apparatus can only give exact readings for one certain temperature and pressure. For large variations, the readings must be corrected correspondingly.

Boiler Burned Out by Neglect

About two-thirds of the plant of the New England Pin Co., Winsted, Conn., was shut down recently owing to the burning out of a boiler. The man in charge at night had opened the blowoff valve and when it was closed some obstruction lodged on the seat which allowed the boiler to drain dry.

An hour's good firing failing to show pressure on the gage, the man asked the aid of a brother watchman in an adjoining plant and his investigation showed a dry and very hot boiler. The fire was covered very heavily and everything done to cool the boiler off. Then, acting as far as his conscience would allow, he advised the man to report immediately to his chief.

The boiler inspectors found all tubes needing expanding and a girth seam leaking very badly. The heat which showed on the shell after the fire was covered was from red to white.

Boiler Explosion in Heating Plant

The boiler in the heating plant of the Salem Bank & Trust Co.'s Bldg., Salem, Ore., exploded on Oct. 27. One man was killed and two were injured. One of the injured was hurled through a heavy plate glass window, landing in a hole several feet below the sidewalk that had been made by the force of the impact of the explosion. The damage to the building and heating system will amount to several thousand dollars, nearly every plate glass window in the building being blown out.

The boiler was of the cast-steel sectional type made by the American Boiler Co., and had been in service for the Oregon State Insane Asylum, and was considered in first-class condition. It is the opinion of local engineers that had the heating system been in charge of a competent engineer, the accident would never have occurred.

No Power Show for 1913

Many conditions influenced the National Exhibitors' Association of Commercial Engineers against holding a power show in 1913. The Textile Exhibitors' Association will not hold an exhibition of textile machinery such as was held last April, and with which the power show was allied. Then, again, the National Exhibitors' Association is to hold its annual exhibition of power-plant apparatus, machinery and supplies at Springfield in September, 1913, at the same time as will be held the annual convention of the National Association of Stationary Engineers; the Universal Craftsmen, Council of Engineers, holds its convention in Boston in August, 1913, and it may be that an exhibition will be held in conjunction therewith, and the Massachusetts State Association of the N. A. S. E. convention and exhibition is to be held in Lawrence during July, 1913. All these matters being taken into consideration it was the general opinion of the members that the holding of a power show would not be advisable.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Yes, it's Wilson, a good brand. Never mind now how we voted, it's up to us to dig in all the harder and look pleasant. Then these good old United States will continue to be God's country.

An Oklahoman complains that his town's electric-lighting service is too darn good; his fowls don't know when it's dark and won't go to roost. We have in mind several old roosters who are similarly afflicted. Sounds like a fowl joke!

Mayme Pixley, a young Indiana woman, is getting all the big contracts in her home state to paint the lighting-plant and distillery smoke-stacks. "My Pappy got hurt, and I took up his trade." Hooray for Mayme!

"Be proud of your calling," advises *Valve World*, and it quotes a speaker as saying: "I am not a banker or a broker; I'm just a plumber." Great Crœsus! If some of us engineers were plumbers, we'd be hiring bankers and brokers to take care of our money.

HORAN—Why do they be sayin' "the unspeakable Turk," Terry?

DORAN—That's aisy. Ye can't pronounce th' names they has. The half ov thim looks loike the chimical symbol for Good's sassyparilly.

Hi Sibley has a blast in the *American Machinist* anent the factory-whistle nuisance. He suggests a calliope playing topical songs to cheer the ear-weary employee. We joyously trotted behind the circus calliope in our infantile days, but, now—good night!

The government fed the engineers on the Panama Canal $111,000 worth of ice cream last year. It kept one plant continuously busy.

One of the moving-picture fraternity has bought two switching locomotives and will sic one on the other in a "head-on" collision, replacing the engineer with dummies after the engines are under headway. The "villain" (another dummy) is to be ground into goulash and presumably not enough of the old pushers will be left to furnish kindling to light a gas stove. Now let's have a "fillum" that makes a hero of the operating engineer; then there should be action aplenty.

A. S. M. E. Meeting in Germany

The plans for the meeting of the American Society of Mechanical Engineers with the Verein deutscher Ingenieure, in 1913, which promises to be the most remarkable tour of the industries in Germany ever offered to American engineers, are being most satisfactorily arranged. Conrad Matschoss, dozent of the Royal Polytechnic High School of Berlin, has been visiting the United States as a representative of the German society and has been conferring with the committee of arrangements regarding itinerary professional papers, etc.

June 11 will be the date of departure. The party will arrive in Germany on Saturday, June 21, be received by the municipal authorities and given an opportunity to visit the great shipyards before going on by special train to Leipzig the same or the following day.

Sunday evening, June 22, there will be an informal gathering to promote acquaintanceship before the formal opening on Monday, June 23. At this opening the visitors will be welcomed by the King of Saxony and the municipal authorities of Leipzig, and two addresses, one by a representative of the German and the other by a representative of the American society which will be on general subjects, as the history of engineering, relation of capital and labor, and effect on industry of technical education.

Tuesday, the last day to be spent in Leipzig, will be devoted to technical papers, to various social events and participating in the patriotic gathering in commemoration of the one hundredth anniversary of the battle of Leipzig.

On Wednesday the party will leave for a tour of industrial Germany, visiting the principal cities, Dresden, Berlin, Düsseldorf, Cologne, Frankfurt, Nürnberg, Munich, etc. In all probability, the Krupp Works will be thrown open to the party, the Baroness von Bohlen (formerly Miss Krupp) receiving the visiting American engineers in person. After the trip up the Rhine from Bonn to Rüdesheim, there will be a grand festival on the Rüdesheim embankments. Independence Day will be celebrated at Frankfurt under the auspices of the American Embassy.

The final meeting will be held in Munich, July 7, in connection with the Museum of Technical Arts, of which Dr. von Miller is the director. It is expected that the Prince of Bavaria and the Mayor of Munich will also receive the party. The entire trip through Germany and the various official receptions are being arranged by a committee of prominent men of the arts and sciences in the German society and much interest is being shown by municipal authorities and the heads of industrial establishments.

The itinerary of the visit of the American society will be as follows:

June 21. Reception at the landing-place by the local board.

June 22. Leipzig, official reception by the Verein deutscher Ingenieure.

June 23, 24. Participation in the exercises of the annual meeting.

June 25. Dresden.

June 27, 28. Berlin.

June 29. Düsseldorf. Reception by the Rhine-Westphalian Section of the Verein.

June 30 to July 2. Various functions arranged by the Rhine-Westphalian Section, and trip to Cologne and Essen.

July 3. By railway to Bonn. By boat up the Rhine, with festival at Rüdesheim. By rail to Frankfurt, arriving about 6 p.m. Evening, reception of welcome at Frankfurt.

July 4. Independence Day at Frankfurt.

July 5. Nürnberg.

July 6. Munich. Evening, reception of welcome.

July 7. Morning, visit to the German Museum. Evening, trip on the Starnberg Lake. Final exercises in the Rathaus.

Heating Engineers to Meet at Indianapolis

The executive committee of the National District Heating Association met in its mid-year session in the city of Indianapolis, Ind., on Oct. 30 and 31, to arrange the program for the fifth annual convention and to fix the time and place for holding that convention in 1913. Indianapolis, Ind., was unanimously selected and the time set for May 27, 28 and 29, with headquarters at the Claypool Hotel. Papers upon the following subjects were arranged for the convention: "District Heating on the Pacific Coast," "Possibilities of Hot Water Heating," "Line Condensation in Different Forms of Construction," "Results of Using Bleeder Turbines," "Appraisal of Heating Properties," "Station Designs with Special Reference to Overload Capacities," "Uses and Abuses of Public Service Commissions," "Hot Water Heating for Residence Districts," "Public Policy;" also two other papers, the subjects to be announced later. In addition, reports were asked for from the standing committee on meters, station records and rates, the report of the committee on meters to call for experience of those stations that have changed over from flat rate to meter rate.

The largest screw power-testing machine in the world, capable of carrying 1,000,000 lb. on two screws, has just been shipped by the Riehlé Bros. Testing Machine Co., Philadelphia, Penn., to the American Steel Foundries, Alliance, Ohio.

Educational Work of Newark Association No. 3

Newark (N. J.) Association No. 3 has begun its winter educational classes in subjects that apply to power-plant operation. The course will include a review of essential mathematics; thermodynamics, as heat losses in engine and boiler practice, and means for diminishing them; furnace efficiency; economic combustion of fuel; draft; flue-gas analysis, etc. The engineering problems will embrace the efficiency of boiler joints, allowable and safe working pressures, heating surfaces, and horsepower of boilers, and the need of greater safety in operation, minimizing the possibilities of boiler explosions, will be brought forward.

The class is under the supervision of Charles H. Bromley, associate editor of POWER. Edward Sears and C. L. Johnson, members of No. 3, are engaged in designing two power plants, the plans and specifications for which will be discussed by all the members.

Ohio Mechanical, Electrical and Steam Engineers' Convention

The twenty-sixth meeting (eleventh annual meeting) of the Ohio Society of Mechanical, Electrical and Steam Engineers will be held, Nov. 21, 22 and 23, at Akron, Ohio, with headquarters at the Hotel Portage. The program is as follows:

THURSDAY. 1-2 p.m.—Registration.

2:30 p.m.—Trolley-car trip to the Wellman-Seaver-Morgan Co., manufacturer of large power machinery.

7:30 p.m.—Informal reception in Hotel Portage, to meet the Akron Branch, N. A. S. E., and the Universal Craftsman Council of Engineers.

8 p.m.—Address of welcome, by F. W. Rockwell, Sr., mayor of Akron; "The City of Opportunity," by S. F. Ziliox, president Akron Chamber of Commerce. Response by E. M. Adams, president of the society.

8:30 p.m.—"A System of Gas Engine Governing," by George S. Cooper, mechanical engineer, Buckeye Engine Co., Salem, Ohio; "The Lentz System Applied to Steam Engines," by Siegfried Rosenzweig, mechanical and consulting engineer, Erie City Iron Works, Erie, Penn.

FRIDAY. 9 a.m.—A trip of inspection through the mammoth plant of the B. F. Goodrich Rubber Co.

1:30 p.m.—Take cars for Barberton, Ohio, and visit the Babcock & Wilcox Co.

6 p.m.—Complimentary dinner at Hotel Portage. Visiting members and ladies will be the guests of the Akron members.

7:30 p.m.—Business session in Hotel Portage. Election of officers.

8 p.m.—"The Manufacture of Tin Plate," by Ed. W. Patton, master mechanic, Follansbee Bros. Co., Follansbee, W. Va.

SATURDAY. 9 a.m.—Trolley-car trip to the new power stations of the Northern Ohio Power Co., at the Gorge.

Ladies' program, Thursday, 2:30 p.m.—Automobile trip through the residential districts of Akron.

8 p.m.—Theater party at Colonial Theater. Vaudeville.

Friday, 9 a.m.—Visits to several Akron schools.

2 p.m.—Automobile ride around the lakes.

6 p.m.—Complimentary dinner.

8 p.m.—Guests at the meeting for paper No. 182.

Saturday, 9 a.m.—Visit to O. C. Barber's farm at Barberton, Ohio.

OBITUARY

REUBEN WELLS

Reuben Wells, a veteran railroad man, one of the ablest locomotive constructors in this country and for 20 years superintendent of the Rogers Locomotive Works, died at his home, 202 Carroll St., Paterson, N. J., on Nov. 8.

Mr. Wells was born in Pennsylvania, Jan. 1, 1829. At seventeen he entered the works of the Reading company as a machinist apprentice. After finishing his trade, Mr. Wells was sent out as an engineer to run a train in the coal regions between Palo Alto and Port Richmond. As was the custom, he was given an engine which no experienced man would run if he could get a better one, but young Wells made as good a record with the discarded engine as did the others with the better ones.

He was with the Reading until July, 1851, when he went to Columbus, Ohio, as engineer on the Columbus & Xenia R.R., remaining until September. He next became connected with the Rushville & Shelbyville R.R., which was 20 miles long and about 30 miles southeast of Indianapolis. This road had a "strap rail" track. The rolling-stock consisted of one locomotive, two passenger coaches and 30 freight cars and Mr. Wells made one round trip a day. In June, 1852, he was master mechanic of the Jefferson & Indianapolis R.R., and later with the Jefferson, Madison & Indianapolis R.R., where he remained for 26 years, and then was superintendent of the Louisville & Nashville R.R.

In 1868, Mr. Wells designed the first locomotive for grade climbing; it was used on incline work at Madison, Ind. Up to that time it required locomotives with four pairs of drivers and two vertical cylinders, with a pinion wheel working into a "rack" between the rails of the track, which was a slow method. Mr. Wells invented a locomotive in July, 1868, with a tank engine and two cylinders, five pairs of coupled drivers, no truck and with side tanks for water; and it remained in service for 20 years. This locomotive was sent to the "Locomotive Museum" of the Purdue University, at Lafayette, Ind.

In 1873, Mr. Wells was appointed a trustee of Purdue University by the governor of Indiana to represent the mechanical interests of the state. In 1878 he went to Kentucky as superintendent of machinery of the Louisville & Nashville R.R. system, in 1885 becoming general manager and in 1886 second assistant to the president.

Mr. Wells once changed the gage of 1762 miles of track within 20 hours, making the change from the regulation 5-ft. gage to 4 ft. 8½ in., the gage of practically all the other roads in the country.

REUBEN WELLS

It was in 1887 that Mr. Wells left the Louisville & Nashville R.R. to accept the position of manager and superintendent of the Rogers Locomotive Works in Paterson.

Throughout his career he maintained the excellent standard he had set for himself, and for 57 years he was constantly at work and rising to higher positions.

Mr. Wells was a devoted family man and always took an active interest in his business. The late Jacob Rogers, principal owner of the Rogers plant, considered Mr. Wells' services invaluable, and always reposed the most explicit confidence in him. Thousands of Paterson mechanics now working elsewhere were under Mr. Wells' supervision, and by all he was held in high esteem.

He is survived by his widow, two daughters, and two step-sons.

SOCIETY NOTES

At the regular November meeting of the Detroit Engineering Society, held in the hall of the Detroit Wheelman's Club on Friday evening, Nov. 15, Henry Hess, of the Libby Glass Co., of Toledo, Ohio, read a paper entitled "Glass." A light buffet lunch was served after the meeting.

Charles L. Huston, of the Lukens Iron & Steel Co., will give an illustrated lecture on "The Manufacture of Steel Boiler Plate" before the American Institute of Steam Boiler Inspectors at the Engineering Societies Building, 29 West Thirty-ninth St., New York, on Friday evening, Nov. 29. As Mr. Huston is one of the foremost authorities on this subject, the lecture should prove both interesting and instructive, and a large attendance is looked for.

PERSONAL

R. F. Pack, general manager of the Minneapolis General Electric Co., has been appointed a member of the Smoke Prevention Committee of the Minneapolis Civic and Commerce Association.

Corneil Ridderhof, formerly general manager and treasurer of the Wilmarth & Morman Co., Grand Rapids, Mich., is now with the Western Gas Engine Corporation, Los Angeles, Calif., as assistant general manager.

Alfred Johnson, chief engineer of a large department-store building in Chicago and prominent in N. A. S. E. affairs, and John W. Lane, editor of the *National Engineer*, left Chicago on Nov. 14 for a visit to the Panama Canal.

C. E. Fairbanks has been promoted to master mechanic for the Gorham Manufacturing Co., at Providence, R. I. He will take charge of all the mechanical equipment and continue in charge of the company's electrical matters.

Gove S. Taylor has accepted the position of special representative in the Pittsburgh and surrounding territory for the Magnolia Metal Co., New York City. Mr. Taylor was for years manager of the Peerless Rubber Manufacturing Co., and is widely known to the mill-supply and power-plant trade.

B. F. Eyer, who founded the department of electrical engineering in the Kansas State Agricultural College, has resigned and will enter commercial life, Jan. 1. Prof. Eyer was a teacher of physics and chemistry in the Topeka high school for five years and was vice-principal in that school when he resigned to come to the Kansas State College. He is a member of the American Institute of Electrical Engineers, the National Electric Light Association, and a past president of the Kansas Electric Light Association.

Moments with the Ad. Editor

Four thousand dollars for one page of advertising in just one issue! That's what one of the best known American weeklies charges for its space.

* * *

"How in the world," you ask "can a manufacturer get back a profit from such an enormous advertising expenditure?"

Well, in the first place, the advertiser has, or believes he has, an article that will give perfect satisfaction to whoever buys it—

So the sales he makes on that $4000 advertisement will make other sales to friends of the "ad-buyers" and bring repeat orders from them, themselves.

In other words, the merit of the goods will back up the ad and will go on making sales to people who first got acquainted with the product through the ad—

All of which means that the $4000 advertisement pays in proportion to the merit there is in the goods advertised—

Or, as we have said before, advertising pays the manufacturer because it pays the reader of the ad to buy the article advertised.

* * *

Now if this is so true of the $4000 ad in the *general* publication, how much more important it is to you readers of "Power."

Because a large per cent. of the articles advertised in general magazines are *luxuries* which pay the buyers only in so far as they satisfy all the requirements expected of that kind of goods—

While in "Power" every article is a useful necessity manufactured for a useful purpose, to help you get better results in your plant.

* * *

Therefore, if it pays the readers of general magazines to read the ads and buy the goods that are protected by advertising, you can see how much more important it is for you to take advantage of these benefits of publicity—

For *you* are buying power plant equipment to help you do more work, to get it out cheaper and more easily and increase your income—while the reader of the general publication is, in many cases, buying some luxury that has no effect on his daily work.

* * *

Are you getting all you can out of the ads in "Power"? You should follow them closely every week and see to it that all the information they contain becomes yours.

Remember, they are put there to pay you—and if they *don't* then everybody loses.

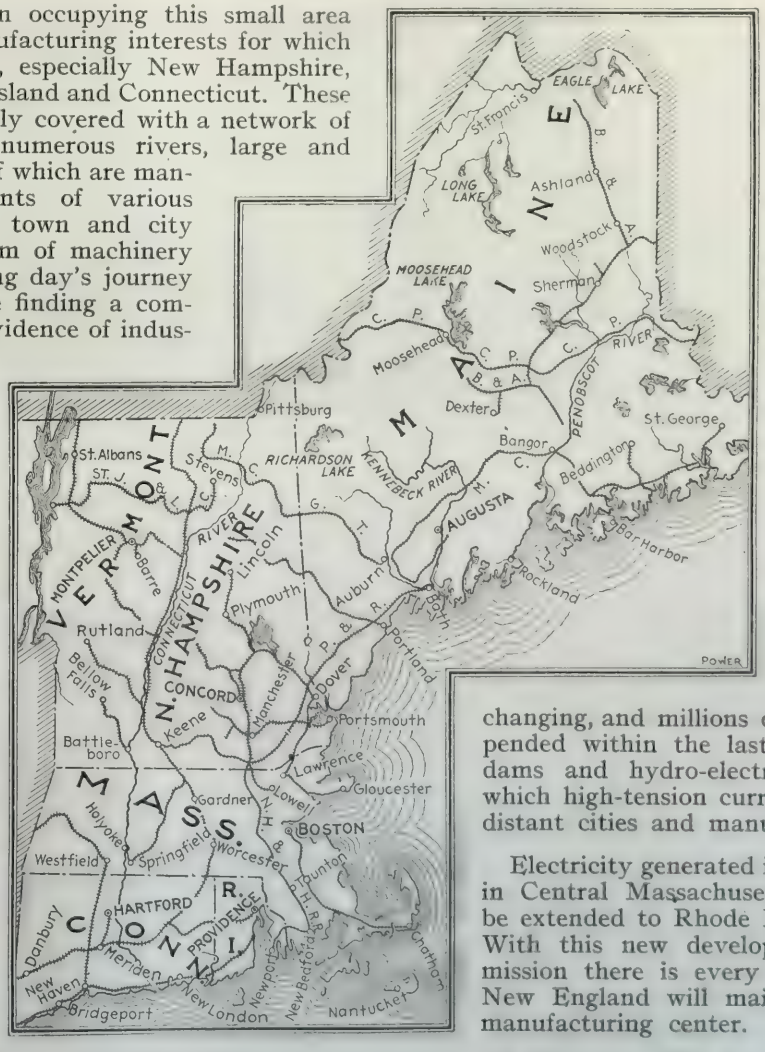
POWER

Vol. 36 NEW YORK, NOVEMBER 26, 1912 No. 22

New England of Today

FEW realize that the six New England states, although having less than one forty-fifth of the area, have about one-fourteenth of the inhabitants of the United States.

This large population occupying this small area is due to the vast manufacturing interests for which New England is noted, especially New Hampshire, Massachusetts, Rhode Island and Connecticut. These four states are practically covered with a network of railroads. There are numerous rivers, large and small, upon the banks of which are manufacturing establishments of various natures. Each hamlet, town and city is enlivened by the hum of machinery and one would go a long day's journey in New England before finding a community free from this evidence of industrial progress.



While the water power of these states plays an important part in the manufacturing field, the steam power plants predominate. The chimneys of thousands of plants stand as evidence to the many industries to which the boilers furnish motive power.

Blowing whistles break the silence of the morning quiet, calling hordes of workmen to their daily toil, and again sound release at night, when the day's work is over, to the same great army of toilers.

Each of these mills necessitates the operation of from one to sixty or more boilers, and a relative engine capacity. Well may it be imagined, then, to what a great number employment is given, of all ranks and degrees of skill from firemen, oilers, etc., to chief engineers, master mechanics, superintendents and consulting and designing engineers.

It would be difficult to estimate the thousands upon thousands of dollars that have been invested in power plant apparatus. These plants range from installations equipped with the simplest apparatus to extensive power plants in which the most economical and modern equipment has been installed.

Until recent years, the output of the electrical power plants of New England was confined to local use, the central stations distributing energy to but comparatively short distances. This condition, however, is changing, and millions of dollars have been expended within the last few years in building dams and hydro-electric power plants from which high-tension current will be delivered to distant cities and manufacturing centers.

Electricity generated in Vermont is being used in Central Massachusetts and eventually will be extended to Rhode Island and Connecticut. With this new development of power transmission there is every reason to believe that New England will maintain its prestige as a manufacturing center.

In this issue will be found descriptions of several power plants in operation in New England, and used for various manufacturing purposes.

Wachusett Dam Hydro Electric Plant

By Warren O. Rogers

The water in the Wachusett Reservoir in flowing to the aqueduct passes through four 1200-hp. turbine waterwheels, each directly connected to a 1000-kw. alternating-current generator. Each waterwheel is capable of passing approximately 44,000,000 gal. of water every 12 hours. The flow is measured by venturi meters.

The generation of electrical energy from water flowing from a reservoir into an aqueduct of a water-works system has been successfully accomplished by the Metropolitan Water Works Board of Massachusetts at the Wachusett dam, Clinton, Mass.

Water supplied to the city of Boston, Mass., by the Metropolitan Water Works is obtained from Lake Cochituate, the Sudbury River and the Nashua River. The former two sources of water supply were acquired by the city of Boston in 1846 and 1872 respectively. The construction work for storing and conveying the water of the Nashua River from the Wachusett Reservoir was begun in 1895 and completed in 1905. Water was first drawn from this source March, 1898.

Water is collected and stored in the Wachusett Reservoir on the Nashua River, in eight storage reservoirs on the watershed of the Sudbury River, and in Lake Cochituate. From these reservoirs the water is conveyed to the Metropolitan District through four aqueducts. The Wachusett Aqueduct, 12 miles long, conveys water from the Wachusett Reservoir to the Sudbury Reservoir. The Western Aqueduct, 12 miles long, conveys water from the Sudbury Reservoir to a point within 10 miles of the Metropolitan District. The Sudbury Aqueduct conveys water 17.4 miles from the reservoir on the Sudbury River to the Chestnut Hill Reservoir, where the well known Chestnut Hill high- and low-service pumping stations, with an aggregate capacity of 204,500,000 gal. for 24 hr., are located. The fourth aqueduct conveys water from Lake Cochituate 13.7 miles to the Chestnut Hill Reservoir, from which the city of Boston and 17 other municipalities obtain their supply.

The Wachusett dam is in a narrow gorge on the Nashua River, about one-half mile upstream from the town of Clinton, Mass. The dam is 1396 ft. long, 207 ft. high and the high-water elevation is 395 ft. above the city of Boston. The area of the reservoir's water surface is 4195 acres and the available capacity is 65,000,000,000 gal. of water. The reservoir is 8.41 miles long with a maximum width of 2.05 miles. The average depth of water is 46 ft. Some interesting figures regarding the dam are given herewith:

Height of top of main dam above full-reservoir level	20 ft.
Height of top of main dam above ground below dam	114 ft.
Maximum height of dam above rock foundation, about	207 ft.
Maximum thickness of dam at bottom, about	185 ft.
Thickness of dam at full-reservoir level	25 ft.
Maximum depth of water above dam	129 ft.
Masonry required for dam	280,000 cu.yd.

Of the 12 miles of the Wachusett aqueduct, two miles is a rock tunnel, seven

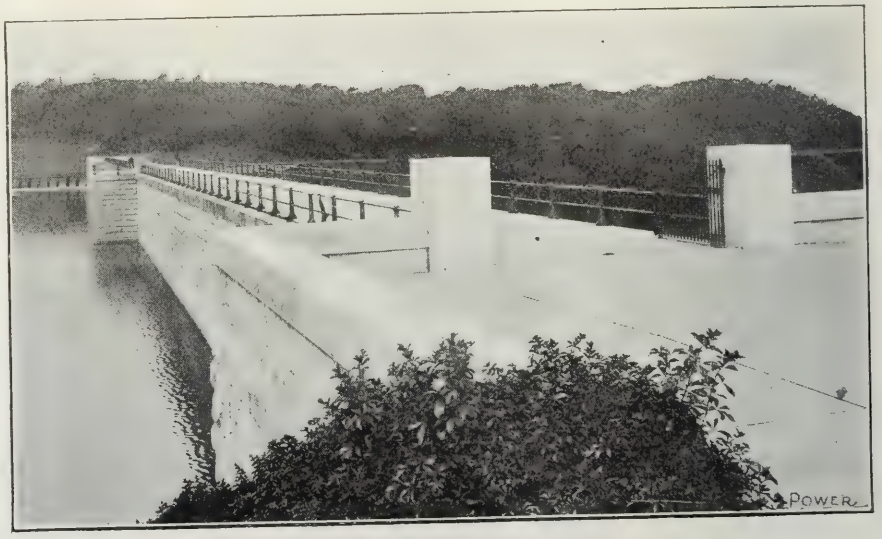


FIG. 2. WACHUSETT DAM PROMENADE

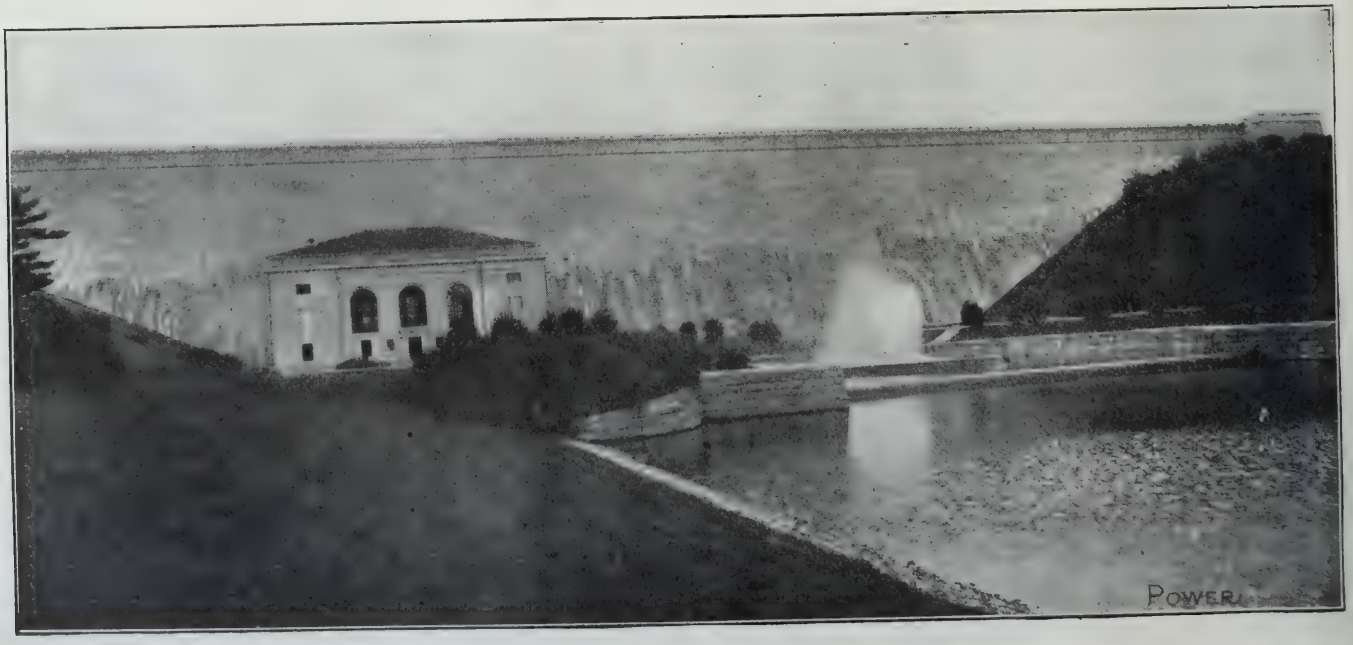


FIG. 1. WACHUSETT DAM AND HYDRO-ELECTRIC POWER HOUSE

miles concrete and brick masonry, covered with earth, and three miles an open channel. The masonry section is 11 ft. 6 in. wide and 10 ft. 6 in. high. The aqueduct has a capacity of 300,000,000 gal. per day.

Although the hydro-electric power-plant building was erected at the time the dam was built, it was not equipped with machinery until recently. It was estimated that about 3000 hp. could be developed by placing turbine waterwheels between the reservoir and aqueduct and utilizing the flow of water passing through them. A view of the dam and power house is shown in Fig. 1. A view of the promenade above the dam is shown in Fig. 2.

There are four 1200-hp. turbine waterwheels of the volute-casing type, each directly connected to a 1000-kw., three-phase, 60-cycle, 13,800-volt, alternating-current generator running at 400 r.p.m. and controlled by a hydraulic governor.

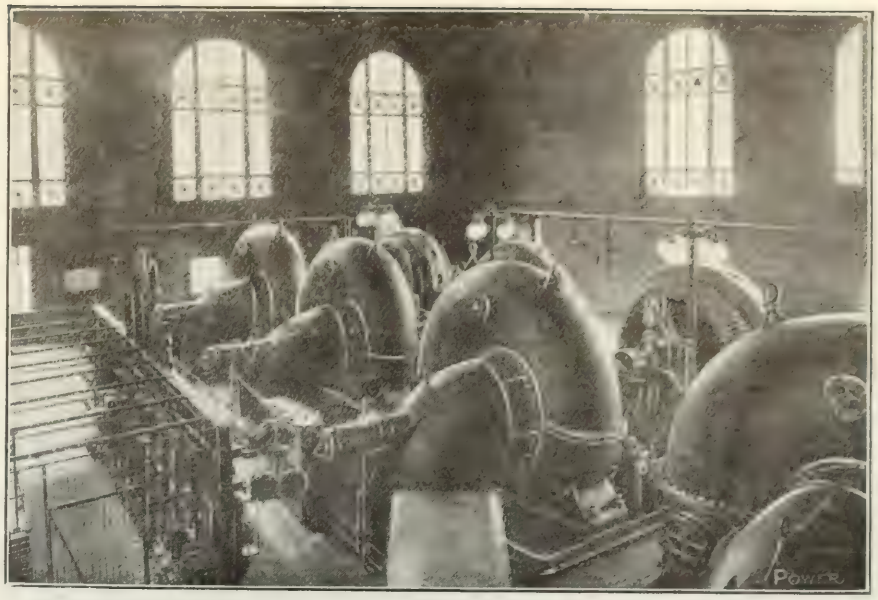


FIG. 4. VIEW OF THE WATERWHEELS, GENERATORS AND SWITCHBOARD

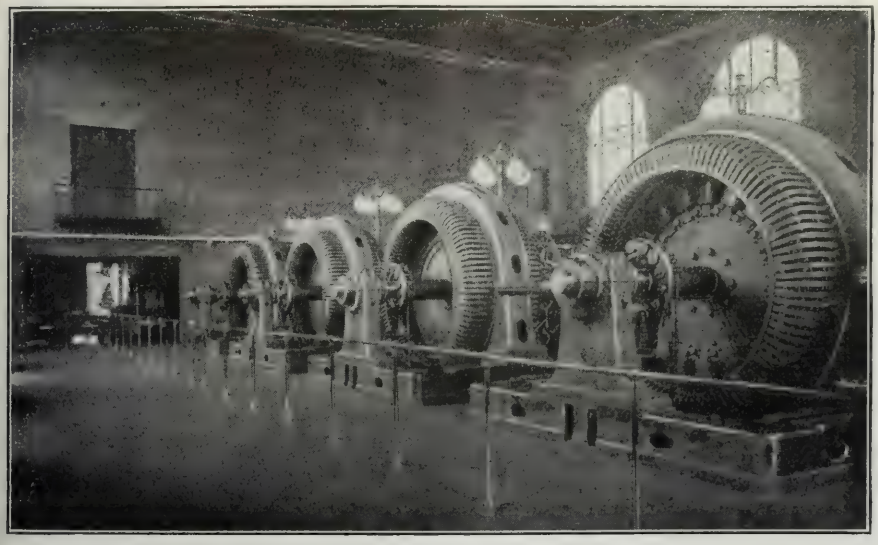


FIG. 3. GENERAL VIEW OF THE ALTERNATING-CURRENT GENERATORS

The units are arranged as shown in Figs. 3 and 4.

Two 60-kw., 125-volt direct-current exciter units are driven by turbine waterwheels, at 1000 r.p.m. They are set in a bay at the end of the generator room as shown in Fig. 3. Details of the installation are given in Fig. 5.

All the water from the Wachusett Reservoir goes through four vertical outlet wells, each 84 in. in diameter and 111 ft. high, and built in the dam. These are connected to 48 in. horizontal cast-iron pipes about 130 ft. long, which are also embedded in the dam masonry, and terminate in the power-house foundation. The turbine wheel foundations form the roof of the aqueduct. After the water passes through the waterwheels it discharges into the aqueduct. A general idea of the dam design and intake-pipe construction may be obtained from Fig. 6.

That accurate measurement of the total amount of water delivered to the aqueduct and also to the turbines may be had, four special venturi type meters have been installed. The meters also act as a check upon the economical operation of the turbines.

It was impracticable to place the upstream pressure chambers in the 48-in. pipe on account of the velocity of the water, about 14 ft. per sec.; therefore a special form of pressure chamber with numerous vent holes, flush with the inner contour of the 84-in. wells was constructed. Special throat pieces 40 in. in diameter are attached to the interior of the 48-in. castings at points approximately 90 ft. from the upstream chambers. These castings have annular pressure chambers and are lined with bronze, bored to the standard venturi curve.

Brass pipes, attached to the interior surfaces of the wells and of the 48-in.

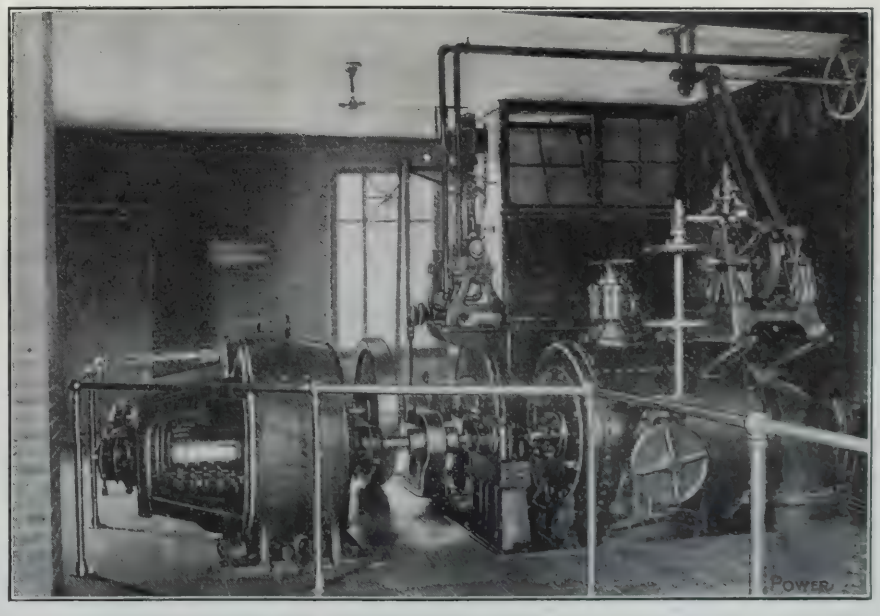


FIG. 5. WATER TURBINE-DRIVEN EXCITER UNITS

pipes, connect with the eight chambers and transmit pressure to the four meters on the main floor of the turbine room. One of the instruments is shown in Fig. 7. Each meter has three dials; one indicates the flow of water in gallons per day, one continually records the rate of flow upon a paper chart and another registers the total quantity of water in gallons.

These instruments are very sensitive, responding to the slightest change in the gate opening. The recording chart, besides registering the amount of water passing through the waterwheels, also shows at what time each started or shut down, and the rate at which it has operated.

At the time of the writer's visit, only two of the four units were running. Through these approximately 88,000,000 gal. of water passed every 12 hours. The water comes to the wheels under a 90-ft. head at a pressure of 37½ lb., there being a slight loss due to friction. The wheels run with a vacuum of 14 lb. Only three of the four units can be run at the same time, as the discharge from them practically equals the capacity of the aqueduct. If it is desirable to shut down all of the machines the water can be bypassed from the reservoir to the aqueduct.

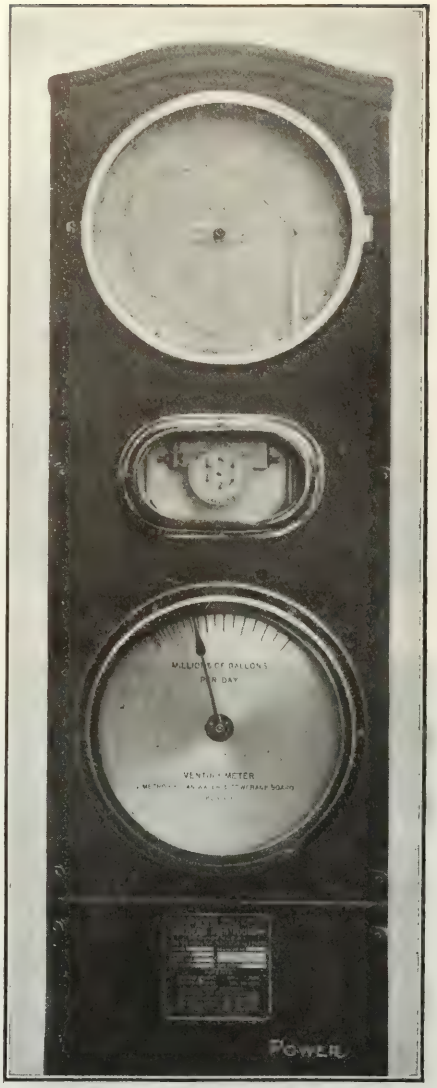


FIG. 7. VENTURI METER BOARD

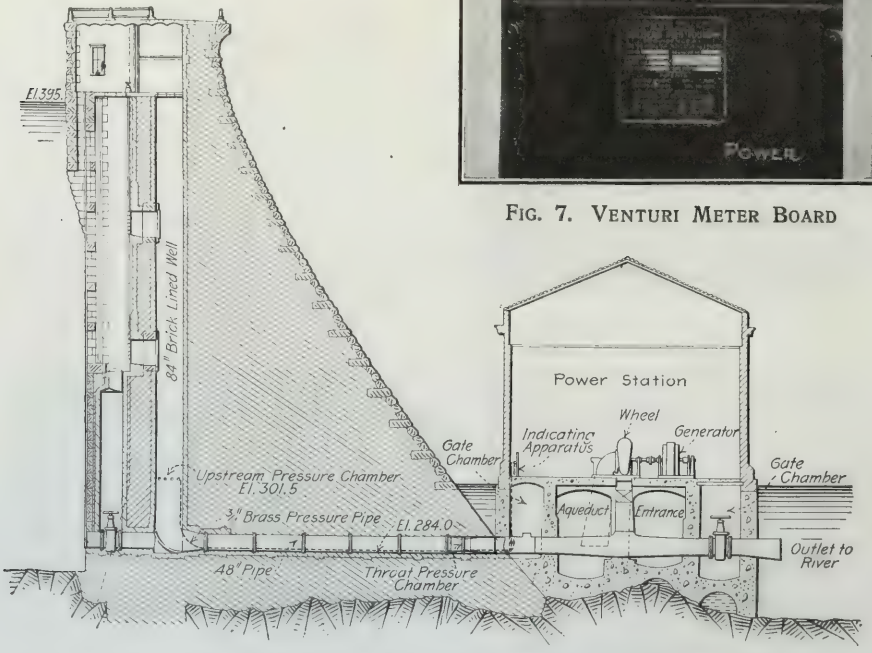


FIG. 6. SECTION OF DAM AND CONNECTION OF VENTURI METERS

The power house is of granite with red-tile flooring. The inside walls are of brick and the ceiling is of oiled hardwood finish. The electrical energy generated at this station is sold to the Connecticut River Power Co., and is transformed by it for commercial use.

Power Cost in a 6500-kw. Station

Figures for the Fall River (Mass.) Electric Light Co.'s plant for the year ended June 30, 1912, are given in *Electrical World*, Sept. 28. The station is a tidewater plant of 6500-kw. rating, containing six 350-hp. water-tube boilers and one 4000-kw., one 500-kw. and two 2000-kw. turbines. The 4000-kw. unit was new in 1911. In 1911 the plant

FALL RIVER STATION OPERATING COSTS
1912

Fuel (0.47c. per kw.hr.)	$34,164
Oil and waste	900
Water	1,206
Wages at station (0.19c. per kw-hr.)	13,687
Repairs of building	1,688
Repairs of steam equipment	1,538
Repairs of electrical equipment	686
Miscellaneous	1,490
Total, excluding fixed costs (0.76c. per kw-hr.)	$55,359

produced 5,764,466 kw.-hr. at a manufacturing cost of $51,840, or 0.9c. per kw.-hr., coal costing $3.67 per ton. The fuel consumption averaged 3 lb. per kw.-hr. This year the plant generated 7,293,783 kw.-hr. at a station cost of $55,359, or 0.76c. per kw.-hr., coal costing the company $3.53 per ton. The payroll of the generating department listed 16 men in 1911 and 18 in 1912.

The costs given include no fixed charges or administrative expenses, but show the results of station operation in 1912. The station load-factor for the year was 32.5 per cent.

In recounting his experience in the early days of railroading, when ball-and-lever safety valves were the only kind in use, a veteran locomotive engineer said: "The safety valves were made to blow at 110 or 120 lb. pressure and the amount of steam in the boilers was gaged by the weight required to keep the valve down while the engine was rounding a curve." In those days a boiler would blow itself every time the engine rounded a curve.

EQUIPMENT OF WACHUSETT DAM POWER PLANT

No.	Equipment	Kind	Application	Kw.	Volts	Amp.	Phase	Cycles	R.p.m.	Hp.	Type	Manufacturers
4	Turbines	Water	Main unit	..	....	...	...	....	400	1200	..	S. Morgan Smith Co.
4	Generators	Alt. current	Main unit	1000	13,800	42	3	60	400	..	..	Westinghouse Electric & Mfg. Co.
2	Turbines	Water	Exciter	..	....	...	...	....	1000	..	..	S. Morgan Smith Co.
2	Generators	Direct current	Exciter	60	125	480	...	....	1000	..	..	Westinghouse Electric & Mfg. Co.
4	Governors	Hydraulic	Water wheels	..	....	...	...	....	..	..	Q	Lombard Governor Co.
2	Governors	Hydraulic	Water wheels	..	....	...	...	....	..	..	..	Lombard Governor Co.
4	Meters	Venturi	Water flow	..	....	...	...	....	..	..	..	Builders Iron Foundry
1	Switchboard	Slate	Electrical	..	...	...	...	....	..	..	..	Westinghouse Electric & Mfg. Co.
1	Crane	12 ton	.....	..	....	...	...	....	..	..	..	Niles-Bement-Pond Co.

Power Plant of the Charlton Mills

By Warren O. Rogers

There are six 250-hp. return-tubular boilers with dutch-oven furnaces. A 1350-hp. cross-compound engine exhausts into a 600-kw. low-pressure turbine. A 300-kw. alternating generator, belt driven, is run in parallel with the turbine unit, and acts as a balancer of the electrical load.

Although the generating units in the power plant of the Charlton Mills, Fall River, Mass., are of standard design the arrangement is different from that found in other power plants used for similar work. The mill is adjacent to Cook's pond from which boiler-feed water and condenser-circulating water are obtained.

Boiler Room

The main feature of interest in the boiler room is the application of dutch-oven furnaces to the six 84-in., 250-hp. return-tubular boilers. Five of the furnaces are shown in Fig. 1. These have shaking grates and are hand-fired. When the regular delivery of coal fails to keep up the supply in front of the boiler, coal is wheeled from the pile in the yard.

Instead of being mounted close to the boiler fronts, as in general practice, the water columns and steam gages are brought out to the fronts of the furnaces. The steam and water connections to the water column are supported by hooks attached to vertical I-columns, as shown. A steam gage is piped to the steam connection of each water column by a U-tube and supported on the I-column by a bracket.

Smoke uptakes are placed vertically between the boilers and the main smoke flue. The latter runs above and across the front end of the boilers to a 150-ft. brick chimney, not shown. An economizer heats the feed water from 210 deg. F., as it comes from the exhaust-steam heater, to 270 deg. F., at which temperature it goes to the boilers. Exhaust steam from the auxiliary pumps is used in the feed-water heater.

A main feed pipe runs in front of the furnaces with a branch pipe connected to each boiler. Each branch feed pipe has a gate and globe valve with a check valve between. This permits of examining the check valve at any time.

The boiler house is built of stone and is separated from the engine room by a driveway. The plank roof is covered with tar and gravel.

Engine Room

A view in the engine room is given in Fig. 2. The main unit is a 26 and 42 by 60-in. cross-compound Corliss engine of 1350 hp. capacity, running at 85 r.p.m., with 145-lb. steam pressure. It is belt-connected to the mill lineshaft, which also drives a belted jackshaft in the basement; the latter is used to drive a 300-kw., three-phase, 60-cycle, belt-driven alternating-current generator at 514 r.p.m.

Exhaust steam from the low-pressure cylinder in going to the condenser, passes through a 600-kw. exhaust-steam turbine, Fig. 3. This machine is directly connected to a 750-kw., 600-volt, three-phase, 60-cycle alternating-current generator running at 3600 r.p.m. Exhaust steam comes to the turbine at 7-lb. pressure absolute, and a vacuum of 28 in. is maintained by the condenser to which the turbine is connected. The generating units are in a room on the first floor of the mill building and are supported by brick foundations in the basement.

Exciter Units

Two exciter units generate exciting current for the alternators. One set is a horizontal steam turbine directly driving a 50-kw., 125-volt direct-current generator, running at 2250 r.p.m. The other unit is a 25-kw., 125-volt generator, directly driven by a 32-hp. induction motor at 1150 r.p.m. The motor-driven unit is generally used; the turbine driven one is for use at the noon hour, or during overtime work, when there is a small motor load.

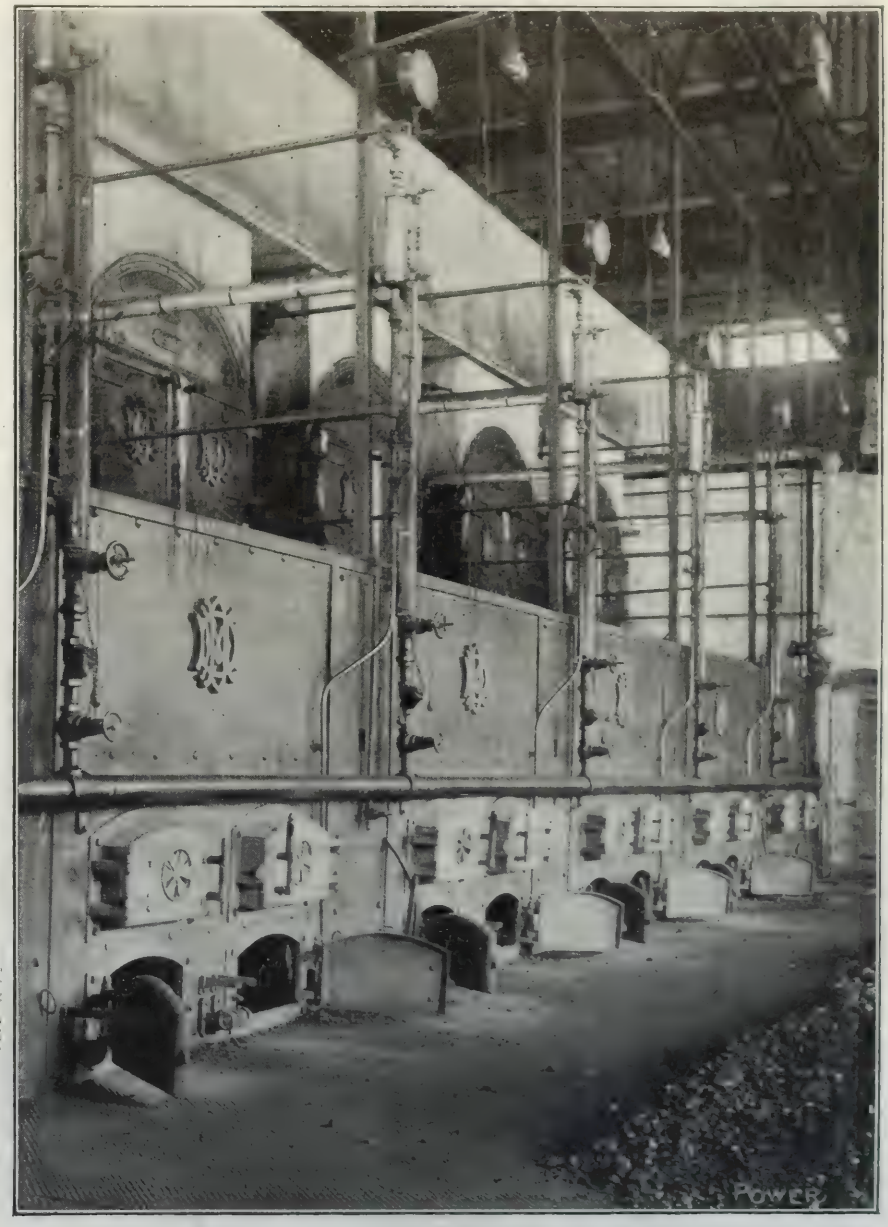


Fig. 1. Return Tubular Boilers and Dutch-oven Furnaces

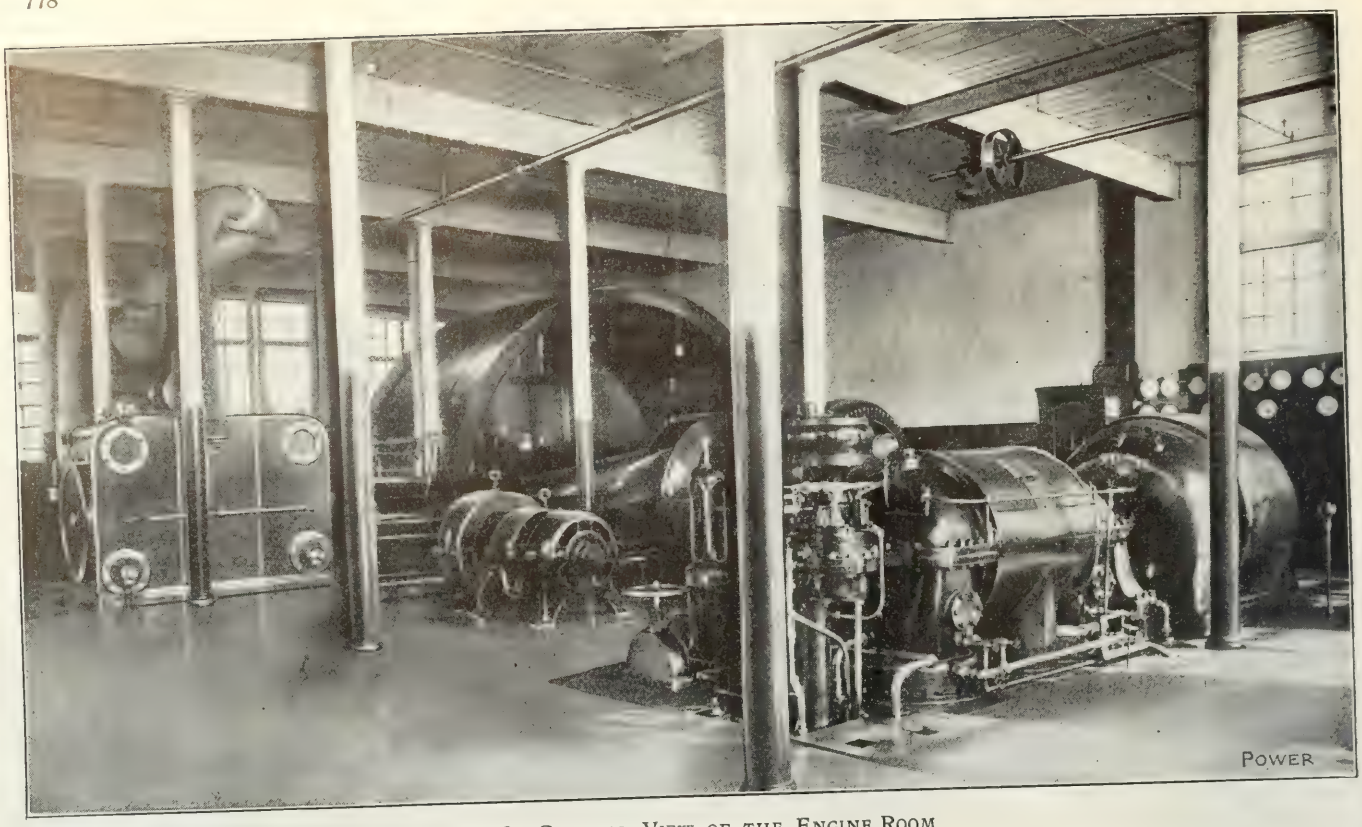


FIG. 2. GENERAL VIEW OF THE ENGINE ROOM

AUXILIARIES

All auxiliaries are in the basement. The condenser is shown in the foreground of Fig. 4. The other equipment includes an 8x10-in. triple-plunger power pump used for pumping water for the humidifying system in the mill, and a 5¼x6¾-in. triple-plunger pump that supplies the mill with water for various uses. For boiler feeding a 5¼x6¾-in. six-plunger power pump is used. These pumps are belt-driven from a shaft hung from the basement ceiling and driven from the jack-shaft in the basement.

A 4½x4½-in. belt-driven air compressor furnishes air for the humidifying system. An 18 and 10 by 12-in. duplex pump is connected to the fire-service lines with which the mill is protected. The exhaust steam from the exciter turbine, condenser turbine and the steam pumps piped to the mill-heating system, goes to the feed-water heater. The returns from the mill and drips from the steam pipe run into a tank from which they are pumped to the boilers. The condenser takes water from a well, under the basement floor, that is connected with the adjacent pond. No attempt is made

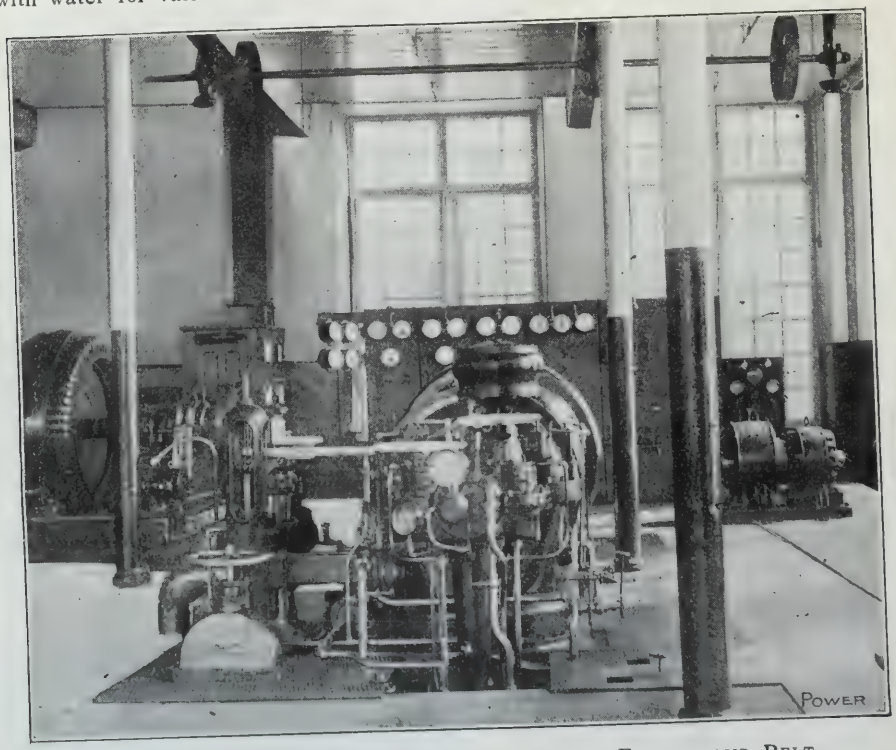


FIG. 3. EXHAUST STEAM TURBINE, MOTOR-DRIVEN EXCITER AND BELT-DRIVEN ALTERNATOR

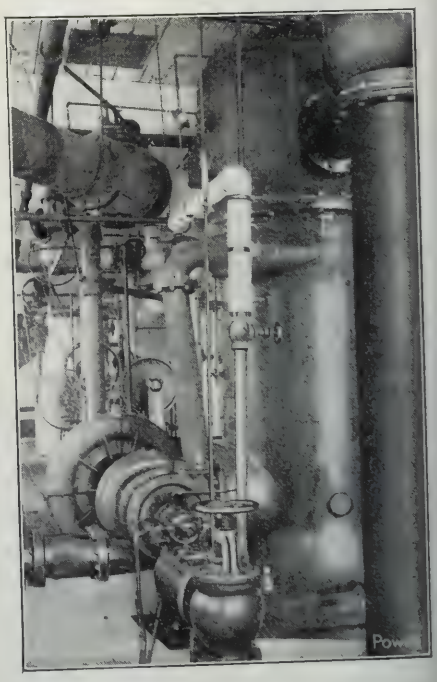


FIG. 4. LEBLANC CONDENSER AND AUXILIARY PUMPS

to save the condensing water and it is discharged to the sewer.

Operation of Alternators

In the mill are 11 induction motors, supplied with current by the alternators. Two direct-current motors in the card room and one in the machine shop are driven by the small direct-current turbine unit. Four of the induction motors are of 100 hp. capacity; the others of 35 hp. capacity. During the noon hour, nights and at other times when it is necessary to run the card room or machine shop, the steam-driven exciter is used. The engine, belt-driven exciter and low-pressure turbine unit are shut down at noon, but the turbine-driven exciter is operated to furnish exciting current for the belt-driven alternator.

After a shutdown the motor-driven exciter, which is left switched in on the switchboard, starts as soon as the alternator generates current, and as it reaches normal speed the exciting load for both alternators is switched to it.

If the steam-exciter unit becomes disabled the motor-driven machine can be belt-driven by a countershaft, shown in Fig. 3, belted to a pulley on the shaft of the belt-driven alternator until the exciter unit is up to speed and the motor switches thrown in, when the belts are removed.

An interesting feature of the plant is the way the two alternators work together. The belt-driven unit runs only when the steam engine is operated. The exhaust-steam turbine is also dependent upon the engine for steam. Upon shutting down, the turbine generator is cut out of circuit at the switchboard and only phased into circuit after both alternators have gained their normal voltage after starting up.

This arrangement of alternators produces a balancing of the load between the engine and turbine-driven units. The load on the alternators is greater than the capacity of the turbine, which tends to slow down the turbine, but as the two alternators are run in parallel, the excess load is thrown on the engine-driven unit. This causes the engine to cut off steam later in the stroke and thus exhausts more steam to the turbine, which again carries its load. This balancing of the load takes place so long as the units are operated in parallel.

A six-panel switchboard carries the recording instruments and controlling equipment for the energy generated by the electrical units. The engine and boiler rooms are light and roomy. The engine room is finished with an oiled, hard-wood floor, pea-green walls with dark-red wainscoting and white ceiling. The plant is under the direct charge of Francis Clegg, chief engineer.

EQUIPMENT OF THE CHARLTON MILLS POWER PLANT

No.	Equipment	Kind	Application	Kw.	Volts	Amp.	Phase	Type	Cycles	R.p.m.	Size	Pres. lb.	Hp.	Manufacturers
6	Boilers	Return tubular	Steam								84"x20	145	250	D. M. Dillon Steam Boiler Works
1	Economizer	Sturtevant	Feed water										...	B. F. Sturtevant Co.
1	Heater	National	Feed water										...	The National Pipe Bending Co.
1	Engine	Rice & Sargent	Main unit							85	26x42x66"	145	1350	Providence Engineering Works
1	Generator	Alt. current	Belt driven unit	300	600	288	3		60	514				Westinghouse Electric & Mfg. Co.
1	Turbine	Low pressure	Main unit	600						3600		7 abs	...	Westinghouse Machine Co.
1	Generator	Alt. current	Turbine unit	750	600	725	3	...	60	3600	...	...	...	Westinghouse Electric & Mfg. Co.
1	Condenser	Leblanc	Turbine								11	...	...	Westinghouse Mfg. Co.
1	Turbine	Single stage	Exciter unit	50						2250		145		Westinghouse Mfg. Co.
1	Generator	Direct current	Exciter unit	50	125	400				2250		...	...	Westinghouse Electric & Mfg. Co.
1	Motor	Induction	Exciter unit		600	34.2	3	C.C.L.	60	1150		...	32	Westinghouse Electric & Mfg. Co.
1	Generator	Direct current	Exciter unit	25	125	200				1150		...	...	Westinghouse Electric & Mfg. Co.
1	Pump	Triple plunger	Humidifying water								8x10"	...	...	Deane Steam Pump Co.
1	Pump	Triple plunger	Mill water								5½x6½"			I. B. Davis
1	Pump	Six plunger	Boiler feed								5½x6½"			I. B. Davis
1	Compressor	Belt driven	Air								4½x4½"			Ingersoll-Rand Co.
1	Pump	Duplex	Fire								18x10x12"			Warren Steam Pump Co.
1	Switchboard	Slate	Electrical control											Westinghouse Electric & Mfg. Co.

A Power Plant Data Sheet

By J. C. Hawkins

The accompanying data sheet was laid out for an 800-hp. power plant. It can be modified to suit the requirements of any ordinary demand.

Power-plant records are necessary in keeping track of the efficiency of the plant. Before an engineer can calculate the cost of the energy generated he must have the data at hand from which to make his calculations.

The systems of records used in most large central stations are necessarily complicated and each item is figured out in detail. This is desirable, but the clerical work attached thereto prohibits its use in the small plant, where the engineer has to do his own bookkeeping, but every steam plant should have some form of daily record of its operation. After several years' experience in plants where various kinds of records were kept, the one shown herewith was drafted for present use, but may be changed to fit any conditions.

The sheet is 13x16 in. and is filed in a loose-leaf file. To illustrate the scope of the record sheet and make it more easily understood, a plant of 800 hp. capacity is taken as an example, in which are four engines and four boilers, a heating system, using forced circulating hot water, the heaters using exhaust steam, supplanted by live steam when necessary, and also furnishing hot-water service. There is also an ice-making plant and a vacuum-cleaning machine, the general layout being such as is found in a modern office building. The sheet is laid out for three eight-hour watches.

Under the heading "Data" the various temperatures and pressures are recorded every two hours, or oftener if required. The outside temperature and condition of the weather will help greatly in keeping the water in the heating system at the proper temperature; any change in the outside temperature is met by a change in the heater temperature before that of the building varies to any extent.

The temperature of the brine in the ice tank is recorded, as any change will affect the output of ice. It is also well to take the temperature of the gas in the suction and discharge pipes and that of the cooling water. The four items, temperature of flue gas, per cent. of CO_2 in the flue gas, draft and condition and time of cleaning the fires, play a very important part in the economical operation of the plant, for it is here that much of the waste of the plant occurs and even if suspected is apt to be neglected.

To obtain this information, a high-reading thermometer or pyrometer, a draft gage and some form of CO_2 testing out-

fit will be required. If a CO_2 recorder is used, its reading should be noted so that the temperature, draft, etc., can be readily compared. If the fires are given a reasonable amount of attention, the use of these instruments will soon pay for themselves in the increased economy of the boilers.

Both volt and ampere readings should be noted, as they are a check on the operation of the boilers and engine, also the variation of load during the 24 hr. This data is noted on the sheet every two hours, or oftener, if desired, by the assistant engineer or oiler. An examination of the report will show any variation in the unit of output per unit of input, and after the engineer has watched this data for awhile, any variation in the various items will show just where the trouble lies.

Weights of coal, ashes and feed water should be recorded at the end of each watch, as they will not only reveal any mismanagement on the part of the employees, but will show just what conditions of fire draft, per cent. CO_2, etc., will give the highest evaporation per pound of coal, and whether or not instructions are being carried out.

Where live steam is used to help out the exhaust steam in heating or for any purpose of manufacture, it should be metered, or the condensation measured if more convenient. If used in heating, any increase in live steam used in proportion to the temperature changes, condition of the weather, exhaust steam, etc., would indicate waste somewhere, and would lead to an investigation; also measuring the live steam and deducting it from the total weight of steam made, and dividing this by the horsepower will show whether the steam consumption of the engine is as low as it should be, and help to locate any unseen steam leaks or other waste.

Under the heading "Operation," the time of starting and stopping each piece of machinery is recorded, also the hours run. This is desirable for many reasons, one of which, in the case of the boilers, is the period of service since cleaning. If any piece of machinery is in operation 24 hr. a day without a stop, the engineer on the first watch when making out a new sheet, writes "On" in the column headed "Start" and the engineer finishing the last watch marks up 24 hr. in the "Hours Run" column.

THE W. H. JOHNSON COMPANY
DAILY REPORT OF ENGINEER

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Data	First Watch 7 A.M.	9 A.M.	11 A.M.	1 P.M.	Second Watch 3 P.M.	5 P.M.	7 P.M.	9 P.M.	Third Watch 11 P.M.	1 A.M.	3 A.M.	5 A.M.
Steam Pressure												
Exhaust Pressure												
Outside Temperature												
Weather												
Temp. B'l'g H'd'r Supply												
Temp. B'l'g Return												
Temp. H.W. service												
Temp. Brine in Ice Tank												
Temp. Feed Water												
Temp. Engine Room												
Temp. Flue Gas												
% CO_2 in Flue Gas												
Draught-Inches												
Fire Cleaned-Condition												
Voltmeter												
Ammeter												
Coal and Ashes	Coal		Ashes		Coal		Ashes		Coal		Ashes	
Feed Water Meter	Read		Used		Read		Used		Read		Used	
Live Steam Meter	Read		Used		Read		Used		Read		Used	

Operation	Start	Stop	H'r's Run		Start	Stop	H'r's Run
Engine No.1				Boiler Feed Pump No.1			
Engine No.2				Boiler Feed Pump No.2			
Engine No.3				General Service Pump			
Engine No.4				Vacuum Pump No.1			
Fan Engine No.1				Vacuum Pump No.2			
Fan Engine No.2				Circulating Pump No.1			
Boiler No.1				Circulating Pump No.2			
Boiler No.2				Sump Pump			
Boiler No.3				Vacuum Cleaning Machine			
Boiler No.4				Ice Machine			
Fire Pump				Coal Crusher & Conveyor			
Assistant Engineer							
Oiler							
Firemen							
Coal Passer							
Other Labor							

Wattmeter	Negative Lt'g	Negative P'r	Diff'ce	Positive Lt'g	Positive P'r	Diff'ce		
Unit No.1							Total Coal	Total K.W.
Unit No.2							Total Ash	Total L.S't'n
Unit No.3							Total F.W. per	Ice Pulled
Unit No.4							Evap. Pr. Lb's	Gen. Gas Wat'r
Power Load							Coal Rec'd	Ashes Rm'v'd

K.W's | Remarks, Repairs, Material used etc. cont'd. otherside

Chief Engineer

Power

SIMPLE AND COMPACT DATA SHEET

All generators are on the three-wire system with the lighting circuits tapped between the outside and neutral wires, with a wattmeter in each side between the machine and busbars. The power circuit is taken from the two outside busbars, double the lighting voltage. The readings of the wattmeters on the machines give the total kilowatt load and the reading of the power wattmeter subtracted from this gives the total kilowatt lighting load. With several buildings supplied it would be desirable to have a meter on the main circuit to each building.

Under the heading "Totals" the total amount of coal, water, ashes, live steam, etc., are recorded, also oil, grease and waste, and coal received. The wattmeters are read and the totals figured by the engineer at the end of the third watch, and the meter readings are copied on the new sheet by the next man when coming on duty.

Under remarks, repairs, etc., are recorded all repairs and changes made, the material used and any other information that may be of interest, and signed with the initials of the engineer in charge of the work.

After the report is completed it is examined and signed by the chief engineer or superintendent and placed on file for future reference.

Each individual plant will require some change in the items to make the report sheet fit the local conditions, but the general makeup will fit many plants, and enough information may be had from it to keep the plant in first-class condition if use is made of the information that is gained.

Looking for a Substitute for Gasoline

What promises to be an investigation of more than usual interest and utility is being conducted by the State Engineering Experiment Station at the Pennsylvania State College under the direction of Prof. J. A. Moyer, of the mechanical-engineering department. With the increase in the price of gasoline has come the demand for some cheaper fuel that will give as good results. Kerosene would meet the demands if a satisfactory carburetor could be designed, and it is with a view of determining the merits and defects of various types of carburetors that the investigation is being carried on. The price of gasoline has nearly doubled in this country in the past year, while in England it has risen correspondingly. Many advocate the use of a mixture of gasoline and kerosene, but point out that without an efficient carburetor no such mixture may be used. Prof. Moyer has already installed at the college a motor built to utilize the energy from either of the two fuels mentioned.

Steam and Hydro Electric Power Plant

By R. O. Warren

For many years the smaller towns and cities of New England have depended upon small steam power plants for local lighting and power. During these years many thousands of water horsepower have been allowed to go to waste, with little or no attempt to utilize the natural resources with which these states abound.

A change is taking place, however, and hydro-electric plants are being installed to utilize this water power which, in many instances, has been considered of no commercial value.

By installing a hydro-electric unit, the ordinary station load is carried without the aid of the steam engines, which are held in reserve.

At peak loads or during stages of low water, the load above the capacity of the turbine is carried by one of the steam units.

FIG. 1. POWER PLANT AND CANAL AT WINDSOR LOCKS, CONN.

The Northern Connecticut Light & Power Co., Windsor Locks, Conn., can be taken as an example of what is being done in this direction. Its power plant is at the lower end of a canal supplying several mills with water power. It is built of brick, as shown in Fig. 1.

An interior view of the engine room, Fig. 2, shows the layout of the three steam engines. One is a 13 and 25 by 16-in. double flywheel, center-crank, compound engine, belt connected to a 165-kv.-a. generator running at 600 r.p.m. The generator shaft extends through the bearing at one end, upon which two pulleys are mounted and which carry the two belts from the engine. A third pulley carries a belt driving a compound-wound, 11-kw., 125-volt exciter running at 11.25 r.p.m.

A 13 and 25 by 16-in., 325-hp. condensing Corliss engine is belted to a 200-kw. alternator. The engine runs at 150 r.p.m. and the generator at 600 r.p.m. A 120-volt, direct-current exciter-generator is belt driven from the alternator shaft at 1200 r.p.m.

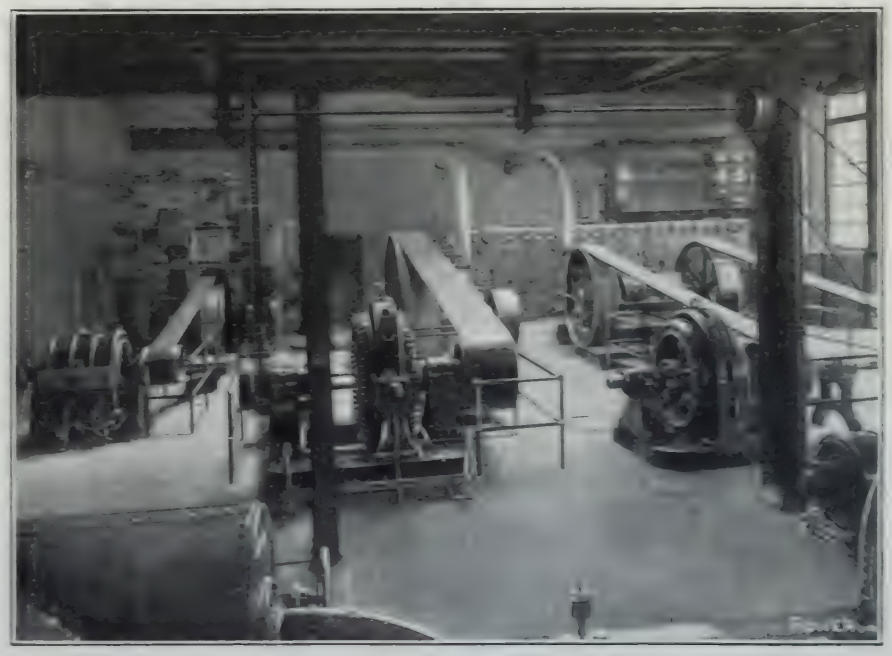


FIG. 2. PARTIAL VIEW OF THE ENGINE ROOM

The third unit is driven by a 12 and 19 by 16-in. vertical compound engine of 250 hp. It is belt connected to a 135-kw. generator running at 900 r.p.m. The 120-volt exciter for this unit is belt driven from a pulley on the generator shaft at 1100 r.p.m.

These units receive steam from three return-tubular boilers, Fig. 3. Forced draft is supplied under the grates by a steam turbo-blower for the limited period the boilers are in service, which is during the peak load each day. The engines run condensing and are piped to two barometric condensers. One has been made from the condenser part of an old jet pump. A vacuum of 25 to 26 in. is maintained.

The day and night load is carried by a 250-hp. water turbine and generator. This unit is at the rear of the engine and is shown in Fig. 4. The wheel runs at 277 r.p.m. and takes water from the canal at a 26-ft. head. The water is discharged through a steel penstock to the Connecticut River. A hydraulic waterwheel governor controls the speed of the unit.

Directly coupled to the shaft of the waterwheel is a 180-kw. generator. A 125-kw., direct-current, 125-volt exciter is belt driven from the generator shaft. This unit is relieved of peak loads by one of the steam-driven generators. Also at periods of low water. The four generators are three-phase, 60-cycle, 2300-volt machines.

A blue Vermont-marble switchboard at one end of the engine room contains four exciter panels, four generator

panels, one transmission panel, one local totalizing panel, three commercial lighting and power panels and one street-lighting panel.

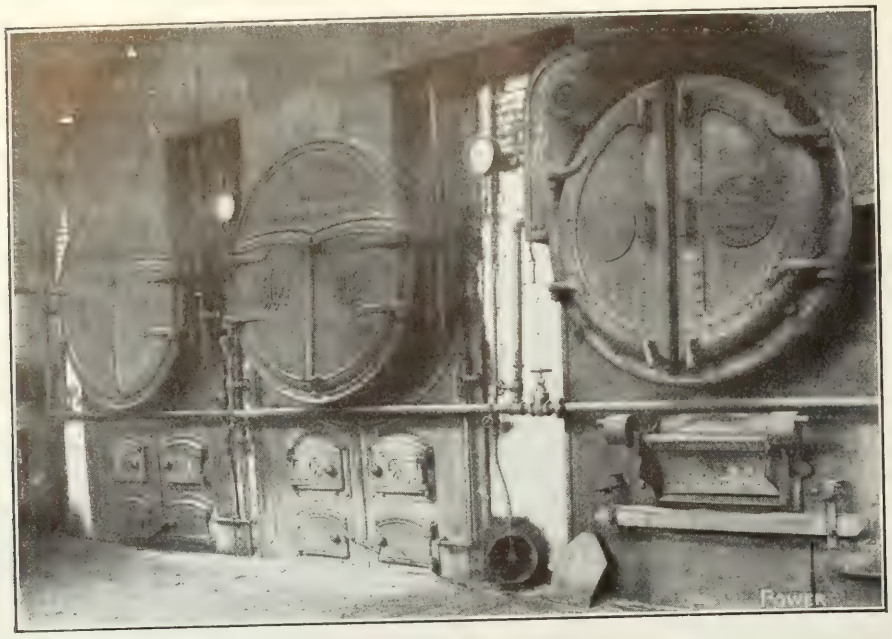


FIG. 3. STEAM BOILERS AND STEAM-DRIVEN BLOWER

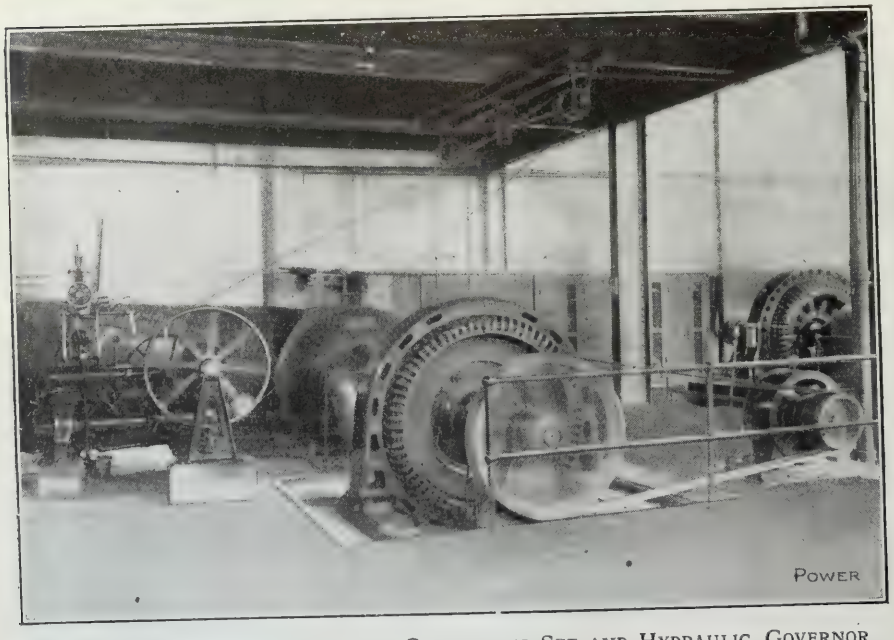


FIG. 4. 250-HP. WATER-TURBINE GENERATING SET AND HYDRAULIC GOVERNOR

Three 75-kw., 2300-6600-volt transformers step up the potential for the circuits to the Thompsonville substation. The longest transmission line is 17 miles. The average load of the station is 1600 kw.-hr. during the summer months and 2700 for the winter months. A record is kept of what the plant is doing, readings being taken at stated intervals and the record filed in the office.

Bearing Data Sheets and Binder

The New Departure Manufacturing Co., Bristol, Conn., has got out a series of data sheets devoted exclusively to the adaptability of "New Departure" ball bearings to machine tools and shaft hangers.

These data sheets cover the three types of bearings manufactured, each of which has its peculiar and particular qualifications. The sheets are in convenient letter size, bound in a loose-leaf binder, and give technical data of specific mountings in milling-machine heads, speed cones and drive pulleys of drilling machines, air-compressor bases, high-speed grinding spindles, grinding or polishing heads, etc.

Other sheets describe the construction of a shaft hanger equipped with a special ball bearing, friction-elimination qualifications under varying loads and a self-aligning feature for this character of work. The sheets and loose-leaf binder will be sent upon request.

What is claimed to be the largest steam turbine-driven centrifugal pump ever built in this country is to be installed by the De Laval Steam Turbine Co., Trenton, N. J., in the Ross pumping station of the Pittsburgh water-works. The rated capacity of the unit is 100,000,000 gal. per day, against a total head of 56 ft., or more than 980 water horsepower. The pump is guaranteed to show a duty of 115,000,000 ft.-lb. per 1000 lb. of dry steam, all the steam used by the condensing equipment being charged against the main unit.

MECHANICAL EQUIPMENT OF THE NORTHERN CONNECTICUT LIGHT AND POWER CO.

No.	Equipment	Kind	Purpose	K.w.	Volts	Amp. per terminal	Phase	Cycles	R.p.m.	Size	Hp.	Pres. lb.	Manufacturers
1	Engine	Corliss	Main unit	...	...	...	...	...	150	16x30"	325	120	Allis Chalmers Co.
1	Engine	Ball	Main unit	...	...	...	...	...	225	13x25x16"	300	120	Ball Engine Co.
1	Engine	Erie	Main unit	...	...	...	...	...	...	12x19x16"	250	120	Lake Erie Engine Co
1	Generator	Stanley	Main unit	135	2300	...	3	60	...	...	...	...	Stanley Electric Co.
1	Generator	Crocker-Wheeler	Main unit	165	2300	41.1	3	60	600	...	...	...	Crocker-Wheeler Co.
1	Generator	Bullock	Main unit	200	2300	44	3	60	600	...	...	...	Allis Chalmers Co.
1	Generator	Crocker-Wheeler	Main unit	180	2300	45.2	3	60	277	...	...	...	Crocker-Wheeler Co.
1	Generator	Direct current	Exciter	11	125	88	...	...	1250	...	...	...	General Electric Co.
1	Generator	Direct current	Exciter	...	120	50	...	...	1200	...	...	...	Allis Chalmers Co.
1	Generator	Direct current	Exciter	12	120	16.7	...	...	1100	...	...	...	Crocker-Wheeler Co.
1	Generator	Direct current	Exciter	12.5	125	100	...	...	750	...	...	...	Westinghouse Electric & Mfg. Co.
1	Boiler	Return tubular	Steam	...	...	...	...	...	...	60"	90	120	Holyoke Steam Boiler Co.
2	Boilers	Return tubular	Steam	...	...	...	...	...	...	72"	150	120	D. M. Dillon Steam Boiler Works.
1	Blower	Sturtevant	Forced draft	...	...	...	...	...	...	...	250	...	B. F. Sturtevant Co.
1	Water wheel	Hanover	Main unit	...	...	...	...	...	...	...	...	...	Lombard Governor Co.
1	Governor	Hydraulic	Water wheel	...	...	...	...	...	...	...	...	...	Dean Bros. Steam Pump Works.
1	Condenser	Barometric	Engine	...	...	...	...	...	...	...	...	...	Home-made.
1	Condenser	Barometric	Engine	...	...	...	...	...	...	...	...	...	

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

An Emergency Rheostat

By C. V. Hull

A short time ago conditions arose which demanded that a 30-hp. 220-volt motor be used with a wide variation in speed and load. The conditions were quite hard to meet because the load would sometimes be quite heavy at low speed and light at high speed.

The range of speed and load conditions made control by changing the field strength of the shunt field entirely out of the question. The compound winding was not used. The motor was required immediately, and there was not sufficient time to get a standard speed-control rheostat. Even if the time had been ample, it would have been unwise to order an expensive rheostat which would not be needed after two or three days. Accordingly, the electrical department arranged the rheostat shown in the illustrations.

The rheostat proper was made up of

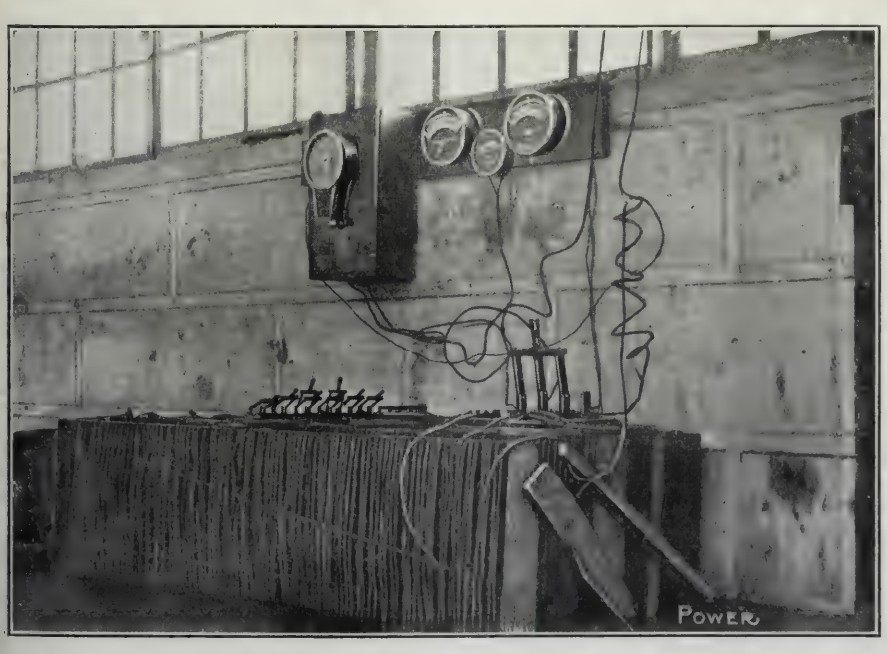


Fig. 1. View of Rheostat and Instruments

2x4-in. hemlock frames, covered with sheet asbestos, and wound about with ½-in. strips of No. 20 galvanized iron, cut from sheets 8 ft. long. These windings of sheet metal were divided into 10 steps. Since it was not practical to use a row of contacts and a sliding lever contact, as is usual, ten 15-amp. double-pole, single-throw switches were substituted in place of single-pole, single-throw switches as none were in stock.

All the switches were opened before the motor was started. Then switch No. 1 was closed and the motor speeded up by closing as many of the small switches as necessary to obtain the desired speed, which could be reduced at will by opening the required number of small switches.

In operation there was so much resistance in the rheostat that the motor would not start, with even a small load, until one of the small switches was closed. It will be seen that the speed of the motor could be controlled by the number of small switches closed or open. Thus it was possible to get almost any speed at any load within the limit of the motor's power.

One voltmeter *B* gave the line voltage while the other *C* gave the armature drop. An ammeter *A* and a wattmeter *D* were connected for collecting the data of the tests.

As a temporary outfit, the rheostat was more than satisfactory. The speed control was simple and accurate and the flat windings radiated heat so rapidly that there was no trouble on that score. The addition of a field rheostat would have given an almost perfect speed control.

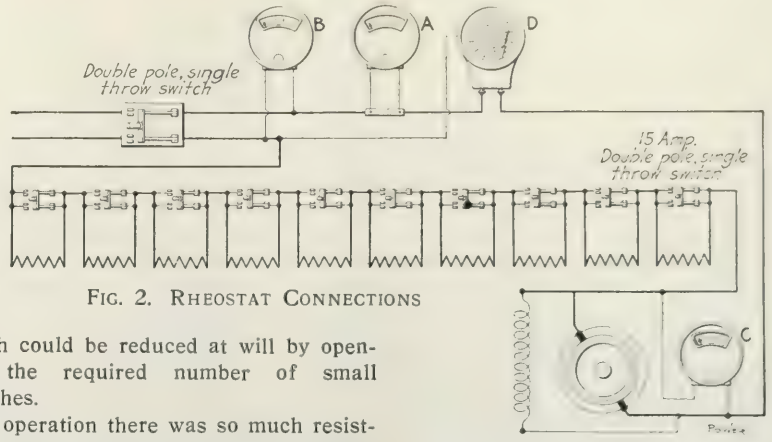


Fig. 2. Rheostat Connections

Simple Synchronizing Method

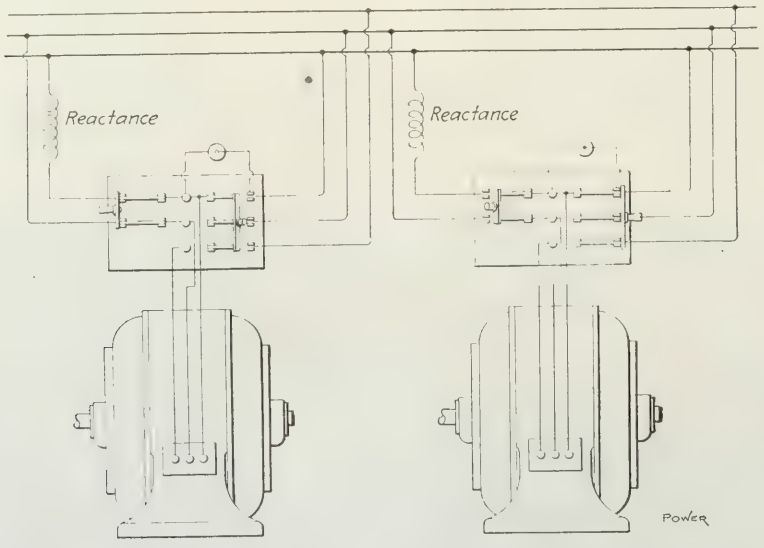
By Gordon Fox

The operation of synchronizing alternators in large central stations has long been accomplished with comparative ease. In the larger plants there are two or more men on duty and it is possible for one to manipulate the throttle or governor of the incoming engine or turbine while the other does the switching at the board. In some installations provision is made for controlling the governors electrically from the switchboard. There are, however, many smaller "one man" plants in which the problem of getting two machines in synchronism is less simple.

With the present prevalent method of switching it is necessary for the operator, when cutting a new machine onto the line, to adjust the speed of the incoming alternator to obtain exactly correct frequency, then to adjust the voltage and await opportunity to close his main switch when the frequency and phase correspond with busbar conditions as indicated by lamps or a synchroscope. A fluctuating load causes slight changes in line voltage or frequency. Also it is difficult to maintain a constant speed on the incoming machine running without load. For

these reasons the synchronizing operation is often a rather difficult and tedious one.

A switching arrangement has recently been tried out which permits cutting in an alternator without the necessity of first obtaining a synchronous condition. The principle involved is merely that of throwing a single phase of the incoming machine onto the busbars with a sufficient reactance inserted in series to choke down the initial rush of current. The so called synchronous impedance of the armature circuit is increased and the reduced cross-current quickly draws the machine into step. The main switch can then be closed and the reactance cut out of the circuit. The result is accomplished in a manner free from jerk or jar and without detrimental effect upon the busbar voltage.



SWITCHES AND CONNECTIONS FOR PUTTING TWO GENERATORS IN PARALLEL

The circuits may be arranged as shown. Double-throw oil switches with the interlocking devices removed may be used readily, although for simplicity in the illustration ordinary knife switches are shown. In switching in a new machine the generator is driven at approximately correct speed and the voltage adjusted to correspond with that of the busbars. The left-hand switch on the incoming generator panel is then closed, throwing the single phase on the line with the series reactance inserted. If synchronizing lamps are provided they will quickly indicate synchronism. The main right-hand switch may then be closed and the left-hand switch opened. The rheostat of the incoming machine is then adjusted to obtain a minimum wattless current.

It will be seen that with this method of operation it is almost as simple to parallel alternators as direct-current machines. The method has been tried successfully in several plants. It has been found possible for one man to start a unit from rest and place it in operation on the busbars in less than one minute. The simplicity and effectiveness of the system should recommend its general use for small plants.

CORRESPONDENCE

Generator Trouble

The trouble with Mr. Landmesser's generators mentioned in the Oct. 15 issue is probably due to the compounding, assuming the engine regulation to be good, and is caused by varying voltage due to a change of load. As the load changes the voltage may run up on one machine sufficiently to allow the current to flow into the other generator. The most likely remedy would be to adjust the shunts on the series winding so there will be less variation in the voltage as the load changes. It is hardly to be expected that the series winding on the large machine is connected in opposition to the shunt winding which would tend to reduce the voltage as the load increased.

There are automatic voltage regulators on the market, but leads for these would have to be run between generators to cause the regulators to operate.

It might be well to see that trolley lines and feeders do not make temporary connection with other wires carrying high-voltage currents. I had the same trouble, caused by a lighting line coming in contact with a trolley wire every time a car passed a certain spot. The trouble was not located until one Sunday when the plant was shut down and the voltmeter needle was seen to move. An extended investigation of the neighborhood soon showed the cause, and when this was removed the trouble ceased.

Of course, it may be that the engines do not regulate as quickly as they should, and thus allow the voltage to rise or fall as the load varies, or a poor field connection might cause the machine to be slow in building up when a heavy load came on.

THOMAS H. WATSON.

Chicago, Ill.

What Reversed the Motor?

Referring to the letter in the Oct. 29 issue under the title "What Reversed the Motor?" an examination of the diagrams will show that, probably due to an oversight, the resistance and lamp are shown in one of the main leads whereas the article states they were inserted in the field circuit. The fact that the motor ran satisfactorily with the resistance in circuit and would reverse and run at a higher speed when the 50-watt lamp was substituted would show that with the lamp in series the magnetizing effect of the field winding was reduced to such a low value that the field set up by the armature reaction demagnetized it. With the brushes set with a backward lead the armature reaction would set up a field for the armature currents to react on in an opposite direction to that set up by the field winding, causing the motor to reverse.

With the brushes set with a forward lead it would continue to run in the same direction but with a motor of any considerable size the sparking would be severe. Mr. Boone will find that with his field circuit open entirely and a forward or backward lead to his brushes, by passing current through his armature alone it will revolve, although, due to the weak field, the torque developed would be very small.

If circumstances warrant the expense I would suggest that the field winding be changed to a regular shunt. The data for this purpose could be obtained as follows: With the motor connected to the line and running at normal speed and full load, measure the current with an ammeter in series. In the case of a series motor this will measure the current through the field. Obtain the number of turns in the series field. The number of turns times the current will equal the ampere-turns required in the new shunt winding. Allowing about 1200 circ. mils per ampere; the size of the wire may be readily calculated.

P. JUSTUS.

Cleveland, Ohio.

Blown Fuse

In answer to Mr. Koppel's question as to which fuse was blown under the heading of "Blown Fuse" in the Nov. 5 issue, I would say that as the middle transformer which is fed from the line wires *B* and *C* delivered its proper voltage to the motor, it must have been the fuse in line *A* which was blown.

F. S. PIPER.

Manchester, N. H.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Operating Gas Engines

By A. L. Brennan, Jr.

Although internal-combustion engines have reached a high stage of perfection as far as design and construction is concerned, still as regards their running qualities, much naturally depends upon the operator. Unless he is versed in at least the fundamental principles of these self-contained power-generating units the efficiency will be very low.

Trouble is often encountered when starting a gas engine by its failure to pick up. In such cases faulty gas mixture or ignition troubles are usually to blame. The former is sometimes caused by insufficient fuel supply, due to a valve being closed, the pipe clogged, or the vent closed. The latter may be due to an open circuit, to poor compression, to water in the cylinder due to a leak in the jacket or to moisture sucked in from the exhaust.

Low-tension ignition of the make-and-break type should give but little trouble, yet it is well to know how to trace a trouble in this circuit. To ascertain at once if the existing trouble is in the wiring or igniter, place the switch in position to close the circuit and then with a screw-driver form a circuit from the stationary electrode on the igniter to the cylinder. If a good spark results by moving the screw-driver, the circuit is complete and the trouble is in the igniter. Having traced the trouble to the igniter, it may be due to dirty points or to the actuating springs working improperly.

On the other hand, if no spark is had it will be necessary to look over the battery connections and terminal points of all the wiring, for an open circuit is in nine cases out of ten responsible, as the low-tension current is not as apt to short-circuit as in the case of the high-tension or jump-spark system.

Briefly considering ignition troubles pertaining to the high-tension system it will be found that when the engine refuses to start, stops suddenly or misses fire, it is best to trace the trouble by a process of elimination. This is accomplished by turning the engine over by hand and noting if each section of the coil buzzes. If each vibrator works all right, the trouble cannot be in the primary side of the system, that is, in the batteries, timer battery connections, primary winding of the coil or contact points on the vibrator; hence the only parts remaining are the high-tension wires and the spark plugs.

To test the high-tension wires to the plug, hold the end of the wire about $\frac{1}{16}$ in. from the plug with the engine on the firing point for that cylinder, and switch on the battery. If a good spark jumps across the gap the existing trouble has been traced to the spark plug. This may be tested by first removing the plug from the cylinder, wiring as before and placing the threaded part on the cylinder. Throw on the switch, and if no spark appears, substitute another plug for the same test to prove whether this is the true location of the trouble. Plug troubles are sometimes overcome by merely cleaning the points.

On the other hand, if upon turning the motor to the firing position no buzz is had, any one of the following causes may be looked for: A disconnected ground wire from the coil to the engine, a broken or disconnected wire in the battery box, poor contact at the timer, a broken connection inside of the coil, vibrator contact points out of adjustment or badly pitted, or the coil otherwise wet or damaged, weak batteries or a dead cell.

Again, assume that some, but not all of the sections buzz. In that case look to the contact points of the sections not buzzing, and to the wire from that section to the timer. If this does not show the trouble look for an internal defect in that section. If the units of the coil are interchangeable the last point may be positively determined by removing the faulty section and replacing it with another that is known to be in working order. Unless one is familiar with coils, however, or the conditions warrant an attempt to remedy a defective coil, the unit should be sent to the manufacturer or some other reliable place. In cases where all the sections buzz, but no spark appears at the plugs the trouble can usually be traced to a loose connection on the secondary side of the coil.

To clean a spark plug thoroughly it should be taken apart and washed with gasoline, the points being trued with a fine flat file and brightened with emery cloth. Defective insulation allowing partial short-circuits and other troubles are of a wide range, but if one becomes familiar with the general troubles the special ones will be easily found.

Coils should be adjusted when the motor is in operation, and the tremblers should be allowed only sufficient tension to produce positive ignition. Many operators have the idea that sparking efficiency can be greatly increased by considerable tension on the tremblers. However, this only to a certain extent holds true and it is found that a stiff adjustment of the contact points results in quickly running down the batteries, pitting the contact points and impairing the efficiency.

Engines operating in an irregular manner will not give high efficiency; hence it is better to shut down for a time, if possible, to remedy the trouble, if it cannot be corrected while the machine is in operation. The chief factors in the troubles of this kind are water in the carburetor, faulty mixture, gasoline feed partly choked, loose terminals, broken-down insulation on the wires, faulty contact at the coil or timer, dirty or defective spark plugs, or mechanical faults causing poor compression, such as badly pitted or broken valves.

If the cylinders fire in order, but the engine does not develop its full power, the cause may be faulty mixture, insufficient lubrication, poor compression, exhaust partly obstructed, or defects in the ignition system.

Overheating is usually due to some failure in the cooling system, although insufficient lubrication will produce the same result. If the engine is operating in a weak manner and apparently the cooling system is not at fault, the trouble may be due to a crack in the piston head or to burned gas getting past the piston because of worn rings or a scored cylinder. Either of these will cause the crank case to become hot, which will help in locating the trouble.

Hissing sounds are in nearly every instance an indication of the compressed gas escaping, which may be due to a broken spark plug, a partly blown-out gasket, open compression cocks, etc. A sucking sound is invariably the result of the intake manifold being loose. If this fault is not corrected the mixture will be weakened by the addition of air, and the engine will lose power, although it may appear to operate regularly.

Muffler explosions are frequently the result of the gas mixture being too lean to fire in the cylinder, although they may also be caused by an inefficient spark, over-retarded spark, or to pitted or stuck exhaust valves. When a motor slows down gradually accompanied by misfiring it is nearly always an indication that

the gasoline supply is at fault. The carburetor may be choked by dirt at the jet, the gasoline tank may be empty or air bound, or the gasoline valve may be partly closed. Weak batteries or dirty spark plugs will also contribute to this. Sudden stopping is nearly always due to the ignition giving out, such as a broken wire, loose terminal, broken spark plug, trembler on spark coil stuck, faulty contact at the timer, switch out of contact or no gasoline.

Explosions in the carburetor or inlet pipe are often directly attributed to a lean gas mixture or mechanical failures such as the inlet valve or spring or the valve itself getting out of time; or the spark too far retarded.

Overheated exhaust often indicates a clogged pipe or an over-retarded spark. Knocking or pounding can usually be traced to a loose flywheel, loose cylinder or crank case, loose or worn bearings or to the spark too far advanced, too rich a mixture or defective lubrication which will at first produce a squeaking before the surface is badly worn.

Undoubtedly to some, the foregoing remarks will give an impression that the gas engine is still subject to the many complications of former years. This to a certain extent holds true but, at the same time, it should be realized that the greater majority of the present-day troubles can be traced directly to carelessness or neglect on the part of the operator.

A large percentage of engine failures as far as operation is concerned can be directly attributed to over or poor lubrication of the cylinder. Too large a quantity of high-grade cylinder oil will produce the same results as the correct amount of poor oil; that is, it will gum up the piston rings, produce excessive carbon, and cause dirty points on the igniter or spark plugs. In fact, low flash point oils are little better than useless when subjected to the high temperatures of an internal-combustion motor.

These points are mentioned to lay stress upon the importance of looking carefully after the details and thus avoiding the complications which are bound to ensue. As soon as an operator realizes the importance of paying strict attention to methods others have found to be the best, his own troubles will become a thing of the past.

The Junkers Engine—V

By F. E. Junge

Pressure on Guides

To arrive at a complete understanding of the action of forces in Junkers and Diesel engines, respectively, it remains to study the pressure on the guides and the resulting pressure on the main bearings, as well as the inertia forces and tilting moments. The different pressures on the crosshead guides, resulting from those previously demonstrated are shown in Figs. 24 and 25. It is apparent that in the Junkers engine the pressure on the

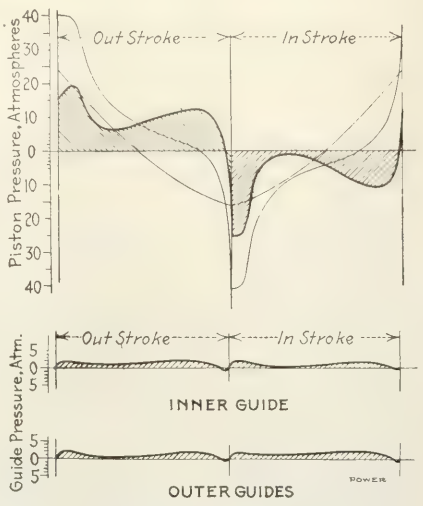


Fig. 24. Resultant Pressures in Junkers Engine

guides experiences a hardly perceptible reversal toward the end of the stroke. In the Diesel engine, however, the crosshead is lifted from its guides during the second half of the compression stroke, because of the change of direction of the pressure on the guides. For a definite rotation, therefore, wear takes place on both guides. This is much enhanced due to impact as soon as there is enough play between the crosshead and its guides. If the piston is to serve as a crosshead, there will be wear on both sides of the piston and the cylinder; both will experience about four times the wear on one side. This will lead to leakage and the loose fit of the piston will augment undue wear.

Pressure on Main Bearings

In the Junkers engine the load on the main bearing is made up of the unbalanced inertia forces, acting along the center line of the engine and the pressure on the guides, at right angles to the center line. In the following it is assumed that the weights of both mechanisms are the same; these conditions conforming with practice.

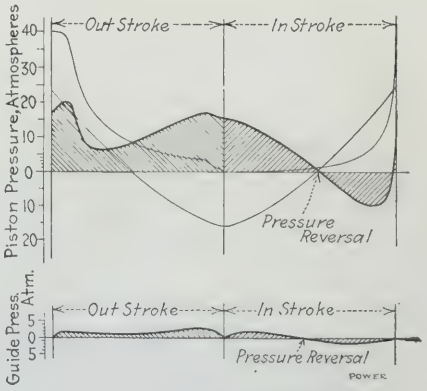


Fig. 25. Resultant Pressures in Diesel Engine

The magnitude and nature of the inertia forces are graphically shown in Fig. 26. That such forces should occur is due to the fact that the opposite motion of the pistons in the Junkers engine is not identical at all points of the stroke. The arrangement of two connecting-rods on the same side of the crank circle brings about the opposite motion of the pistons. But the inertia forces, due to the obliquity of the connecting-rod, augment one another and are responsible for the remaining unbalanced force. The inertia forces acting on both the middle and outside pistons have been plotted. It should be noted that when one set of pistons is on the forward or out stroke, the other is on the inward or back stroke. The two inertia curves are superimposed, being referred to the travel of the middle mechanism. The inertia forces neutralizing each other are shown cross-hatched in Fig. 26.

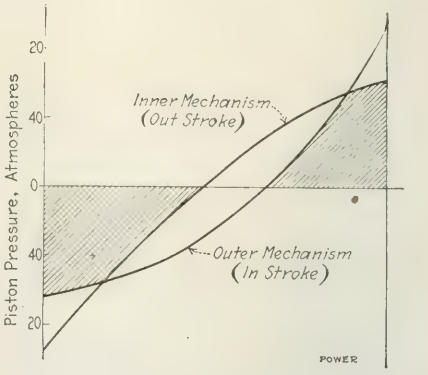


Fig. 26. Inertia Forces in Outer and Inner Mechanism

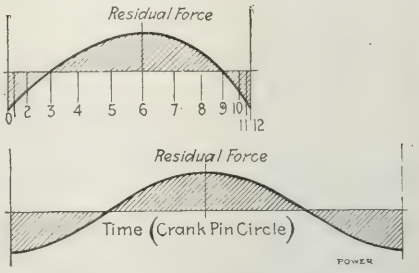


Fig. 27. Unbalanced Forces

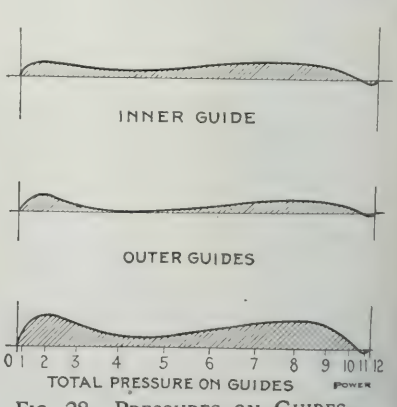


Fig. 28. Pressures on Guides

Fig. 27 again represents the unbalanced forces; in the upper diagram they are referred to the travel of the center mechanism and in the lower to the crankpin circle. As may be seen from the diagrams, the remaining force completes its cycle during one stroke. This corresponds with its character as the sum of force errors which in first approximation travel at twice the rate peculiar to the fundamental acceleration term for an infinitely long connecting-rod. With a connecting-rod of a length equal to that of five cranks, the maximum value of the unbalanced force is two-fifths of the inertia force in the dead center position with an infinitely long connecting-rod. If the latter be 295 lb. per sq.in., the maximum remaining force will attain a value of 118 lb. per sq.in., in this case.

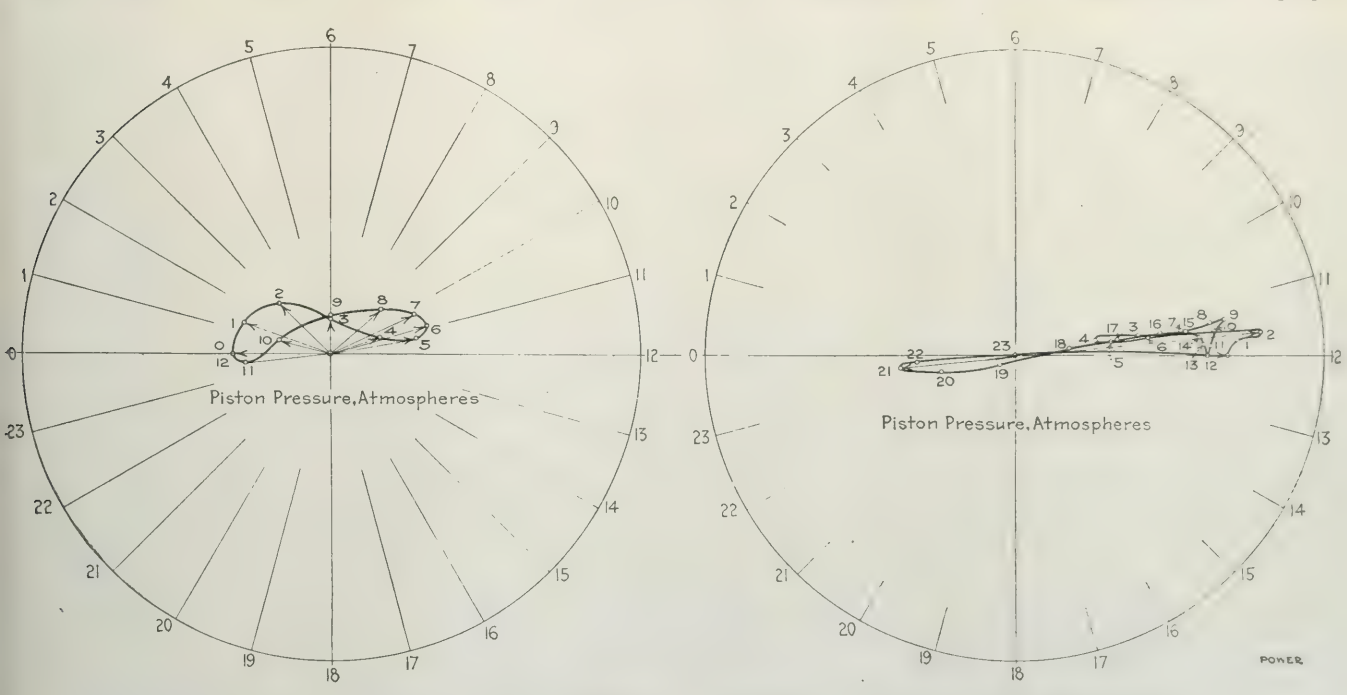
The nature of the pressure on the guides is graphically represented in Fig. 28, showing the pressures on the inner and outer guides, respectively, and the combined pressure of the two.

Finally the resulting load on the main bearing is represented in Fig. 29, for successive positions of the crank. This is simply the geometrical resultant of the corresponding unbalanced force in the direction of the cylinder axis, and of the pressure on the guides, acting at right angles to the center line of the engine, the total bearing presssure being given.

For the Diesel engine the resulting pressure on the main bearing is made up of the gas pressure on the piston, the inertia forces, coinciding in direction with the center line and the pressure on the guides, at right angles to the latter. The first named force results from a compensation of the working stresses not being effected within the crank mechanism; hence the main bearings are exposed to the full working stresses, as was discussed in Article IV.

Fig. 30 shows the corresponding load on the main bearing for corresponding positions of the crankpin. The resulting pressure on the bearing of the Diesel engine is more than that in the case of the Junkers engine. Moreover, the load on the bearing is fluctuating in the Diesel engine, whereas in the Junkers engine it is almost uniform. The changing of the loads supported by the bearings induces an unequal application of the pressure and unequal wear. The weak and rapid force undulations in the Junkers engine are totally eclipsed by the slow and strong force fluctuations in the Diesel engine. It should be further noted that when starting up, the pressures on the bearings are about twice those indicated, because the inertia forces are then not acting compensatingly.

Regarding the action of forces in an engine it is important to consider not only their effect upon the engine itself, but to reflect also upon their effects on the surroundings. This should be done quite aside from whatever the conditions set up in the engine itself by these forces may be, or how they are taken up by the separate elements, and the magnitude of the resulting stresses. In stationary engines it is generally possible to provide foundations substantial enough to cope effectively with prevailing conditions. In marine engines, however, this matter commands more detailed consideration, because of the greater complexity of the actions concerned. This is especially so when several separate units are to form one compact aggregate of machinery.



FIGS. 29 AND 30. RESULTANT BEARING PRESSURES IN JUNKERS AND DIESEL ENGINES RESPECTIVELY

CORRESPONDENCE

Producer Operation

Gas-engine users, also steam engineers, are commencing to have more respect for the gas engine than they formerly had, and realize that it requires as much skill to be a successful gas-engine operator as it does to run a steam engine. Furthermore, a producer operator requires as much knowledge as a fireman. Our greatest trouble was to get the producer operators to thoroughly poke the fuel bed and to break up clinkers. However, when they learned that thorough poking keeps the fire clear and the clinkers from forming, they took care that none formed. As long as the fuel bed is in proper condition it does not require any hard poking; also the proper adjustment of the air and steam is soon found giving a good quality of gas.

Our producers are given a thorough poking down with a heavy bar before starting, and after blowing 15 to 20 min. the fuel bed is tamped with a tamping bar, and then after blowing a few minutes we have a good quality of gas. We always keep a charge of coal in the hopper when running or idle (in our case 100 lb.), also water running through the stack valve when blowing, 15 min. after shutting down and, of course, while running. Our producers have not had the fire out in five years, they run an average of 10 hr. a day, a 100-lb. charge of pea coal being dumped on the fire according to the demand, and also upon shutting down. The ashes are removed every other day or as the fuel bed reaches a height, generally about 4 ft. thick.

Middleboro, Mass. J. W. FRIES.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Cost of Exhaust Steam Heating*

By Ira N. Evans†

"Our heating costs us nothing, when our power is operative, as we utilize exhaust steam entirely."

This statement is frequently made by owners and engineers, and it is thought by many to be universally true. That there is a saving by utilizing the latent heat of the exhaust steam after it has been expanded in an engine is unquestioned, but has the maximum power been obtained before the steam enters the heating system?

The heat represented by work done by the steam engine is only about 10 per cent. of the amount introduced into the fluid to make it available for power purposes, while 90 per cent. can be used for heating or goes to waste through the injection water of the condenser. All additional power requires 100 per cent. of the heat, while 10 per cent. only can be utilized, so that it is all the more imperative that this feature of the problem should receive close attention.

The heating system should not only require a minimum weight of steam, but be capable of utilizing the steam at the lowest temperature possible, in order that it can be expanded between wider limits of temperature and thus increase the engine efficiency. This statement holds true regardless of the type of engine, although some give better efficiency for wider ranges of temperature than others, or method of heating.

Heating and power are seldom properly correlated. In many cases the power plant is designed independently and the heating system left as an after consideration. The more uneconomical the power equipment the greater will be the apparent saving by utilizing the exhaust steam for heating, and the greater the cost of heating and power combined. The real saving is the amount of steam utilized for the power load under the most economical conditions of full vacuum. All steam additional due to the higher temperature of the exhaust is a charge against the heating system. This charge is still there even if the engines are modified and made more uneconomical to suit the heating system. A condensing engine may be operated noncondensing to suit the heating system and the power balance the heating, showing a large saving, but the cost of the heating is still the difference in the amount of steam required for operation at the final temperature conditions. Incidentally the rated capacity of the machine is reduced as the temperature range of the steam decreases.

It will be evident that hot-water heating only is applicable for operation under temperatures to any extent below atmosphere. The water system should be designed and operated, so that the maximum power will be obtained from the steam before it enters the heating system, which should be made as efficient a condenser as possible. This means the lowest temperatures of steam and water compatible with the initial cost of radiation, as the lower the temperature at which the steam is utilized the more costly the heating system. If the heating system requires high exhaust-steam temperatures or the engines are operated under conditions of lower vacuums or atmosphere, the economy of the combination is reduced.

The principle outlined applies to all exhaust-heating systems, and most forcibly to central stations and plants where there are periods of surplus peak load.

If the heating load is considerably in excess of the power load it makes little difference in the total economy whether or not economical engines are installed. The only question then involved is to select a heating system using the minimum weight of steam.

Excess heating loads are seldom found in large plants or central stations and even if they do exist, there is always the probability of the power load increasing. For the most economical results the heating load should not be greatly in excess of the power load.

In the issue of Sept. 3, 1912, the heating plant at Oak Park was described and several schedules of operation given. A schedule averaged from these and showing a maximum drop of 66 deg. in 20-deg. weather is used in this article to show the effect and advantages of doubling the pumpage and reducing the maximum drop to 32.5 deg., or to approximately one-half. The discussion does not apply to this particular plant, as it is a live-steam proposition and the mains are not arranged to work successfully with high velocities.

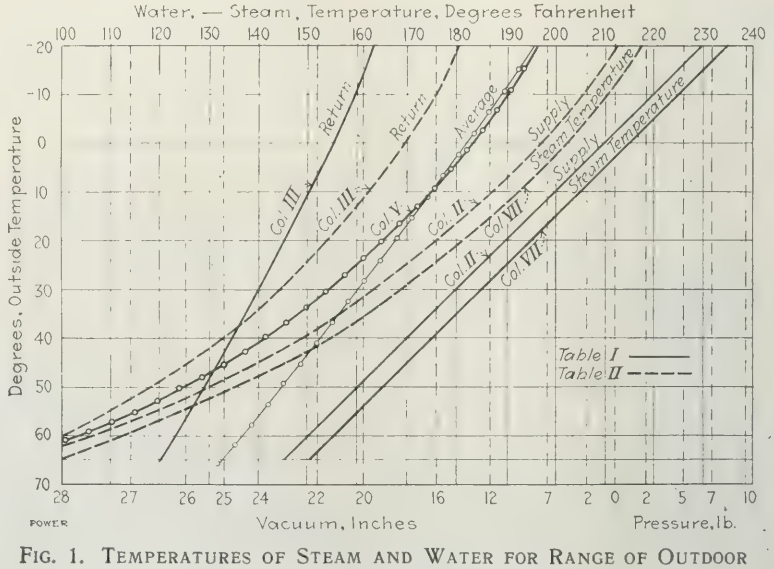


Fig. 1. Temperatures of Steam and Water for Range of Outdoor Temperatures

Tables 1 and 2 and Fig. 1 show the temperature conditions of operation with the pumpage as at present and with it doubled. Supply and return temperatures, average temperatures of water and required steam temperatures are all given in separate columns, and the same Roman numeral at the head of the column refers to the same quantity in either table and in the curves of Fig. 1. The steam temperature was assumed to be in all cases 5 deg. higher than the outboard temperature of the water. For the Oak Park plant the amount of heating surface is

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†Consulting engineer, 156 Broadway, New York City.

TABLE 1. OPERATING CONDITIONS BEFORE DOUBLING THE PUMPAGE

I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XV	XIII	XIV	XVIII	XIX	XX	XXI	XXII	XXIII	XXIV	XXV	XXVI	XXVII	XXVIII	XXIX	XXX
Outside Temperature Deg. F.	Supply Temp. Deg. F.	Return Temp. Deg. F.	Drop, Deg. F.	Average Water Temp. Deg. F.	Average Water Temp. — 70 Deg. F.	Temp. Steam in Heater	Cor. Press. or vac.	Steam Rate per kw.-hr. (500 kw. unit)	Pounds Water Circulated Per Min.	Steam per Hour Used on System (Latent heat 1000 B.t.u. per lb.)	Net Available kw (pump deducted)	Power on Pump, kw. (70% Eff.)	B.t.u. per hr. per sq.ft. per deg. dif. (570,000 sq.ft.)	Returned in Friction,. lb. Steam	Cost Pumpage 160 lb. to vac. Col. VIII Steam Rate Col. IX	Steam to Raise Feed to 212 Deg. F.	Corrected Steam Rate Col. IX+Col. XX	Cost of Heating per kw.-hr. Col. XXI—22.24	Cost of Heating Over full Vacuum Col. XV×Col. XXII	% Time Each Temperature Period	Hours Each Temp. Period	Total Cost per Season lb. Steam XXIII×XXV	Net Available kw.-hr. per season XV×XXV	Total lb. Steam for Heating and Power Col. XI×XXV	Steam per hr. at 28 in. vac. Col. XV×22.24 lb.	Total kw-hr. Pump.
20	229	163	66	196	126	234	8 lb.	49	30,810	122,008	2377	112	1.671	266	203	37	49	26.76	63,609	0.05	0.5	31,804	1,189	60,984	52,864	56
15	221	161	60	193	123	229	6	47	30,300	114,375	2324	109	1.618	257	200	39	47	24.76	57,542	0.08	0.8	46,034	1,859	91,500	51,686	
10	210	158	57	190	120	225	5	46	30,500	108,885	2258	109	1.6	257	200	40	46	23.76	53,650	0.14	1.4	75,110	3,161	152,439	50,129	23¼ hr.
5	205	156	54	188	118	220	5	43	30,500	104,310	2201	109	1.571	257	199	42	43	21.26	48,707	0.21	2.1	102,285	4,811	219,051	50,952	109 kw.
0	210	156	54	183	113	215	1	42	30,500	98,820	2866	109	1.534	257	201	42	42	19.76	44,341	0.52	5.2	230,573	11,669	513,864	49,907	2,562
5	205	153	52	179	109	210	0	41	30,500	95,160	2784	109	1.531	257	208	44	41	18.59	42,064	1.40	14	588,896	30,576	1,322,240	48,572	119 hr.
10	200	150	50	175	105	205	2 in.	40	29,900	89,700	2690	109	1.5	242	195	46	40	16.78	39,861	2.20	22	876,942	47,300	1,973,400	47,816	100 kw.
15	195	147	48	171	101	200	6	38	29,900	86,112	2497	105	1.496	242	190	48	38	16.216	35,124	4.20	42	1,475,208	90,972	3,616,704	48,172	
20	190	145	45	168	98	195	8	36	29,900	80,730	2341	105	1.445	242	187	50	36	14.372	30,800	5.50	55	1,694,000	117,865	4,440,150	47,660	11,900
25	185	141	44	163	93	190	10	35	28,100	74,184	2151	93		202	160	52	35	13.53	27,520	8.60	86	2,366,720	174,924	6,379,824	45,236	527 hr.
30	180	139	42	159	89	185	12	34	28,100	70,812	2054	93	1.4	202	162	54	34	12.68	25,335	17.40	174	4,408,290	347,652	12,321,288	44,436	85 kw.
35	173	136	39	156	86	180	14	32	28,100	65,754	1917	93	1.341	202	160	58	32	11.30	21,900	14.40	144	3,153,600	279,072	9,468,576	43,101	
40	170	134	36	152	82	175	16	31	28,100	60,696	1790	93	1.3	202	158	60	31	9.911	18,561	12.30	123	2,283,003	230,379	7,465,608	41,655	44,795
45	165	131	33.5	148	78	170	17	30.2	26,400	53,064	1539	81.4	1.215	166	131	62	30.36	8.12	15,570	10.00	100	1,557,000	168,700	5,306,400	37,519	
50	160	129	31	145	73	165	19	29	26,400	49,100	1424	81.4	1.15	166	134	64	30.36	7.22	13,179	8	80	1,054,320	129,840	3,928,000	36,069	330 hr.
55	155	126	29	141	71	160	20	28	26,400	45,936	1332	81.4	1.135	166	129	66	29.46	6.46	11,343	6	60	680,580	94,260	2,756,160	34,939	70 kw.
60	150	123	27	136.5	66.5	155	21	27	26,400	42,768	1240	81.4	1.128	166	129	68	28.54	6.30	9,538	5	50	476,900	75,700	2,138,400	33,671	23,100
65	145	120	25	132.5	62.5	150	22	26.5	26,400	39,600	1148	81.4	1.110	166	130	70	28.14	5.90	8,402	4	40	336,080	56,960	1,584,000	31,670	
						100	28	20									22.24				Totals	21,437,345	1,866,889	63,747,588		82,413

TABLE 2. OPERATING CONDITIONS AFTER DOUBLING THE PUMPAGE

I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII	XIII	XIV	XV	XVI	XVII	XIX	XX	XXI	XXII	XXIII	XXIV	XXV	XXVI	XXVII	XXVIII	XXIX	XXX
Outside Temp. Deg. F.	Supply Temp. Deg. F.	Return Temp. Deg. F.	Drop Deg. F.	Average Water Temp., deg. F.	Average Water Temp. — 70 deg. F.	Temp. Steam in Heater	Cor's'p'd Press. or Vacuum	Steam Rate Per kw-hr. (500 kw. unit)	Pounds Water Circulated Per Minute	Pounds Steam per Hour on System	Head on System, ft.	Power on Pump, kw. (70% Eff.)	B.t.u. Per Hour per sq.ft. per Deg. Dif. (510,000 sq.ft.)	Net Available kw. (Pump Deducted)	Boiler hp. at 34.5 lb.	Dif. in Total Heat 160 lb. and Col. VII	Cost Pumpage 160 lb. to vac. Col. VIII Steam Rate Col. IX	Steam to Raise Feed to 212 Deg.	Corrected Steam Rate Col. IX+Col. XX	Cost of Heating per kw. Over full Vac. Col. XXI—22.24	Cost of Heating Over full Vac. Col. XV×Col. XXII	% Time Each Temp. Period	Hours Each Temp. Period	Total Cost Season lb. Steam XXIII×XXV	Net Available kw.-hr. Per Season XV×XXV	Total lb. Steam for Power and Heating XI×XXV	Steam per hr. at 28-in. Vac. Col. XV×22.24 lb.	Kw.-hr. for Pump
—20	212.5	180	32.5	196	126	217	1.5lb.	43	61,200	119,340	100	200	1.86	2575	3460	44	379	0.2	472	20.76	53,457	0.05	0.5	26,928	1,288	59,670	57,268	200,000
—15	209	178	31	193.5	123.5	214	0.5	42	61,200	113,832	100	200	1.81	2510	3300	45	378	1.92	472	19.76	49,598	0.08	0.8	39,678	2,008	91,065	55,822	200,000
—10	206	176.5	29.5	191	121	211	0	41.5	61,200	108,324	100	200	1.76	2410	3140	46	382	1.04	472	17.96	46,417	0.14	1.4	64,984	3,374	151,654	53,598	200,000
—5	202	173.5	28.5	188	118	207	3 in.	40	61,200	104,652	100	200	1.74	2416	3033	47	376	0.2	472	16.14	43,391	0.21	2.1	91,121	5,074	219,769	53,732	200,000
0	196.5	169.5	27	183	113	202	5	38	61,200	99,144	100	200	1.72	2409	2874	49	372	0.38	472	16.14	38,881	0.52	5.2	202,191	12,527	515,549	53,576	200,000
5	192	166	26	179	109	197	8	36	61,200	95,472	100	200	1.717	2452	2767	51	367	0.54	472	14 3	35,064	1 4	14	490,896	34,328	1,336,608	54,532	200,000
10	187.5	162.5	25	175	105	193	10	35	61,200	91,800	100	200	1.714	2423	2661	53	371	1.37	472	13 425	32,529	2 2	22	715,638	53,306	2,019,600	53,888	200,000
15	181.7	158	23.5	170	100	187	12	34	61,200	86,202	100	200	1.69	2338	2501	55	374	0.85	472	12 61	29,482	4 2	42	1,238,244	98,196	3,624,264	51,997	200,000
20	176	154	22	165	95	181	14	32.5	61,200	80,784	100	200	1.67	2285	2342	58	377	1.00	472	11 26	25,779	5 5	55	1,415,095	125,675	4,443,120	50,818	200,000
25	168	148	20	158	88	173	17	30.5	61,200	73,440	100	200	1.634	2208	2129	61	372	1.116	472	9 38	20,711	8 6	86	1,781,146	189,888	6,315,840	49,106	200,000
30	163	144.5	18.5	154	84	168	18	30	61,200	67,932	100	200	1.624	2064	1969	63	378	1.29	472	9 05	18,679	17 4	174	3,250,181	359,316	11,820,168	45,903	200,000
35	157	139.5	17.5	148	78	162	20	28	61,200	64,260	100	200	1.62	2095	1863	66	370	1.37	472	7 13	14,937	14 4	144	2,150,928	301,680	9,253,440	46,593	200,000
40	150	134	16	142	72	155	21	27	61,200	58,752	100	200	1.606	1976	1703	69	373	1.54	472	6 3	12,449	12 3	123	1,527,288	243,048	7,226,496	43,946	200,000
45	140	126	14	133	63	145	23	25	61,200	51,408	100	200	1.606	1792	1490	73	377	1.73	472	5 29	9,480	10	100	948,000	179,200	5,140,800	39,854	200,000
50	130	118	12	124	54	135	25	25	61,200	44,064	100	200	1.60	1562	1277	77	374	2.93	472	4 69	7,326	8	80	757,120	124,060	3,525,120	34,739	200,000
55	117	107.5	9.5	112	42	122	26	22.5	61,200	34,884	100	200	1.591	1350	1011	83	375	2.025	472	3,085	6	60	185,100	81,000	2,093,040	30,024	200,000
60	106	99	7	102	32	111	27	21	61,200	25,784	100	200	1.58	1024	745	88	370	2.12	472	2 285	904	6	60	43,050	51,200	1,285,200	22,774	200,000
65	93	88.5	4.5	91	21	100	28	20	61,200	16,524	100	200	1.54	626	479	92	368	2.24	472	0 88	0	4	4		25,040	660,960	13,922	200,000
																							Totals	14,762,277	1,880,928	59,782,363		

given as 550,000 sq.ft. and the mains aggregate 85,000 sq.ft. of actual surface. For calculation it is assumed that the main and heating surface is equivalent to a load of 570,000 sq.ft.

An analysis of the present schedule shows that a portion of the heating surface must be shut off in moderate weather. This is indicated by the low transmission in Col. XIV, Table 1.

In Table 2 all surface will be considered in operation, the pumpage will be the same throughout and every arrangement made to operate on the lowest water temperatures. This will make the heating system so responsive to outside weather changes that automatic heat control and local regulation will be unnecessary. The drop in head was given for the Oak Park plant. From this and the average distance the writer estimated the pumpage. The discharges taken from the different heads are only relative, as undoubtedly those used for the schedule in Table 1 are higher than actually obtained on this plant, but they are near enough for the comparative purposes of the article. The intervals of temperature changes for different outside weather conditions were estimated from an actual installation where the pumpage was nearly the same per square foot as in Table 2.

The writer found a mathematical relation in the ratio of the drop to the average water temperature minus 70 deg. The pounds of water pumped per hour divided by the square feet of surface, which was 7.2 lb. in this case, multiplied by the above ratio must give the rate of transmission of the heating surface per degree per hour. Having the limits given, the intervals can be determined by trial and error. The drop depends on the pumpage and the writer had the actual rates of transmission per square foot per hour, determined from the records of the plant referred to. The actual amount of water pumped, as given in Table 1, was estimated at 30,800 lb. per min. and in Table 2 the pumpage was made 61,200 lb. per min., which will give one-half the drop and deliver the same number of heat units at the maximum average temperature. With this drop and increase in transmission due to the additional pumpage, 10 per cent. less heating surface is required, or in round numbers 510,000 instead of 570,000 sq.ft., which means a reduction of a like amount in the cost of installations to customers.

A maximum average water temperature of 196 deg. was used in both cases. It might be policy in some instances to reduce the radiation and run slightly higher water temperatures. The constant amount of radiation and the constant pumpage have to be so related to the drop that (the square feet of radiation) × (the average water temperature — 70 deg.) × (the transmission per square foot per hour) will exactly equal (the drop) × (the pumpage per hour) for each period of outside temperature. With these quantities determined the schedule can be made up as the drop and (average water temperature — 70 deg.) will give the supply and return temperatures of water and the corresponding steam temperature.

The head of 112 ft. given as the maximum differential on the Oak Park plant schedule is assumed to give with three 10-in. pipe circuits a discharge of 30,810 lb. per min., with about 100 kw. required for the pumpage. For 61,200 lb. per min. three 12-in. mains require 175 ft. head and 350 kw. for the same length of circuit. As the mains require too large a proportion of the power generated to circulate the water, the comparison has been based on three 14-in. O. D. pipes which will discharge 61,200 lb. per min. with 100 ft. head and 200 kw. See Cols. XII and XIII of Tables 1 and 2. Fig. 1 shows that the increased pumpage causes a reduction in the steam temperatures at which it may be used on the heating system of over 17 deg. for the entire heating season. See Col. VII of both tables. This comes directly from the reduced drop in temperature necessary on the system.

In Col. IX the steam rate per kilowatt-hour is given for the various vacuums on a 500-kw. turbine. To find the pounds of steam per hour, Col. XI, multiply the pumpage per hour by the drop, Col. IV, and divide by the latent heat, 1000 B.t.u. By dividing by 34.5 the boiler horsepower is found, Col. XVI. The transmission per square foot is given in Col. XIV.

The latent heat of the steam increases with the reduction in pressure. If the steam can be utilized at lower temperatures and pressures on the heating system the same number of heat units can be transmitted with a greater economy in increased kilowatt output. Columns IX give the maximum steam consumption for Tables 1 and 2 as 49 and 43 lb. per kw.-hr. with 8- and 1.5-lb. pressure above atmosphere, respectively. The maximum vacuum and minimum steam consumption in Table 1 are 23 in. and 26.5 lb. per kw.-hr. and in Table 2, 28 in. and 20 lb. per kw.-hr.

In the case of high vacuum the steam rates should be based on 212 deg. and the amount of auxiliary steam necessary to raise the steam temperature from that corresponding to the vacuum to 212 deg. should be determined. The temperatures in Col. VII are deducted from 212 deg. and the result multiplied by the steam rate in Col. IX and divided by 1000 B.t.u. This gives the correction in Col. XX, which is added to the steam rate in Col. IX, giving Col. XXI. In Col. XXII is given the actual cost of heating per kilowatt-hour. It is obtained by subtracting in each case 22.24 lb., the rate per kilowatt-hour for 28-in. vacuum corrected for 212 deg., from the rates in Col. XXI. This shows that exhaust steam for heating is obtained at considerable expense and not for nothing as is generally believed. What is actually saved is the steam at the maximum vacuum for the load carried, and the more economical the engine the less the saving will be. The cost at — 20 deg. temperature outside for Table 1, Col. XXII, is 26.76 lb. against 20.76 for Table 2, while at 65 deg. outside temperature the cost is 5.9 lb. per kw.-hr. against nothing in Table 2.

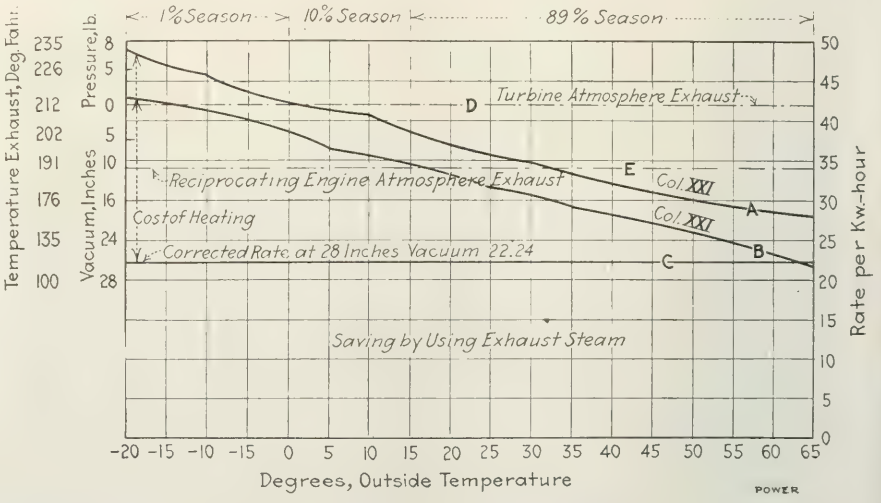

FIG. 2. RATE CURVES FOR THE SEASON

These costs are for hot-water heating under two conditions of operation, while if steam were used at 1½-lb. pressure the cost would be 20.76 lb. for each kilowatt load, whether the steam was needed or not, for the entire heating season.

The ordinates of curves *A* and *B* in Fig. 2, plotted from Col. XXI of both tables, give the corrected steam consumptions per kilowatt-hour. The percentages of time, Col. XXIV, for the different temperature periods are given and also the steam temperatures and correspond-

ing vacuums from Cols. VII and VIII. The ordinate under the line *C* gives the rate for a 28-in. vacuum as 22.24 lb. per kw.-hr., which is the actual amount of steam saved. All distances of ordinates above the line *C* are the charges per kilowatt-hour for heating due to reduced vacuum and are represented in Col. XXII of both tables. If the machine were operated at atmosphere the lines *A* and *B* would become *D* for the entire season. These steam consumptions multiplied by the load on the turbine will give the cost of heating and the total ordinate the cost of heat and power per kilowatt-hour whether the steam is all utilized or not. It may be readily seen that with a given heating load in pounds of steam that more power can be obtained from the steam rates of curve *B* than from the rates of curve *A*. The difference in rate curves is about 7 lb. per kw.-hr. for the entire season. The approximate line for the steam consumption of the reciprocating engine exhausting to atmosphere is indicated by line *E*, and line *D* gives the steam rate of a noncondensing turbine. A like chart may be made for any engine where the water rates are known for various vacuums. The temperature at which the steam is used in the heating system will determine the vacuum and the cost of utilizing the exhaust steam. The rate at full vacuum will determine the saving. The less the steam consumption at full vacuum, the greater the necessity of lowering curves *A* or *B* by operating the heating system at the lower steam temperatures. Fig. 2 will explain how owners think they are making large savings by utilizing exhaust steam on the heating and operating the engine on line *E*. They are actually saving the amount below line *C* if they are able to operate condensing at all.

With a 1000-kw., instead of a 500-kw., unit the saving due to increased pumpage would be appreciably increased on account of the greater economy of the larger machine at full vacuum. The cost of steam for the heating over the power load would also be increased.

Cols. XXIV and XXV give the percentage of time and number of hours probable for each period of operation out of a season of 1000. This is about the period of time during which the peak power load would occur with the heating in the central plant. The percentage of time each outside temperature period occurs has a marked bearing on the average rate for heating and power; that is, the average ordinate for Fig. 2 may not necessarily be the average for the season.

Assuming 70 per cent. efficiency for the pump in both cases and the same steam consumption as the main machine, the kilowatt load for the pumpage is given in Col. XIII. Against 112-ft. head the maximum power for a pumpage of 30,800 lb. is 112 kw. and 200 kw. for 61,200 lb. against 100-ft. head in three 14-in. O. D. pipes.

The actual heat units returned by friction to the water, reduced to pounds of steam, are given in Col. XVIII, Table 1, and calculated for Table 2, where the pumpage is constant, as follows:

$$\frac{61{,}200\,lb. \times 100\,ft. \times 60\,min.}{778\,ft.\text{-}lb. \times 1000\,B.t.u.} = 472\,lb.\,per\,hr.$$

The value 1000 B.t.u. has been used throughout for the latent heat per pound of steam, and, although somewhat in error, is near enough for the comparison when used in both cases.

Col. XVII of Tables 1 and 2 give the difference in total heat of the steam at 160-lb. gage and at the pressures of Col. VIII. Col. XIX gives the heat, reduced to pounds of steam, actually used in power for the circulation of the water. If the exhaust is utilized in the heating system, there will be no further loss from the pumpage. This is obtained as follows:

$$\frac{Col.\,XIII \times Col.\,IX \times Col.\,XVII}{1000\,B.t.u.} = Col.\,XIX$$

Col. XIX is less than Col. XVIII in both tables. This shows that where the available exhaust steam is less than the requirements of the heating, the pumpage, with 70 per cent. efficiency for the pump and less than 45 lb. of steam per kw.-hr., costs nothing. In the case of —20 deg. outside temperature it means for Table 1 that 5500 lb. of exhaust steam would come from the pumpage and in Table 2, 8600 lb. This is compulsory live-steam operation to that extent. The power load on the plant might be so balanced at times that it would be economical to reduce this live-steam operation by reducing the pumpage, which can be easily done with the same apparatus. At the same time reducing the pumpage increases the steam temperature of the exhaust and would increase the kilowatt rate appreciably on a large turbine fully loaded and naturally reduce its capacity.

The reduction in pumpage in Table 1 makes some of the periods slightly lower than for Table 2. This is probably caused by arbitrarily fixing the supply temperature when the operating schedule was obtained and trying to vary the heat supply with the pumpage. At the same time this plant as arranged and operated cannot be responsive to rapid changes in the radiation temperatures on account of the great difference between the supply and return temperatures.

It has been the writer's experience that there is no economy in reducing the pumpage in moderate weather, as the lower the average temperature of the surface and the smaller the drop the greater the vacuum can be carried, with a corresponding increase in power. The total heat required is also reduced. See in Fig. 3 the periods below 40 deg. outside temperature. It is possible where the vacuum is fairly high to carry considerable excess load over what would normally be required for heating on a turbine without appreciably interfering with the engine economy. This would not be possible if high-steam temperature were required for heating as the excess steam due to reduced vacuum might destroy all economy in the use of the exhaust steam.

Col. XI is divided by the steam rate in Col. IX and the kilowatts used on the pump deducted in each case to obtain the net kilowatts available for power, Col. XV. Fig. 3 is plotted with the quantities from Col. XXVI. Curves *A* and *C* are really a combination of three curves. Due to changes in the pumpage in Table 1 they become flatter and flatter as the pumpage is reduced, showing a decrease in the transmission. See Col. XIV. The sum of the squares between the full curves and the dotted curves is the cost of heating in each case. The squares

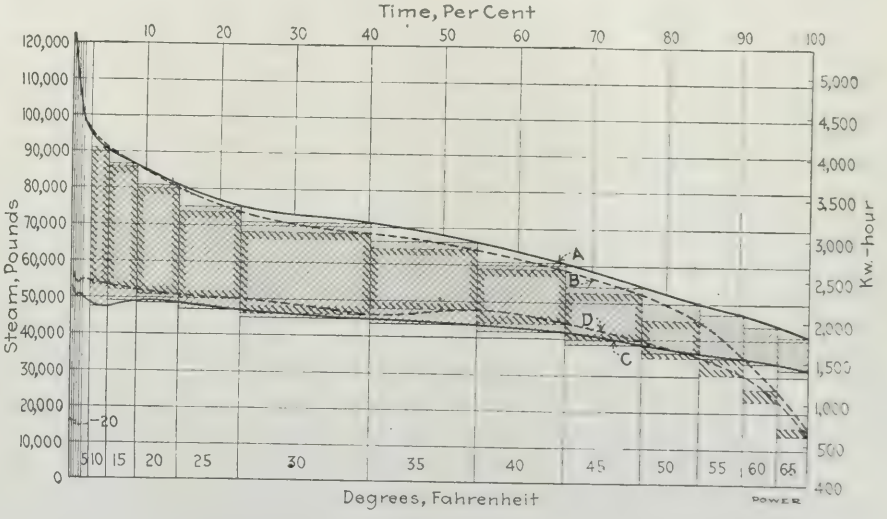

FIG. 3. POUNDS OF STEAM AND KILOWATT-HOURS FOR THE DIFFERENT TEMPERATURE PERIODS

below the curves *B* and *D* show the amount of steam actually saved, Col. XXIX of each table, and equals the sum of the kilowatt-hours in Col. XXVI multiplied by the rate at full vacuum. For Tables 1 and 2 the kilowatt-hours total up to 1,866,889 and 1,890,928 and multiplying by 22.24 gives 41,519,611 and 42,054,239 lb. of steam respectively. Thus 24,039 kw.-hr. were obtained from the steam used in Table 2 over Table 1, exclusive of the extra cost of pumpage.

The total areas for heating and power below the curves *A* and *B*, Fig. 3, amount to the totals of Col. XXVIII of both tables, or 59,782,363 lb. for Table 2 and 63,747,588 lb. for Table 1, a difference in favor of the latter of 3,965,225 lb. This is available for extra power at the rate of 22.24 lb. per kw.-hr. or 180,091 kw.-hr. The total increase in power will then be

$$180{,}091 + 24{,}039 = 204{,}130 \textit{ kw.-hr.}$$

or 10.8 per cent. increase in power by operation according to Table 2. The saving in actual steam, 3,965,225 lb., amounts to 6.22 per cent. The rate per kilowatt-hour for the power and heating, Col. XXVIII ÷ Col. XXVII is 34.146 lb. for Table 1 and 31.6 for Table 2, a difference of 2.546 lb. per kw.-hr. for the heating season of 1000 hr. This is greater than would be obtained by operating a condensing plant at 28 instead of 26 in. of vacuum.

The squares between the curves *A* and *C* of Fig. 3 for Table 1 and *D* and *B* for Table 2 show the cost of heating, or

$$63{,}747{,}588 - 41{,}519{,}611 = 22{,}227{,}977 \textit{ lb.}$$
$$59{,}782{,}363 - 42{,}054{,}239 = 17{,}728{,}124 \textit{ lb.}$$

Per kilowatt-hour this amounts to 11.91 lb. for Table 1 and 9.375 lb. for Table 2. This actually means the equivalent of live-steam operation in order to obtain the saving by the use of the exhaust steam. It is also the average ordinate for the cost of heating in Fig. 2 for the period of 1000 hr. This portion of live-steam operation in the case of Table 1 is 34.9 per cent. of the total steam used and 29.67 per cent. for Table 2. The proportion of live steam used in either case to the actual amount saved is

$$\frac{22{,}227{,}977}{41{,}519{,}611} = 53.5 \textit{ per cent.}$$

$$\frac{17{,}728{,}124}{42{,}054{,}239} = 42.15 \textit{ per cent.}$$

This shows a much larger proportion of live steam used in Table 1. The actual cost of heating on the net power obtained on the plant is shown in Col. XXVI and the totals divided by the net kilowatts gives the rate without the cost of pumping, or for Table 1, 11.483 lb. per kw.-hr. and for Table 2, 7.88 lb. per kw.-hr. These costs subtracted from the rates for total live-steam operation give the cost of pumpage per kilowatt-hour for Tables 1 and 2 as

$$11.91 - 11.483 = 0.427 \textit{ lb.}$$
$$9.375 - 7.88 = 1.495 \textit{ lb.}$$

As these rates are not much more proportionally than required to operate the pumps on a regular condensing outfit, and less than the quantities in Col. XX over 50 per cent. of the season, the author believes that the pumpage on a proper heating system and that for the condenser plant will be about the same and the true cost of heating would be, Col. XXVI, inclusive of the cost of pumping. If taken in this way there would be an added increase in power of the difference in the pumpage or

$$200{,}000 - 82{,}500 = 117{,}500 \textit{ kw.-hr.}$$

or nearly 10 per cent. additional.

The problem resolves itself into an adaptation of the heating system to perform the function of a condenser for the purpose of utilizing the latent heat with a minimum interference with the engine economy. The principles in condenser practice apply to the problem in hand.

In Fig. 3 the actual steam saved is not very much different in amount, due to the reduction in total requirements in weather below 30 deg. outside, but the net available power in Col. XV is about 10 per cent. more for the individual periods, notwithstanding the 100 kw. deducted for the extra pumpage.

Bleeder turbines and low-pressure turbines with reciprocating engines utilizing the steam at 5 lb. pressure are frequently used. This does not give any more economy or as much as an independent engine with 5 lb. back pressure for that portion of the steam used in the heating system. See Fig. 2 for the steam rate for 5 lb. pressure and the difference in the economy of the methods presented will be apparent. The economy of the heating steam for power purposes depends solely on the temperature and pressure to which it is expanded. The fact that there is a condenser beyond the bleeder turbine or the low-pressure turbine does not affect the power obtained from the heating steam in any sense. If the heating steam can be expanded beyond the 5 lb. pressure to 20 in. of vacuum, more power and more latent heat per pound will be obtained and in the case of hot water the heating system will act as a condenser, relieving the regular condenser of that much work.

This discussion relates solely to the use of turbine units for this purpose, although the same is true of the reciprocating engine in a lesser degree, and as the size of unit increases the question of decreased steam consumption on the heating becomes more important. See line *E*, Fig. 2.

If the engine consumption is increased 100 per cent., there is only 50 per cent. saving by utilizing the exhaust steam for heating. This makes it imperative with the increased economies in engine operation that the system of heating be adapted to the new conditions so as to interfere with the efficiency for power as little as possible. When engines used 40 to 60 lb. of steam per horsepower-hour, any steam system would do for the purpose without appreciable increase. With large units operating at 15 to 18 lb. per kw.-hr. on high vacuum the old methods will not do. The losses in mains are greater and the efficiency of the heating system as a condenser is less when using low pumpage.

The importance of considering the amount of power obtained from the heating steam as well as the temperature and weight of steam necessary for the heating system cannot be overestimated. If a heating system required 1000 kw. and it could be operated with 800 kw., or if 800 kw. were used at a high steam rate and 1000 kw. could be obtained from the steam, due to higher vacuum and lower temperatures of exhaust, there would be a saving of 200 kw. In a heating system this saving would be made seven months in the year.

It is important that the load balance the heating as with too heavy power load more steam will be used due to decreased vacuum than would be required if the heating were operated on live steam and the engines run under full vacuum. On the other hand, with the schedule of Table 2 the heating system can be operated on a turbine a large portion of the time with considerable excess load in moderate weather without serious reduction in economy due to the higher vacuum and lower steam temperature required. With a properly balanced load the curves in Fig. 2 show a reduction of 2 to 4 lb. net per kw.-hr. for the entire season, with 10 per cent. less radiation.

This discussion shows a saving in methods of operation on two hot-water plants without any actual increase in first cost of one over the other. If anything the arrangement indicated by Table 2 will cost less to install than that indicated by Table 1.

In time let us hope that the opening paragraph of the article may be modified to read: We are operating our engines under vacuum at all times, our heating system acts as a condenser in winter for what exhaust steam is required for heating and the cost of steam, due to reduced vacuum, is not over 40 per cent. of full vacuum conditions.

Work on the erection of the new $75,000 power plant of the Pittsburgh & Lake Erie R.R., at Dickerson Run, Penn., has been begun. The excavation work is well under way and the plant will be hurried to completion, as the railroad is in urgent need of electrical power to run the machinery of the hydraulic system and the new coaling and ash dumping. Cars will be loaded and engines coaled by machines and the ashes mechanically hoisted and dumped.

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POWER

Wanted, a Name for Steam-Stuff

The English language is deficient in a specific term for H_2O in a mixed state. When it is in the vaporous or gaseous state it is called "steam." If the temperature is above that at which water changes into steam at the existing pressure, it is "superheated steam." If its temperature is that of the boiling point of water under the existing pressure, it is called "dry saturated steam" if the last trace of moisture has been evaporated out of it, or "moist steam" if it carries some entrained moisture. For the liquid and solid forms we have the common terms "water" and "ice."

But when steam is expanded in a cylinder or a turbine it gives up a portion of its heat, and some of it consequently condenses. How shall we speak of the quantity of H_2O, which is now in question, and which is neither steam nor water, but a mixture of the two?

It is common to say, for example, the *steam* after expansion contains so and so many units per pound. What is meant is that each pound of the *mixture* contains the stated amount of heat. That part of it which is steam contains the full amount of heat per pound given in the table for the dry-saturated condition. The mixture is no more "moist steam" than it is "partly evaporated water." That part of it which is evaporated is steam and that part of it which is not evaporated is water; and that part of it which is steam carries just as many heat units per pound as does dry-saturated steam of the given temperature or pressure. What we need is a general comprehensive word which will signify the substance, H_2O, in whatever physical condition, simple or complex, it may be; some simple appropriate term which will satisfy the gropings of authors who hit upon such terms as "mixture," "steam stuff," "working fluid," "medium." etc. Other languages are just as deficient as our own in this respect, and anybody who will suggest a generally acceptable name for this quantity will fill a gap in scientific nomenclature and render an international service to engineering literature.

Cost of Exhaust Steam Heating

An article on other pages of this issue shows how more power can be obtained from the same amount of steam before utilizing it in the heating system. If the practice obtaining in many large plants is any indication, as long as the power furnishes sufficient steam for heating or the heating system utilizes all of the exhaust steam, the owner seems to think that the maximum efficiency is obtained regardless of how wasteful either the power or heating apparatus may be. Heating and power problems are seldom considered in conjunction and correlated in design. The most economical equipment for power alone is frequently installed and also what is thought to be the most economical heating plant (generally from the standpoint of installation cost and operation in zero weather) without much consideration for the economy of the combination in average weather.

To provide for outside weather changes, all heating systems require for economical operation methods of varying the quantity of heat admitted. This may be accomplished by providing automatic heat control and curtailing the time of operation if the temperature of the medium is nearly constant as in a vacuum system, or by varying the temperature of the medium itself as in the case of a hot-water system. The above requirements are independent of the temperature at which the steam is utilized. The lower the temperature the more radiation required and the greater the cost; the higher the temperature the less the radiation and first cost. As long as the steam consumption is varied through the same range the total amount of heat per season will be the same.

The efficiency of the engine or the amount of steam used per unit of power, will vary directly with the difference in

temperature of the steam entering and leaving the engine. Therefore the temperature of the exhaust absolutely fixes the economy of the engine for power, other things being equal. The latent heat is practically the same for all temperatures at which exhaust steam is used for heating, so that nearly the same heat will be obtained per pound of steam whether utilized at a high or low temperature for heating. It is impracticable to use exhaust steam for heating under final temperatures most economical for power, except in very mild weather, due to the excessive cost of installation that would be required. It therefore becomes necessary to decrease the power economy of the engine to obtain steam at temperatures high enough to bring the cost of the heating system within reasonable limits. The amount of the sacrifice in operation is the charge against the heating system.

This will be the difference in steam consumption due to the final temperature most economical for power operation, and that required by the heating system at any outside temperature period. It has nothing to do with the amount of heat required for the heating system, but does affect the steam required for power. This difference in the steam rate will apply to every kilowatt-hour taken from the engine, whether the resultant exhaust steam is utilized or not in the heating system. If the engine plant is modified to provide high temperatures for heating in winter, the economy of the summer power load is reduced accordingly.

In the article it is shown how a vacuo hot-water heating system utilizing exhaust steam between atmospheric pressure and twenty-eight inches of vacuum will allow the maximum power to be obtained from the combination. Vacuum is carried on the engine ninety-five per cent. of the time and as the weather moderates the steam consumption per kilowatt-hour is greatly reduced. The heating system is performing the function of a condenser for the power load furnishing the exhaust steam.

A vacuum steam system requires exhaust steam at atmospheric pressure throughout the entire heating season, which in the case of a turbine would so increase the steam consumption that the saving would be negligible, to say nothing of the reduction of the rated capacity of the machine.

The article in no way contradicts the general question of the saving engendered by utilizing exhaust steam, but shows how the heating can be done more economically without destroying all vacuum more than five per cent. of the time. The question is not so urgent when the power is inadequate to furnish sufficient steam, but becomes vitally important when the peak load occurs and there is a surplus. This occurs in almost every plant at times, so that a close study of the article may be worth while.

Standards That Need Standardizing

In this day of enthusiasm over the standardization of everything possible, strange it is that the engineering profession has so long overlooked the various quantities and units of measurement used in its own work.

Take the British thermal unit. It is defined as the quantity of heat required to raise the temperature of one pound of water one degree Fahrenheit at or near 39 degrees. Again it is defined as above except that the temperature used is 60 degrees. It is also defined as $^1/_{180}$ of the heat necessary to raise one pound of water from 32 to 212 degrees Fahrenheit. While the difference is small, in any exact use of the British thermal unit the particular one used must be defined to prevent misunderstanding.

Our steam tables are numerous and their variations are also numerous. The various computers have devised formulas representing more or less accurately the results of Regnault's experiments, and all three of the above definitions of the British thermal unit are used. In any exact use of the steam table it is necessary to mention the particular table used and in acceptance tests it is possible to have a test either above or below the guarantee, depending on the tables used in working up the data.

A horsepower is defined as the energy required to raise 33,000 pounds one foot per minute against the attraction of gravity. But as the attraction of gravity varies with latitude and altitude to the extent of ½ of 1 per cent., it is necessary to include the latitude and altitude in an exact definition of horsepower.

The Centennial rating of a boiler horsepower was the evaporation of 30 pounds of water, fed at 100 degrees Fahrenheit, into steam at 70 pounds per square inch. The American Society of Mechanical Engineers has adopted a boiler horsepower as the equivalent evaporation of 34.5 pounds of water per hour from and at 212 degrees. This differs from the Centennial rating only by about 5 British thermal units. Due, however, to the variation of the British thermal units and the steam tables the actual energy represented by a boiler horsepower on the basis of the present definition may vary considerably.

Practically all rating of steam engines is in pounds of water per indicated horsepower. Unless the pressure and quality of the steam are given, these ratings mean nothing. How much simpler it would be if steam engines were rated in British thermal units per indicated horsepower. Then it would be possible to compare the ratings of various engines on a more equal basis. Also in very accurate tests the rating of a steam engine would be affected by the variation in the value of the horsepower, as mentioned above.

Gebhardt, in his book on power plants, makes the following statements in regard to viscosity.

"Viscosity may be defined as the degree of fluidity or internal friction of an oil. It is sometimes called the 'body.' It is determined by a viscosimeter. There are a number of different instruments for this purpose but no recognized standard instrument or method, so that 'viscosity' conveys no meaning unless the name of the instrument, the temperature, and the amount of oil tested are given." It is hardly necessary to comment on the need for a standard in the measurement of viscosity.

Probably other examples of standards that need standardizing might be mentioned, but the above are sufficient to show the desirability of so defining them that it is not necessary to tack an appendix on every report, giving the definitions of the quantities used.

It is good to hear of one with so abounding a faith in his fellows as William B. McKinley, president of the Illinois Traction Co. who has loaned $50,000 in fifteen years to needy students.

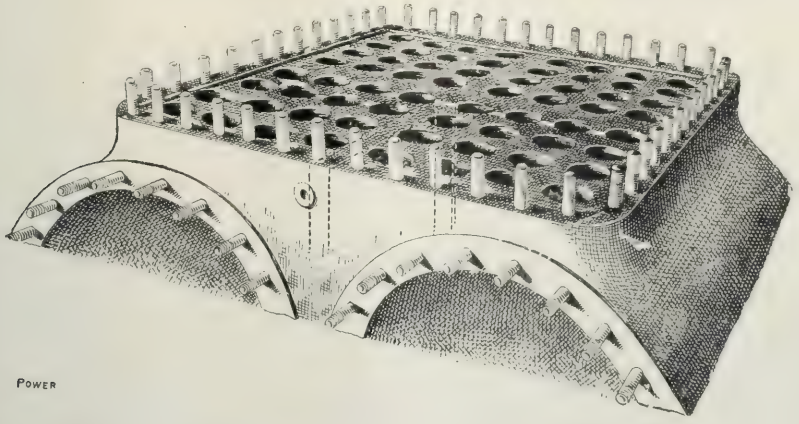
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Pump Valve Deck Badly Cracked

The illustration shows the discharge valve deck of a 16 and 24 by 13 by 18-in. duplex pump running under a discharge pressure of 120 lb. There are 128 four and one-half-inch valves; more than enough to take care of the water displaced by the plunger.

I would like to have interested readers give their opinions as to what cracked the valve deck as shown. The casting is 4½ ft. across the top and the crack



SHOWING CRACKED VALVE DECK AND PARTITION WALL OF PUMP

extends nearly the full width, being open ⅜ in. at the surface. The thick division wall in the middle of the casting is also cracked down to the suction-valve deck. The side walls or other parts of the pump show no signs of fracture or flaws. The metal is quite heavy, being from 1½ to 2 in. thick.

This is the second casting made from this pattern and both have broken in the same place. Is the break due to carelessness in molding, to an inferior metal or to expansion and contraction?

JOHN MCGINNESS.

New York City.

Feed Pipe Heavily Scaled

The two large injectors and feed pump gave trouble in supplying water to our 60-in. by 16-ft. horizontal tubular boiler. After renewing the disks in all globe and check valves and still being troubled, I decided to disconnect the feed pipe throughout its entire length to locate the obstruction, as I concluded such was the cause of the trouble.

The 1½-in. feed pipe inside the boiler has nine distributing outlets through ½ x6-in. nipples. These were so coated with scale that the openings were slightly more than $\frac{1}{16}$ in. in diameter. It was necessary to renew all the internal piping, after which the pump and injectors worked splendidly.

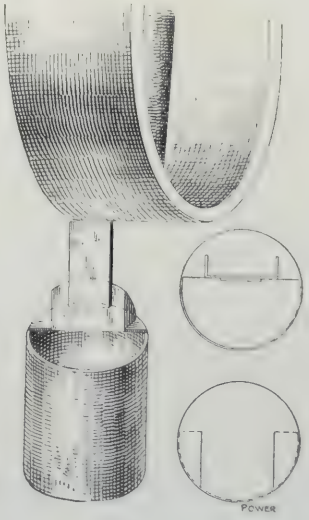
A. N. CARPENTER.

Albany, Ga.

Catching the Oil

Most engineers have been annoyed at times by oil working out on the shaft from the bearings to the hub of the flywheel and on the rim, and from there, due to centrifugal force, being thrown on the walls, floor and ceiling.

The best way to remedy this would be to prevent the oil from getting on the



CAN FOR CATCHING OIL

wheel, but sometimes the flywheel sets so close to the bearing that nothing will prevent oil reaching it, except reducing the supply until there is no excess, which is somewhat dangerous.

A simple remedy is to cut the top from a tin can, as shown by the dotted lines in the illustration, bend it as shown by the heavy lines, solder it in this position and solder a strip of tin to the lid. The can is then to be set in such a position that the tin strip will press lightly against the face of the wheel, the rim running against the end of the tin.

Such oil as reaches the wheel rim will be wiped off into the can instead of decorating the engine-room floor and wall. A large nut or other heavy article in the bottom of the can will serve to hold it steady.

If the appearance of the can is objectionable the wiper can be put on a neat pan which will answer the purpose as well.

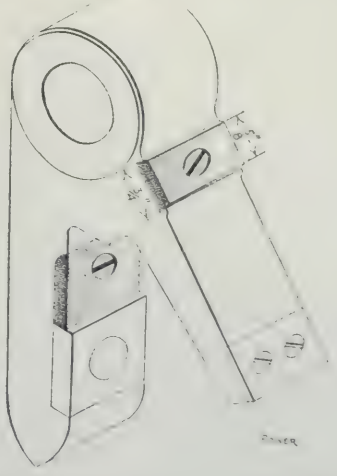
EARL PAGETT.

Coffeyville, Kan.

Silencing Noisy Grab-Hooks

For those troubled with noisy grab-hooks on Corliss valve-gear, I submit the following remedy which I have used successfully.

Where the back of the hook strikes the



GRAB-HOOK FITTED WITH LEATHERS TO PREVENT NOISY OPERATION

spring post when released by the cam, cut a dovetailed slot $\frac{3}{16}$ in. deep by ⅝ in. wide and undercut $\frac{1}{16}$ in. on each side. In the center of the slot drill and tap for a $\frac{3}{8}$x½-in. machine screw. Cut a piece of heavy sole leather to fit the slot accurately, and in this drill a hole that will be directly over the hole in the hook, when the leather is in place.

The hole in the leather should be countersunk so that the head of the screw will be about ⅛ in. below the surface of the leather. After the screw has been tightened, place the hook on its pin, and before the spring is in place, dress the leather down with a file until the plates engage to the required depth. The cushion leather on the other arm of the hook should be used as usual. The illustrations clearly show how the work should be done.

Do not wet the leather, as it will shrink when it dries, work the screw loose and cause trouble.

V. C. Eason.

Mapimi, Dgo., Mexi·

Heated Crankpin Indicator

On all the large stationary engines under my charge, I use the following telltale to indicate a hot crankpin:

On the inside and back of the oil guard, or casing around the crank disk, I have enameled the surface a pure white color, when the pin gets hot the color of the lubricating oil changes, and when it is thrown against the white surface, the engineer can immediately tell that the color of the oil has changed from normal and can at once take steps to prevent a seriously hot crankpin or shut down the engine if it becomes necessary.

A. Gordman.

Swissvale, Penn.

Corrosion of Crosshead Bore

Will some interested reader give his opinion regarding the cause of the following trouble?

Less than two years ago the threads of the piston rod and crosshead of our engine were found to be eaten away. A new rod with a larger thread was made and the crosshead rebored. The same thing has happened again and I am desirous of finding the cause. The corrosion is most pronounced in the middle of the threaded section of the crosshead.

The threaded end of the crosshead is made like a split nut with the two parts about ⅛ in. apart, which would allow oil and steam to blow from the packing sleeve into it.

This engine ran for about 16 years before this trouble developed and it seems as though acid in the oil must cause the trouble.

H. K. Wilson.

New Bedford, Mass.

Substitute for Vacuum Cleaner

We have a substitute for a vacuum cleaner that others may have at hand and not realize. As we have no air compressor to clean the generators, we had to wipe them out the best we could by hand. One day a small piece of the packing in the flange joint, which connects our draft tube to the fan-wheel case blew out. We tried to plug it with white lead, but the suction was so great it would draw it through. We finally stopped it with a piece of shingle.

We decided we could use that suction to draw the dirt out of the generators, so drilled and tapped a hole in the tube for a 1-in. pipe, put on a 1-in. gate valve and reduced down to ¾ in. so as to use a ¾-in. hose. It is surprising the way it will pick up the dirt. The tube is 18 in. in diameter, 15 ft. long and water sealed. All we bought were two 1-in. nipples, one 1-in. gate valve, one 1-in. to ¾-in. reducer and the hose. It will pick up pieces of waste, or most anything that will go through a ¾-in. opening. The generators are 25 ft. from the hose attachment at the gate valve.

V. C. Wood.

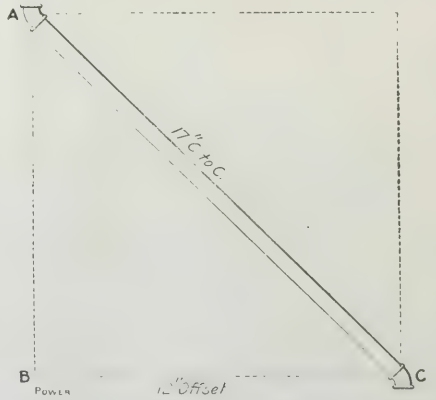
Copenhagen, N. Y.

Measuring Offset Distances Between 45-deg. Fittings

The rule given herewith is one that I find a great help. It is unnecessary with this method to hold the pipes in position or to lay the fittings on a bench, after one has learned this rule. Anyone who knows the multiplication tables can understand the explanation. The rule is an old one but not so generally understood as it should be:

Rule: Muliply the actual offset in inches by 17 and divide the product by 12; subtract whatever is necessary for the fittings and the result is the correct length required in inches.

Cut the pipe that length. If the distance from *C* to *B* is exactly 12 in. and



Illustrating Measurement of Offset Distances Between 45-deg. Ells

the 45-deg. ells are placed as shown in the drawing, the distance from *A* to *C* must be 17 in., center to center, of the 45-deg. ells, and that is where the 17 comes from in the rule.

The offset is to be understood as the difference in position, from center to center, of any two desired runs of pipe. In the drawing there is a 12-in. difference, or a 12-in. offset. The length of the pipe required to properly make the desired offset with two 45-deg. ells would be 17 in. from center to center of the ells "made up." Take out for the two ells and the remaining distance is the exact length to cut the pipe.

Eugene Oster.

Cincinnati, Ohio.

Chain Sling for Removing Cylinder Heads

The illustrations show a chain sling I have used with success for removing cylinder heads.

Fig. 1 shows the sling when ready to put around the edge of the cylinder head.

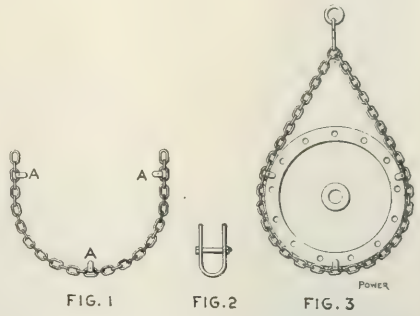


FIG. 1 FIG. 2 FIG. 3

Chain Sling and Clips for Removing Cylinder Heads

Fig. 2 is a piece of flat iron bent U-shape and fastened to a link of the chain with a bolt. Three such pieces are used as shown at *AAA*, Fig. 1.

To use this sling, the nuts are taken off the studs and then the head is pryed forward sufficient to allow the three U-shaped pieces to slip over the edge of the cylinder head. Then by proper tension on the sling, the head can be taken off easily without injury to the threads of the stud bolts. Fig. 3 shows the application of the sling.

James McClure.

Fletcher, N. C.

Test of Cumberland Coal and Screenings Mixed

The following gives the results of a test made in my plant to determine what mixture of the above fuels is the most economical. Clean unmixed Cumberland was first tried; then ⅔ Cumberland with ⅓ hard-coal screenings, and then ⅓ Cumberland with ⅔ screenings. All possible data were recorded. The plant consisted of three 72-in. horizontal return-tubular boilers, one 400-hp. Corliss double-eccentric engine and two other engines of 40 hp. each. The economy effected by screenings mixed with Cumberland coal is a much disputed subject. Some claim there is no economy in mixing the coal, while others use nothing else but ½ to ⅓ screenings mixed with Cumberland coal.

The Cumberland coal used for these

tests cost $3.25 per short ton; the screenings $2. The weight of water evaporated per dollar's worth of fuel is 6485 lb. for pure Cumberland; 6110 lb. for ⅔ Cumberland and ⅓ screenings; 6738 lb. for ⅔ screenings and ⅓ Cumberland.

From my experience I favor about ⅓ screenings mixed with ⅔ Cumberland.

I once had charge of a plant that had two 72-in. boilers burning four tons of coal every ten hours, and I did my own firing with straight coal. I suggested to my superintendent that if he would let me have a fireman, and burn the Cumberland mixed with ⅓ screenings that I could pay the fireman good wages and give my services to repair work. This was done, making things better all

TEST RESULTS ON BOILER BURNING MIXED CUMBERLAND COAL AND SCREENINGS

	Cumberland	⅔ Cumberland ⅓ screenings	⅓ Cumberland ⅔ screenings
Weather..	Oct. 4 fair	Oct. 5 rain	Oct. 6 fair
Time.....	hr. min 11—16	hr. min 11—20	hr. min. 11—25
Boiler pressure, aver. lb.....	76.72	75.42	76.18
Feedwater temperature, aver. deg.F....	205	206	207
Tank water temperature, aver. deg.F....	168	177	179
Coal fired, lb.....	4770	5950	6553
Moisture-total lb....	73	60 100	33 222
Ash and refuse, lb....	570	788	1024
Ash, per cent....	11.1	12.6	14.9
Combustible, lb.....	4359	5223	5510
Steam, quality per cent	0.98	0.98	0.98
Water evaporated total lb.....	48,776	51,881	51,828
Water evaporated per lb. coal, lb.....	9.89	8.63	7.93
Water actually evaporated corrected for priming, lb.....	48,190	51,260	51 215
Equivalent evaporation into dry steam from and at 212F., lb.....	50,260	53,360	53,300
Equivalent evaporation per lb. coal, lb	10.30	8.88	8.16
Equivalent evaporation per lb. combustible.....	11.53	10.45	9.67
Coal burned per hr. per sq.ft. grate, lb.....	6.87	8.34	9.14
Damp. flue gas, aver. deg.F.....	336	332	357
Draft, natural, inches..	½		

around, giving me more time for plant upkeep, which would have been impossible had I used mixed coal, because of the increased work of mixing and handling the extra ashes.

A. C. WALDRON.

Revere, Mass.

An Ingenious Measuring Device

Often in a drawing it is necessary to divide a line into three, four or more equal parts. Unless the distance to be divided is a number of units divisible by the requisite number of divisions the operation is anything but an exact one ordinarily. To overcome this all that is necessary is a small rubber band of uniform width and thickness. Suppose that the distance is $4\frac{4}{10}$ in. and it is required to divide it into ten equal parts.

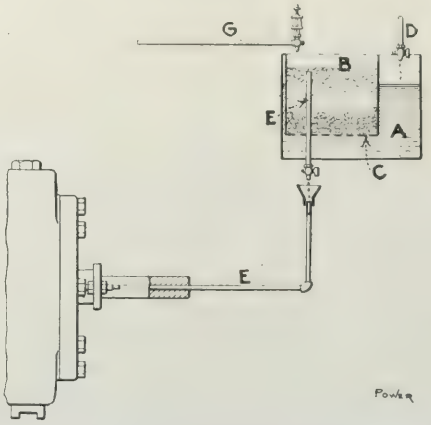
Take a rubber band and mark on it ten spaces of a quarter inch each. Now stretch the rubber until one part of the scale is on one side of the distance to be divided and the other part of the scale is on the other end. Now point off on the piece to be divided, the several stretched divisions as indicated by the rubber. The rubber stretches uniformly, so each division stretches the same until each one is exactly one-tenth of the space between.

A. L. HODGES.

Charleston, S. C.

Soapsuds Lubricator for Air Compressor

The illustration shows a soapsuds lubricator for feeding soapsuds to an air compressor compressing air to 90 or 100 lb. gage pressure in one stage. The temperature of the air after compression is frequently very high, which results in carbon being deposited on the discharge valves and passages, and trouble is therefore experienced in properly lubricating the air cylinder, especially where compressors are run at high speeds and the temperature of the intake air exceeds 80 deg., or where an inferior grade of oil is used. To overcome this trouble, soap-



SOAPSUDS LUBRICATOR FOR AIR COMPRESSOR

suds have frequently been used as a lubricant with good results.

The lubricator consists of a small galvanized-iron water tank *A*, about 7x6x12 in. Inside of it a smaller one *B* is soldered to the large one with its bottom 2 in. above the bottom of the large tank. In the bottom of the small tank are a number of ⅛-in. holes at *C*. Soft brown soap, cut into small pieces, is put into tank *B*. Water is then fed into the large tank through a ¼-in. pipe *D*, and passing through the small holes at *C* dissolves the soap and rises to the top of the ¼-in. pipe *E*, through which it passes down into the inlet of the compressor. The water is regulated by the valve in pipe *D*, the small cock under the tank being left wide open.

Just before shutting down the compressor the water is turned off at *D*, and oil is fed into the small tank *B* from a cup *G*, and the compressor is run with oil for a half hour to prevent rust formation. Soap is used that will sink in the water; otherwise small chips will float and pass down through the pipe *E*. This lubricator is used on a 14x18-in. compressor, and when running at 150 r.p.m. with an air pressure of 90 lb., the water is fed at about 45 drops a minute.

MARTIN MCGERRY.

Holley, N. Y.

Beaded vs Unbeaded Tubes

As this is a subject frequently discussed by engineers, I beg leave to offer a word on it. The beaded end is to be preferred to the unbeaded, though I am satisfied that the latter is more immune from burning off, etc., probably explained from the fact that the structure of the metal is undisturbed. Areas in the other instance it becomes weakened.

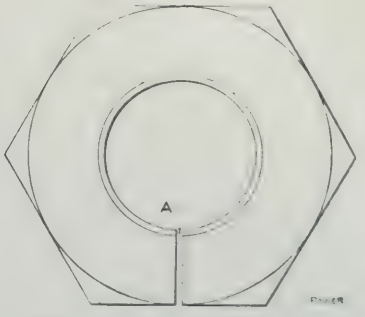
As I see it, all the value that can be attached to the beaded over the unbeaded end is that the bead stiffens the end very materially, and when it is once set up hard and tight it is not so likely to spring away from the head, due to contraction and expansion. If it was left unbeaded there would be more likelihood of its leaking, as sometimes happens when the ends receive a too severe blast of cold air.

LLOYD V. BEETS.

Nashville, Tenn.

Emergency Repair for Stripped Thread Bolt

Sometimes the thread on a bolt or a nut will get stripped; no other is on hand, and a temporary job must be done in a few minutes.



SPLIT NUT FOR USE ON BOLTS WITH STRIPPED THREADS

With a hacksaw cut a slit in the nut as at *A*. If it is a large nut, a few heavy blows with a hammer will bring it together. A small nut can be brought together in a vise. It is surprising what a strain can be placed on the nut without forcing it open at the cut.

JAMES E. NOBLE.

London Junction, Ont., Canada.

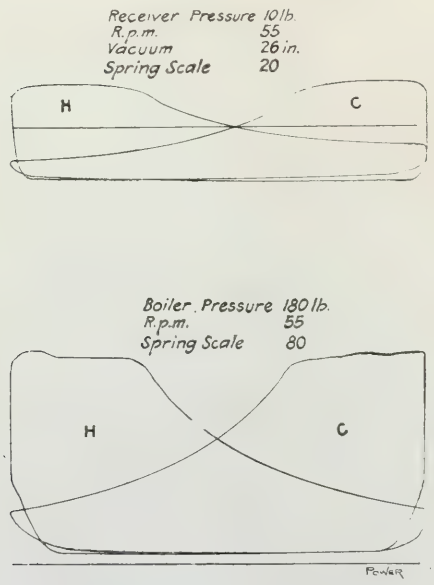
Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Shorter Cutoff, Greater Load

This subject, as POWER has treated it, has greatly interested me. As I am in no way proficient on the subject I will not offer an argument, but I wish some interested reader would criticise the two indicator diagrams which were taken from a 14 and 28 by 42-in. tandem compound Corliss engine, directly connected to a 70,000-gal. rotary pump.

These cards were taken while the engine and pump were working against an 8-ft. head, but when we have to raise the water against a 9-ft. head, the engine will not carry the load unless I lengthen the cutoff on the low-pressure cylinder, thus reducing the receiver pressure from



DIAGRAMS FROM 14 AND 28x42-IN. TANDEM-COMPOUND CORLISS ENGINE

10 to 6 or 7 lb. according to the load. I have occasionally tried manipulating the cutoff on the low-pressure cylinder to raise the receiver pressure to 18 or 25 lb., which I have no trouble in doing, but in this case the high-pressure valves do not trip regularly, but drag, thus making the engine race so badly as to raise and lower the governor so that the safety cams operate, stopping the engine.

We have another unit of the same size and working under the same conditions, which will carry the load at 8 or 9 ft. head and 14 lb. receiver pressure without any trouble. I would be pleased to hear from someone who could tell me how to remedy the trouble without raising the boiler pressure. Surely it can be done, as this engine has always carried the load with ease the same as the other one does, at 14 lb. receiver pressure, but for some reason it will not now.

THEODORE PLACETTE.

Rosedale, Tex.

Friendliness and Familiarity

I read the editorials, "Know your Employees" (July 16) and "Friendliness and Familiarity" (Sept. 24) with pleasure, as they are in accord with my practice, that is, get close to the men. The editorial did not appeal to me as it apparently did to Mr. Pierce.

All of us are men and should be treated as such. Simply because one is chief engineer, is it necessary or right for him to keep constantly before the men the fact that he is chief, and to impress upon them that they are subordinates, inferiors, etc.? The fireman may perhaps be a college graduate doing such work to learn the practical side of the business. In this case one would be "entertaining angels unawares." Such actions toward men, seem to me like the very essence of snobbishness.

It is said that "familiarity breeds contempt"; it may, but I have never yet found the familiarity as it exists in the engine and boiler room breeding contempt for the chief, if he is a true man. But I have found a universal contempt for a *martinet* (an arrogant disciplinarian). I have found it a good thing to get as close to those under me as conditions will allow, make them feel that they are an important part of the plant. By taking this course the men become interested in their work. More intelligent work can be had of them by this means, than by "building a fence" between by acts of snobbishness.

I want the men's ideas. Many good ideas have been obtained from the fireman and even the coal passer. I know that discipline must be maintained, but to do so it is better to make the men feel that they are men worthy of recognition. As the great Napoleon let his men know that they were constantly under his eye and that he noted their bravery, so let the men under us know that their work is constantly observed and the good work duly appreciated.

Mr. Pierce says: "When a manager, superintendent or even chief engineer gets the habit of being familiar or allowing the men under him to become familiar, then the wheels of good management start to clog, and in time a new crew, or new head man will become necessary." This is not so unless the manager is utterly ignorant of the characteristics of men.

As an example of snobbery and "pigheadedness," take the following:

A young man, chief engineer of a large plant, insisted that the men, every morning, when reporting for duty should touch their caps and say "good morning chief," and when anyone was present, address him as chief. This young man did not last long, even the management could not tolerate such a snob.

F. C. B. SPEARE.

Cape May City, N. J.

What Broke the Stop Valve?

On page 463 of the Sept. 24 issue, Mr. Keil asks the opinion of POWER readers as to what caused a stop valve to break. I believe it was due to water-hammer. I do not think that closing the valve on boiler No. 1 had anything to do with the accident to the valve on No. 2 boiler.

With the drain pipe connected to the boiler-feed line, as shown in Mr. Keil's sketch, I cannot see how the water could drain from the pipe between the valve and the header, as there is more pressure on the feed line, due to friction than on the boiler.

Mr. Keil states that the pressure on all the boilers was equal at the time of the accident. That being so, the instant valve No. 2 was opened the flow of water was in the direction of boiler No. 2, because the pressure in the pipe, due to the head of water, was greater than that in No. 2 boiler. The water flowing toward No. 2 might therefore have caused a damaging water-hammer against the valve. If the pressure on No. 2 had been enough greater than that in the header to cause the steam to flow into the header when the valve on No. 2 was opened, I do not think the accident would have occurred, as there would have been no cause for water-hammer.

When cutting a boiler into a header, it is well to have the pressure on the boiler a little higher than that on the header.

S. F. FARLEY.

Galveston, Tex.

Air Compressor Efficiencies

In the article by E. M. Ivens in the Oct. 15 issue it is stated that the mechanical efficiency of an air compressor should include the horsepower equivalent of the heat put into the jacket water added to the indicated horsepower of the air cylinder. In the first place, it may be mathematically shown that all of the power of an air compressor goes into heat. After air has been compressed and then cooled back to its original temperature at constant pressure, it contains no more energy than it did before. It is possible, by figuring air weights and temperatures due to compression, to show that the air horsepower exactly equals the heat appearing in the air, if it is not cooled. If, now, the air is partly cooled by the jackets, a portion of this heat goes into the water jacket; but the total heat, represented by the temperature and weight of air and temperature and weight of water, is still equal to the actual though reduced indicated horsepower, barring radiation of course. Carrying this still further, if the jackets could perfectly cool the air during compression there would still be the indicated horsepower of isothermal compression and then there would be all the horsepower in the jacket water, which under these theoretical circumstances would be equal to the isothermal horsepower itself. This would give twice the isothermal horsepower as the actual power developed, which would result in some delightfully high mechanical efficiencies something like 110 or 120 per cent., allowing for the heat still left in the air and the mechanical friction.

To say that the horsepower in the heat of the water jackets should be added to the indicated horsepower of the air cylinder is as outrageous as to say in a prony brake test that the horsepower of the cooling water should be added to that calculated by the pressure of the brake arm. The heat is the result of the work done and equal to it in both cases, so that their sum would give twice the horsepower actually developed.

In the third column on page 579, referring to Fig. 2, it is stated that the observed volumetric efficiency is $ad \div bd$. In the first place this gives another case of 110 per cent. efficiency, because ad is longer than bd; but even if this is written backward, it would not be any better because the true volumetric efficiency as shown by the indicator diagram is $bc \div ad$. Here bc represents the actual amount of air admitted to the cylinder, at atmospheric pressure. The length bd would not represent anything real, unless to show the clearance loss only. The length dc is as clearly a loss as ab and sometimes, with some kinds of inlet valves, dc far exceeds ab.

To blame the compressor for the kind of air furnished it, as Mr. Ivens does, when he says volumetric efficiency depends upon temperature and pressure of intake, is like upbraiding a man for eating poor food when he is too poor to buy a better quality. The volumetric efficiency of a compressor is the ratio of its actual output in cubic feet referred back to the temperature and pressure of the actual air furnished to it. This is just as logical as to credit a steam engine with the heat passing away in the condensate. If anybody defined, as Mr. Ivens claims, the volumetric efficiency as referring to 14.7 lb. absolute pressure and 60 deg. F., the definer took a good deal upon himself, for one might say such atmospheric conditions occur about once a year at some places, and never at all at others.

As a matter of fact, altitude would really affect a single-stage compressor somewhat, because a given gage pressure at an altitude means a greater number of compressions than at sea level and so a greater clearance loss by expansion. A compound machine would be affected hardly at all, because the number of compressions in the first stage, where the volumetric efficiency is produced, depends upon the cylinder ratio, which in a given machine is fixed. The ratio of compression in the high-pressure cylinder would be slightly greater, due to altitude, and so its efficiency would be slightly less, tending to increase the effective cylinder ratio, and so slightly reducing the efficiency of the low-pressure cylinder. This effect, however, is small and entirely unrelated to Mr. Ivens' remarks, as he simply charges the compressor because the atmosphere is below 14.7 absolute at high altitude. Obviously this is unfair.

As to heating of the air through the compressor-intake passages, this is chargeable against the machine because of the design, but this heating is impossible to measure, after the air is actually in the cylinder, so it does not apply to the elaborate formulas appearing in the article in question.

By the way, what is the use of burdening an article with such formulas, even if they do happen to be right? Several of the expressions given could be more easily solved by simply measuring the card lengths and dividing one by the other. The proceedings of the American Society of Mechanical Engineers had a staggering collection of just such useless formulas on gas compression not long ago, many of which were simply ratios of card lengths.

I have used the system of measuring air, as depicted in Fig. 6, quite extensively, but it is not so accurate as it at first appears. The trouble is to obtain the final, or even the initial, temperature of the air in tank C. Owing to the pressure, a thermometer well is necessary, and there is considerable lag in temperature rise. I have used a thermocouple, but have some doubts of this also. If a cooling coil is inserted between tanks B and C, the air going to tank C is certainly cool and this temperature is not hard to measure; but some able pneumatic engineers claim that the air in C will become heated again due to the compression of the increasing pressure within itself. This looks to me like perpetual motion, because this would be adding heat and so energy, to tank C, and there is no place for this to come from except the motor driving the compressor. This second tank C should then increase the power required to drive the machine, but it does not because the power of the motor must be the same, with or without C, as long as the regulating valve keeps the discharge pressure in B constant. The energy cannot come from air in B because its pressure and temperature remain constant, and if the heat is removed by a cooling coil between B and C, the energy put into it by the compressor motor is expended in this cooling coil. I rather think that this two-tank method, if a cooling coil is used between the two tanks is a good one and commercially accurate.

After all his mathematical discussion, Mr. Ivens asks the use of the term "volumetric efficiency" if it depends only upon clearance. There is no use for it if it is figured his way, because he has entirely neglected the loss at the "toe" of the card. This latter item is a matter of inlet-valve throttling, producing a pressure below atmosphere at the commencement of the stroke. With proper mechanical valves and suitable intake, this pressure may equal atmosphere or actually exceed it a little, when air inertia is allowed to act, just as in the setting of inlet valves for modern automobile engines.

Besides all this, however, Mr. Ivens has said but little about slip, except in scored cylinders and with leaky valves. There is as much, and more, to learn about this item as there was, and still is, to learn about the discrepancy between steam consumption by the indicator diagram and by the condenser.

R. S. BAYARD.

New York City.

Steam Gage Problem

In answer to the inquiry of Angelo Belmonte regarding his defective steam gage, I would say that possibly the spring has lost its elasticity. Then, again, I think he will find that the small gears that mesh to produce the movement of the hand have become so worn that their movement is impeded. I would suggest that he examine the gage again to ascertain if this is not the reason the gage registers a wrong pressure.

I once knew of a steam gage that would work all right up to 100 lb. pressure, but the hand would stay there un-

til the pressure rose to 110 lb. when it would jump from the 100-lb. to the 110-lb. mark. As the safety valve opened at 110 lb. the fireman was somewhat surprised when the gage first acted this way, although it had been working badly for some time. It would register 100 lb. and then the pointer would gradually move on until 110 lb. was registered.

The gage was taken apart and it was found that where the gears meshed the constant vibration of the pointer had caused a shoulder to be worn on two of the teeth. This shoulder caused the gears to bind at 100 lb., but when the pressure rose to 110 lb. it was enough to overcome the resistance offered by the shoulders. The shoulders were filed off and the teeth smoothed up and when the gage was put back it worked all right.

LEON L. POLLARD.

Fairfield, Maine.

Mr. Wylde's Troublesome Injector

Relative to E. S. Wylde's troublesome injector as mentioned in the Sept. 24 issue, I offer the following: As the injector itself has been tested and found all right, it is logical to suppose that something in the manner of connecting it up is wrong. In the first place, the steam supply to the injector should not be taken from the header, for unless the header is of ample capacity, the flow of steam from it to the engines or other apparatus will be detrimental to the proper working of the injector. Instead, the injector steam-supply pipe should be connected directly to each boiler through a system of valves and pipes, so that steam can be taken from any boiler, and that the other boilers may be shut off from the injector at will.

Another feature of the connections as illustrated in Mr. Wylde's letter which I believe to be bad, is connecting the injector discharge pipe into the pump discharge pipe through a tee. If the injector must discharge into the pump discharge, then it should do so through a Y-fitting. Possibly these two changes may cause the injector to work as it should. If not, then the suction pipe should be increased to 2 in. diameter and the discharge pipe made one size larger than it is at present.

CHARLES J. MASON.

Scranton, Penn.

Reading about E. S. Wylde's injector trouble reminded me of a similar one I was called upon to remedy.

The plant was a small one, just erected, and the injector refused to work. I noticed that the piping was all too small, and after replacing the piping with that of the proper size, I tried the injector, and found to my surprise that it would not work. I next examined it and found the steam nozzle out of line with the combining tube. I took it back to the dealer, who argued that it was all right, until I showed him the trouble. He gave me another of the same make, which has been doing good work ever since, and that was about four years ago.

The injector acted exactly like Mr. Wylde's, that is, it would prime, but not force. I suppose it could not get velocity enough to open the check valve against the pressure, so when the tubes were out of line, the water all came out of the overflow.

PERRY BARTRAM.

Cedarpoint, Ill.

I have recently had trouble with an injector similar to that mentioned by E. S. Wylde. After trying repeatedly I was unable to get water into the boiler with this injector, so I started the feed pump, but without success. I then took the injector apart and found it almost choked with small fragments of packing from the city main pump, these I removed. After replacing the injector it furnished water for the boiler without further trouble.

I informed a friend of this remedy, who had me look over his plant to see why his injector, which had recently been purchased, would not supply the boiler with water under the pressure for which it was recommended. I took the injector apart and found it so heavily scaled that the proper amount of steam could not enter the nozzle. When cleaned this injector worked satisfactorily.

The manager then asked me to look over a new injector in a small lighting plant in a neighboring town. This injector never worked and they had to run cold water into the boiler with a feed pump, or let the steam run down to 30 lb., and put the lights out. Upon examining it I found it in good condition. I then decided to trace the feed line to the boiler, and later entered the boiler and found a cold chisel in the mouth of the injector discharge line so badly scaled that it almost closed the opening. When the boiler was steamed up again the injector was tried and it worked well. If Mr. Wylde will see that the delivery tube is clean, the check valves in working order, the feed pipe to the boiler clear, the injector nozzles clear and repeat some of the foregoing processes, he will remove his difficulty.

J. E. STROTHER.

Yuma, Ariz.

Emergency Repair of Air Compressor

Mr. Bayard's communication on page 689 of the Nov. 5 issue has been read with much interest. The writer's original description of this emergency repair job in the issue of Sept. 17 was, perhaps, not so complete as might have been desired.

The cross-compound compressor referred to was a 20 and 8 and 12 by 18-in. and not an 8-in. stroke machine. The 10x10-in. compressor was a duplex machine.

The high-pressure cylinder of the cross-compound compressor having been wrecked in starting up, by water coming from a split intercooler tube, and the 10x10-in. duplex machine being of insufficient capacity to give the required amount of air for the operation of the plant, the discharge from the low-pressure cylinder of the cross-compound compressor was piped into the suction of the duplex machine.

The only reason for limiting the pressure on the suction of the duplex machine to 25 lb. was, that it was not considered advisable to carry a pressure higher than 25 lb. in the suction base or bedplate in which the suction chamber was comprehended.

The arrangement as outlined, supplied sufficient air for the requirements of the plant, and apparently did not strain the small compressor, and prevented a costly shutdown; which was the end sought.

Mr. Bayard's analysis of the problem is unusually interesting and instructive, but, on a breakdown job, there is little time for elaborate analysis or calculation. The important thing is to "get going."

W. M. FLEMING.

Holyoke, Mass.

Excess Air Indicated by Superheat

On page 268 of the Aug. 20 issue the editorial on "Excess Air Indicated by Superheat" is interesting. While excess air causes a reduction of furnace temperature, reducing the amount of heat transmitted to the water in the first pass, it is not apparent why the temperature of the gases should not be reduced to the same point through the first pass, as if the furnace temperature was higher, unless the velocity of the gases, on account of their greater volume, would be increased so much that insufficient time would be available for heat transfer.

Would it not be reasonable to suppose or assume that the increase in superheat with larger amounts of air is due to the fact that excess air, reducing and wasting furnace temperatures, reduces the capacity of the boiler, and therefore the amount of steam passing through the superheater is less? With the velocity reduced and with the same transfer of heat, the temperature of the lesser steam volume would be greater.

It would be interesting to learn if the evaporation at times of high superheat and excess air was less than when opposite conditions prevailed.

SOREN ANDERSON.

Butte, Mont.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Piston Valve

What advantage has the piston valve over the common D-slide valve?

E. D.

The piston valve is balanced and can be operated with very much less friction than the ordinary slide valve, and most forms have the advantage of being easier to construct and repair.

Flash Point of Oils

What is meant by the flash point of an oil?

E. D.

The flash point is the temperature at which an oil gives off inflammable gases. Oils whose specific gravities are below 0.85 generally have flash points below 60 deg. F. while those whose specific gravities exceed 0.85 usually have flash points above 60 deg. F.

Closing Cracks in Boiler Settings

What is a good method of stopping cracks in the walls of boiler settings?

J. H.

Fill the cracks to a depth of 2 or 3 in. with asbestos wool, leaving a crevice at the surface of the wall about ¼ in. deep and fill with a putty made of pulverized lime and linseed oil having a small proportion of asbestos wool mixed through the putty for a binder. The elasticity of the asbestos material and plasticity of the putty will cause those materials to remain much longer in position to keep the cracks sealed than a pointing of ordinary lime or cement mortar, which after setting, soon becomes crumbled from expansion and contraction of the brickwork.

Volume of Air per Pound of Coal

In ordinary steam-power plants, what volume of air is required for combustion per pound of coal?

N. P.

Theoretically about 12 lb. of air per pound of coal is required, varying according to the proportions of the combustible ingredients. With natural draft and ordinary furnaces, about 24 lb. of air is required per pound of coal to secure perfect combustion; and with artificial blast and some kinds of coal the amount of air may be only a small percentage in excess of the theoretical amount required for complete combustion.

A cubic foot of air at 60 deg. F. and under atmospheric pressure at sea level, weighs about 536 gr. and 13.06 cu.ft. weigh 1 lb. Therefore under the atmospheric conditions stated the necessary air for complete combustion of a pound of coal is about $12 \times 13.06 = 156.72$ cu.ft., but with ordinary furnaces and natural draft about twice that quantity, i.e., about 313 cu.ft. is usually required, and with artificial blast from about 160 to about 300 cu.ft. of air per pound of coal.

Distribution of Draft in Flues

How may the uniformity of distribution of furnace gases among the fire tubes of a return-tubular boiler be determined?

M. F.

Considering the products of combustion to be of uniform temperature at their entrance to the flues at the rear tube sheet, the relative temperatures of gases emerging from the flues may be regarded as indicative both of relative draft and relative efficiencies of the flues. The temperatures of emerging gases may be determined approximately by placing in the tubes near their ends small pieces of substances of known temperatures of fusion as follows:

Spermaceti	120 deg. F.
White wax	154 deg. F.
Sulphur	239 deg. F.
Tin	449 deg. F.
Bismuth	518 deg. F.
Lead	621 deg. F.
Zinc	786 deg. F.
Antimony	1166 deg. F.
Aluminum	1218 deg. F.

Low Boiler Pressure vs. Reduced High Pressure

Is it more economical to operate a low-pressure steam-heating plant by generating steam at the pressure required for use, or to carry a higher boiler pressure and use the steam reduced to the same low pressure through a pressure-reducing valve?

A. B.

The amount of heat delivered by the boiler to the point where the reducing valve discharges would be the same in either case. But in carrying higher boiler pressure more heat would be radiated from the boiler and the piping up to the reducing valve, as their temperatures would be higher. Hence if the additional heat so radiated were not utilized there would be loss of economy on this account, due to carrying higher boiler pressure. An additional loss would arise if carrying higher pressure required using a boiler-feed pump, and there also would be some loss with higher boiler pressure due to a falling off of boiler efficiency on account of less efficient absorption of furnace heat by a higher boiler temperature.

Oil Fuel vs. Coal

What are the advantages and disadvantages of petroleum as fuel compared with coal?

H. A. J.

The advantages of petroleum are: (1) Much lower cost for handling, dispensing with cost of stoking and removing ashes. (2) Less space occupied for equal heat value, and storage possible at a considerable distance if desired. (3) Instant regulation of the intensity of combustion to conform to the demand for steam. (4) Increased boiler efficiency and capacity and greater elimination of smoke by keeping furnace doors closed. (5) Less labor required around a plant. (6) Comparatively no dust, dirt or smoke. The principal disadvantages are: (1) Serious conditions imposed by insurance or municipal regulations respecting location of oil tanks, and (2) increase in boiler-repair bill from overheating and burning of boilers by formation of "steam pockets" due to intense heat acting on boiler parts not protected by rapid circulation.

Exhaust Steam Returns

Will low-pressure heating coils, fitted with automatic air valves and supplied with exhaust steam, be more effective in warming a building, if the returns are collected in a closed receiver, or if the returns are run dry to the receiver and the latter is provided with a vapor connection left open to the atmosphere?

W. H. N.

A vapor outlet from the receiver to the atmosphere would make the coils more effective as it would better insure removal of air and a more positive circulation of steam in the heating system. The vapor connection should be provided, however, with a stop valve for partly closing off the escape as might be found necessary to prevent a high velocity and short-circuiting of steam in the mains past the heating surfaces.

Engineers' Study Course—IV

Decimal Fractions

In engineering calculations involving fractions the problems are usually worked out in decimal fractions, because they are more easily handled than vulgar fractions.

We know that one figure as 1, standing alone is one—is a unit. If we add one cipher to the right as 10, we increase the value of 1 ten times; another cipher added would increase the value of 10 ten times or to 100, etc. Every cipher or *place* as it is called added to a given figure increases the value of the preceding figure or figures by 10.

1. 10. 100. 1000. 10,000

Notice that the ciphers are at the left of the period (.), and that every cipher added at the left increased the value of the figure by 10.

Now suppose we carry out this operation still further by turning these figures end for end as,

1000.
100.
10.
1.
0.1
0.01
0.001
0.0001

Just as we increased the value by 10 for every cipher put between the 1 and the period, we decrease its value by 10 for every cipher put between the period and the 1. The period, by the way, is called a *decimal point.* The 1 at the right of the decimal point then is a *tenth* because its value is only $\frac{1}{10}$ that of one, and the expression 0.1 is one-tenth. Continuing to read down the column we say,

0.1 = *tenth*
0.01 = *hundredth*
0.001 = *thousandth*
0.0001 = *ten-thousandth*

There are two chief things to remember about decimals. One is that the number of places at the right of the decimal point equals *the number of ciphers in the denominator of the same fraction expressed vulgarly,* as

$0.1 = \frac{1}{10} = 1$ *tenth*
$0.01 = \frac{1}{100} = 1$ *hundreth*
$0.001 = \frac{1}{1000} = 1$ *thousandth*
$0.0001 = \frac{1}{10000} = 1$ *ten-thousandth*

The other is that the decimal point *affects only the numbers at the right of it,* for any number at the left is a whole number.

Thus,

$8.01 = 8\frac{1}{100}$, $266.266 = 266\frac{266}{1000}$

Whole numbers and decimals as, 2.5, 3.05 and 4.25 are mixed numbers the same as though expressed $2\frac{5}{10}$, $3\frac{5}{100}$ and $4\frac{25}{100}$.

Inserting a cipher between the decimal point and the decimal divides the decimal by 10.

$0.3 = \frac{3}{10}$; $\frac{3}{10} \div 10 = \frac{3}{100} = 0.03$

The opposite of this is true if we take away a cipher from between the decimal point and the decimal; that is, the decimal will be multiplied by 10.

The value of a decimal is not changed by adding or suppressing ciphers at the right of the decimal.

$0.5 = \frac{5}{10}$; $0.50 = \frac{50}{100}$ and $\frac{5}{10} = \frac{50}{100}$

It is plain the value is the same because

$0.5 = \frac{5}{10} = \frac{1}{2} = 0.5$ and $0.50 = \frac{50}{100} = \frac{1}{2} = 0.5$

Thus you will know whenever you see the expression 0.50 that it does not equal any more nor less than 0.5. This applies to all like expressions.

But, *putting a cipher between a decimal and a decimal point divides the decimal by* 10.

Using the same figures as above and inserting a cipher as mentioned.

$0.5 = \frac{5}{10}$; $0.05 = \frac{5}{100}$

We learned that the denominators of decimals multiplied themselves by 10 for each decimal place so,

$0.5 = \frac{5}{10}$; $\frac{5}{10} \times \frac{1}{10} = \frac{5}{100} = 0.05$

The opposite of this is true if we take a cipher from between the decimal point and the decimal, that is,

Suppressing a cipher between a decimal point and a decimal multiplies the decimal by 10.

Using the same figures as above,

$0.05 = \frac{5}{100}$; $\frac{5}{100} \times 10 = \frac{5}{10} = 0.5$

Addition of Decimals

In addition of decimals, whole numbers *must* be kept at the *left* and decimals at the *right* of the decimal point.

Therefore the only difference between adding whole numbers and decimals is the position of the decimal point.

Whole numbers	Decimals	Mixed numbers
6226	0.6226	4.6226
81	0.81	10.81
948	0.948	2.948
7255	2.3806	18.3806

Notice the decimal point continues down in a straight line and that the decimal is close up to the point.

Example: The following number of gallons of oil were used in one month: 2, 11.86, 5.65, 6.225. How many gallons were used altogether?

2.0
11.86
5.65
6.225
25.735 gal.

Rule: *Place the numbers so that the decimal points are directly under one another. Add as with whole numbers, putting the decimal point in the sum under the points above it.*

Subtraction of Decimals

The position of the decimal points and decimals is the same for subtraction as for addition of decimals.

Example: 0.8864 — 0.252

Minuend 0.8864
Subtrahend 0.252
Difference 0.6344

Example: A bunker contained 667.22 tons of coal; during the week 18.5, 26.56, 30.1 and 50.25 tons were used, how many tons remained?

18.5 + 26.56 + 30.1 + 50.25 = 125.41 *tons used*

then

667.22
125.41
541.81 tons remained

Example: A tank contained 22 gal. of oil; 0.225 gal. were drawn out. How many gallons remained?

22.000
0.225
21.775 gal. remained

Rule: *Place the figures and decimal points as in addition of decimals and subtract as with whole numbers, putting the decimal point in the remainder directly under the points in the minuend and subtrahend.*

Multiplication of Decimals

No attention is paid to the position of the decimal point in multiplication of decimals *until* the *product* is found. We place the numbers as in multiplication of whole numbers and proceed to multiply as in multiplication of whole numbers. Then point off as many places in the *product* as there are decimal places in the multiplicand and multiplier combined.

Example: 0.25 × 0.60

0.25
0.60
0 1500

There are four decimal places in all, so there must be four places in the product.

Example: A boiler furnace burned 1125.25 lb. of coal per hr. How much was burned in 4 hr.?

1125.25
4
4501.00 lb.

There are only two decimal places required in the product because the multiplicand and multiplier combined have only two places. The two ciphers represent zero or nothing so they may be dropped.

Example: Multiply 0.464 by 0.002

$$\begin{array}{r} 0.464 \\ 0.002 \\ \hline 0.000928 \end{array}$$

In this example we get only 928 after multiplying by the 2. We cannot put the decimal before the 9 because there are six decimal places in the multiplicand and multiplier combined. To get six places we prefix enough ciphers to the 9 to give the decimal places called for. Rule: *Put the multiplier under the multiplicand. Do not consider the decimal points. Multiply as with whole numbers and point off as many decimal places in the product as there are places in the multiplicand and multiplier; if necessary, prefix ciphers.*

Division of Decimals

In placing the figures in division of decimals we pay no attention to the decimal points until the quotient is found. But, *we must have as many decimal places in the dividend as there are decimal places in the divisor, even though we add ciphers to obtain them.* Then divide as in whole numbers. Now to obtain the correct number of decimal places in the quotient, we must *subtract the number of decimal places in the divisor from the number of places in the dividend; the number of places remaining is the correct number of places to point off in the quotient.*

Example: 0.625 is to be divided by 25.

divisor dividend quotient

$$\begin{array}{r} 25 \;)\; 0.625 \;(\; 0.025 \text{ ans.} \\ 50 \\ \hline 125 \\ 125 \\ \hline \end{array}$$

There were 3 decimal places in the dividend and none in the divisor, consequently one cipher must be prefixed to give the correct answer.

Example: A boiler evaporated 805.18 lb. of water with 93.5 lb. of coal. What was the evaporation per pound of coal?

$$\begin{array}{r} 93.5 \;)\; 805.180 \;(\; 8.61+\text{lb.} \\ 748\ 0 \\ \hline 57\ 18 \\ 56\ 10 \\ \hline 1080 \\ 935 \\ \hline 145 \end{array}$$

In this example there are three decimal places in the dividend and one in the divisor, a difference of 2, so we point off 2 places in the quotient. It will have been noticed that a cipher was added at the right of the dividend. This, as we learned, does not change the value of the fraction, and in this case it was done that it might be added to the 108 remainder so as to carry the fraction out further than the original dividend allowed. We could have carried out the division to any number of decimal places desired by adding a cipher for each place. In carrying out the division in this way point off when all the original figures in the dividend have been used. After doing this we need pay no further attention to the original dividend. When it is not desired to carry the decimal any further than 2, 3 or 4 places, if it does not come out complete, adding a + sign to the quotient indicates that the decimal could have been carried further.

When the divisor contains more decimal places than the dividend, add enough ciphers to the dividend to make them equal.

Example: Divide 10 by 0.0005.

There are four decimal places in the divisor and none in the dividend, so by adding to make the number of decimal places equal in the divisor and dividend we have:

$$\begin{array}{r} 0.0005 \;)\; 10.0000 \\ \hline 20000 \end{array}$$

We cannot subtract any decimal places from the dividend because the number of places in both the divisor and dividend are equal; therefore there is no remainder and no decimal places are needed in the quotient.

Reduction of Fractions to Decimals and Vice Versa

Example: Express ½ decimally.

$$\begin{array}{r} 2 \;)\; 1.0 \\ \hline 0.5 \quad 0.5 = \frac{1}{2} \text{ ans.} \end{array}$$

Example: Express ¾ decimally.

$$\begin{array}{r} 4 \;)\; 3.00 \\ \hline 0.75 \quad 0.75 = \frac{3}{4} \text{ ans.} \end{array}$$

Rule: *Add ciphers to the numerator and divide by the denominator. Point off as many decimal places as there are ciphers added.*

To reduce a decimal to a fraction.

Example: $0.75 = \frac{75}{100} = \frac{3}{4}$

Rule: *Under the decimal, place a one with as many ciphers as there are figures in the decimal, then reduce to lower terms.*

Suppose we wanted to express 0.576 as a fraction having any given denominator, say 64. Then

$$0.576 \times \tfrac{64}{64} = \frac{36.864}{64} \text{ or approximately } \tfrac{37}{64} \text{ ans.}$$

Rule: *Multiply the decimal by a proper fraction having the desired denominator, and the same numerator so that the value of the fraction is unity.*

Examples for Practice

1. Express decimally, one-tenth; nine-hundredths; ninety-five-thousandths; three and nineteen-hundred and fifteen ten-thousands; one hundred and twenty-three and nine-hundredths.

2. Add together, 1010.24; 2.222; 19.005; 0.0026; 51.2; 3.1416.

3. The total weight of water pumped into a boiler was 1029.56 lb. The pump was stopped and 350.05 lb. of the water was evaporated. How much remained?

4. A pump delivers 4.59 gal. per stroke and makes 42 strokes per minute. How many gallons would be pumped in 219.5 min.?

5. An engine showed a steam consumption of 27.5 lb. per hp. per hr. If the engine consumed 15,125 lb. in one hour and 9.5 lb. of steam were generated per pound of coal, how many horsepower did the engine develop and how many pounds of coal were used per horsepower per hour?

Answers to Last Week's Questions

(1) $2409\frac{4}{25}$ lb.
(2) $20{,}390\frac{2}{8}$ lb.
(3) $2790\frac{5}{36}$ lb.
(4) 4.
(5) 100.
(6) 24.

NEW PUBLICATION

FARM GAS ENGINES. By H. R. Brate. Published by the Gas Engine Publishing Co., Cincinnati, 1912. Size, 5x7 in.; 195 pages, illustrated, cloth. Price, $1.

The large number of gasoline engines now employed for driving tractors, pumps, etc., as well as farm-lighting plants, seemed to warrant the publishing of a book devoted to the special needs of the farmer in this line. With this in view, the text has been prepared to cover in an elementary manner the principles of the gas engine, its operation and its care; since most of the common troubles may be traced to ignorance or indifference on the part of the operator.

The author has followed the usual custon of beginning the book with an early history of the gas engine. The next two chapters are devoted to explanations of the two-stroke and four-stroke cycles. Chapters 4 and 5 deal with fuels and carburetion and in this connection considerable space is devoted to the use of kerosene in gasoline engines. Ignition, lubrication and cooling are next taken up, followed by a very practical chapter on farm-lighting plants. The book concludes with a list of the common gas-engine troubles with directions for locating and remedying them.

While written primarily for the use of the farmer, the suggestions apply equally to marine, automobile and all medium-sized stationary gas and gasoline engines.

BOOKS RECEIVED

MODERN ORGANIZATION. By Charles DeLano Hine. "The Engineering Magazine," New York. Cloth; 110 pages, 5x7½ in.; indexed.

COMMERCIAL ENGINEERING. By Alfred J. Liversedge. Emmott & Co., Manchester, England. Cloth; 369 pages, 5½x9 in.; indexed. Price, 7/6.

HOW TO READ A DRAWING. By Vincent C. Getty. J. B. Lippincott Co., Philadelphia, Penn. Cloth; 64 pages, 6x9¾ in.; fully illustrated. Price, $1.

Largest Steam Turbines in New England

There are now being operated in the New England states, five steam turbines, each of 15,000 kw. capacity, a total of 75,000 kw. These are the largest in New England. Two of these machines are operated by the Edison Electric Illuminating Co., two by the Boston Elevated Co., both of Boston, Mass., and one by the Rhode Island Co., Providence, R. I. These are illustrated in Figs. 1, 2 and 3.

These units are of the same design and a description of one will answer for all. The turbines are six-stage Curtis machines. The four in Boston have Worthington base, surface condensers, each having approximately 22,500 sq.ft. of cooling surface. The centrifugal, volute circulating pumps are each driven by a vertical Atlantic steam engine at a variable speed of from 150 to 200 r.p.m. The speed depends upon the temperature of the cooling water, which varies from about 30 to 70 deg. F.

These condensers are connected to a

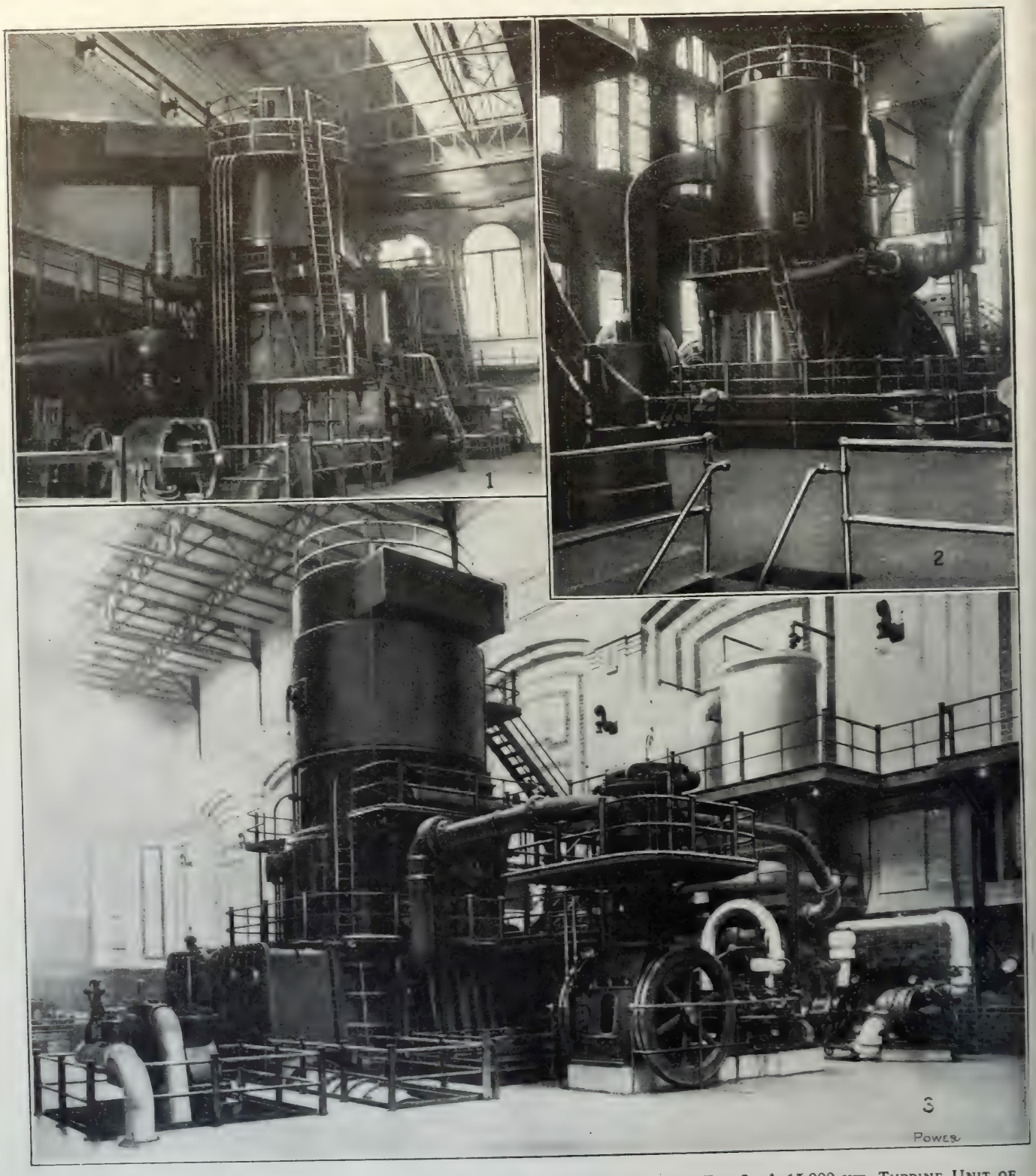


FIG. 1. TWO 15,000-KW. TURBINES IN THE PLANT OF THE BOSTON ELEVATED CO. FIG. 2. A 15,000-KW. TURBINE UNIT OF THE RHODE ISLAND CO. FIG. 3. ONE OF THE TWO 15,000-KW. TURBINES OF THE BOSTON EDISON ILLUMINATING CO.

12 and 31 by 24-in. combination horizontal and vertical Laidlaw-Dunn-Gordon single-cylinder reciprocating dry-vacuum pump and a centrifugal 5-in. wet-vacuum pump directly connected to a 26-hp. Curtis turbine. Water to the step bearings is delivered by a 14 and 3½ by 18-in. duplex pump. A Goulds triplex gear-driven pump takes care of the oiling system, and is driven by a 7½-hp., 550-volt induction motor.

Each turbine is directly connected to a 10-pole, 7000-volt, revolving-field, 60-cycle, three-phase General Electric alternator.

The Providence turbine runs at 750 r.p.m. and is directly connected to an 11,000-volt, 788-ampere, three-phase, 60-cycle alternating-current generator. It is connected to a No. 20 double Leblanc condenser in the basement. The two water-circulating centrifugal pumps are driven by the same turbine coupled to one end of the shaft. This turbine has taken over the entire load of the station that was formerly carried by cross-compound reciprocating units.

Some Old Engines in New England

Several old engines, each having an interesting history, are still being operated and doing good work. Some of them have been in use over 50 years.

New England has been the acknowledged textile manufacturing center of the United States since "long before the war." Scattered here and there may still be found in operation many steam engines that have driven the machinery in individual manufacturing plants since their incorporation, some of them having over 50 years of service to their credit. In some instances all record as to age has been lost, but the fact that they are still in operation demonstrates how well and of what good material these engines of earlier days were built.

Two of these old-time engines are operated every working day at the works of O. B. North & Co., New Haven, Conn. One is a 12x36-in. Corliss, running at 57 r.p.m., shown in Fig. 1. It develops 45 hp. with 80-lb. steam pressure. The engine shaft carries a 9-ft. flywheel, belted to the factory lineshaft.

This engine has an interesting history. It is one of the first George H. Corliss engines built by the Foundry & Machine Co., Taunton, Mass. It carries nameplate No. 53 and was built under the patent of 1851. Originally it was installed at a lumber mill, where it passed through a fire undamaged. It was purchased by the present owner in 1865 from the original owner for $50. Ten years ago the engine passed through a second fire, and was then removed to a vacant lot, where it stayed for about one year, during which period the brasses and other easily removable parts were detached by covetous junk men.

It was then repaired at a cost of $300, missing parts being replaced and worn ones renewed. Although it is known to have been in operation for more than 50 years, the engine runs noncondensing as smoothly as many newer and costlier machines.

In the same engine room is a 12x36-in. Greene engine, Fig. 2, built by the Pacific Iron Works, Bridgeport, Conn., in 1855. It runs at 65 r.p.m. and carries an average load of 28 hp. The frame is of the solid-box type, with the crosshead guides bolted to the crank side of the frame.

There are two eccentrics. The exhaust valves are operated by a rock shaft that extends through the frame to the exhaust valve-gear under the cylinder. The tappet steam valves are operated by a fixed eccentric; the point of cutoff is controlled by a governor supported by a bracket clamped to the steam pipe about 6 ft. above the steam chest. The governor is driven by a round belt from a groove pulley on the engine shaft. The cast-iron throttle valve is made square with a brass seat and disk. The engine has been in operation 57 years and is run every working day.

The Greene engine shown in Fig. 3 is of practically the same design as that in operation at the North company's plant. It was built by the Providence Steam Engine Co., Providence, R. I., has an 18x48-in. cylinder, and has developed 438 hp. with 125 lb. steam pressure, running condensing at 102 r.p.m. Although installed in 1870 it has been in charge of but three men during its 42 years of

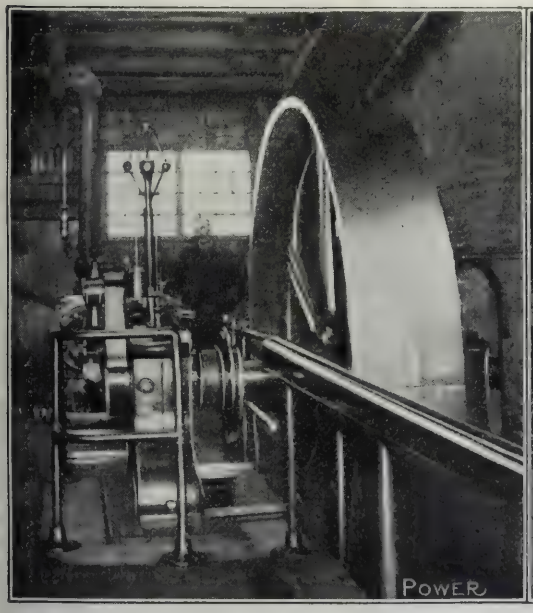


FIG. 1. OLD GEORGE H. CORLISS ENGINE

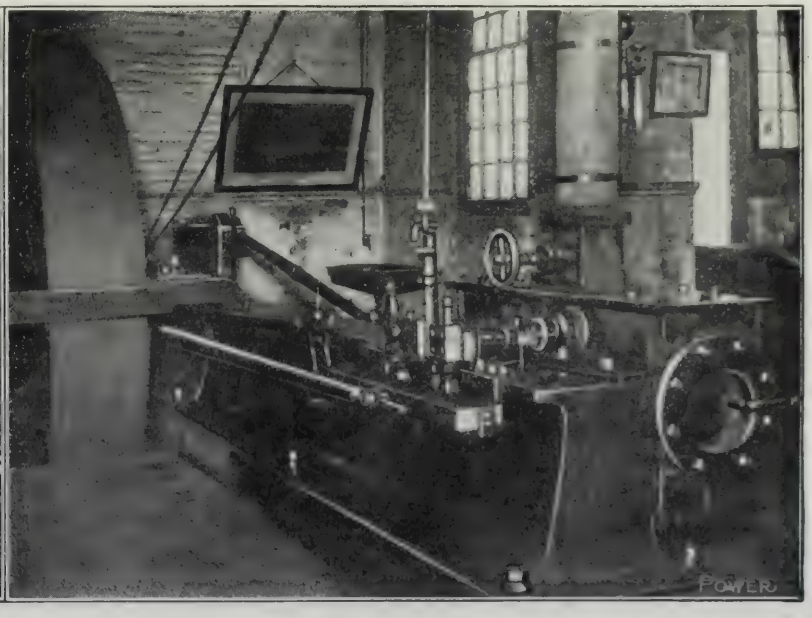


FIG. 2. OLD GREEN ENGINE

service. Luther B. Miller is the present engineer. The governor is steadied by weights on a weight yoke and by a spring dashpot.

The original speed of the engine was 55 r.p.m. This was increased to 67 in 1890 and to 102 r.p.m. in 1900. The original steam pressure was 85 lb. In 1900 the engine was developing 150 hp. Now, 12 years later, the average load is 300 hp., running condensing. This engine is operated by the Dexter Yarn Co., Pawtucket, R. I.

Another old engine with a history, but not now used, is still in the old engine room of the George H. Gilbert Co., Ware, Mass. It is of 50 hp. capacity, and was the first steam engine to be operated in that town. It was built in 1854, or 58 years ago, but has not been in service since 1880, or 22 years, although the driving belt was left in place up to within a year ago. It was built by Corliss & Nightingale, Providence, R. I. The crankshaft is cast iron, and the valves are closed by iron weights. The cylinder is 12x42 in. and the engine ran at 45 r.p.m. It was used to drive the original mill of 48 looms and employing 80 hands. There are now 476 looms and 1600 hands are employed. Unfortunately, the photograph taken was so defective that it is impossible to illustrate the engine.

In Fig. 4 is shown the steam side of an old engine running at the New Haven Copper Co., Seymour, Conn. It is of the walking-beam type, has a 42-in. by 7-ft. cylinder and runs at 22 r.p.m. It was built by Woodruff & Beach, Hartford, Conn., in either 1854 or 1855, or 58 years ago. It is constructed much after the design of most walking-beam engines, with an A-frame to support the walking-beam, a crankshaft supported by a solid-box bed on one side of the center frame and a vertical cylinder on the other. The governor hangs from an arm secured to the side of the cylinder.

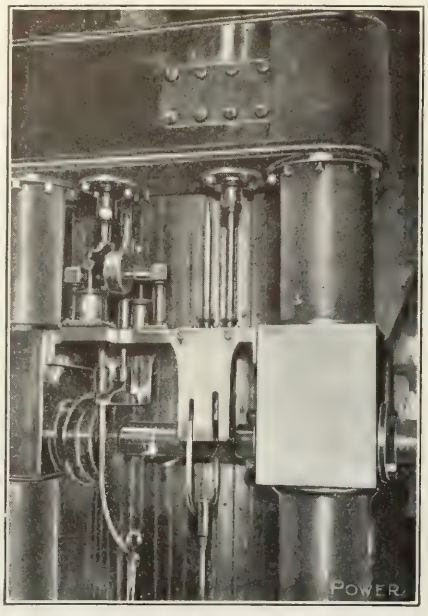


FIG. 4. VALVE GEAR OF BEAM ENGINE

The engine has poppet steam and exhaust valves, which are operated by cams. The cutoff of the steam valves is controlled by a cone-shaped sleeve, which is moved in or out in a horizontal position by the governor. The latch blocks and cam are oscillated to a front and rear position by gravity as the valve is tripped or latched for opening. The steam valves are at the top and bottom of the cylinder in separate steam chests. Live steam passes through cast-iron upright columns to the cylinder.

This engine runs condensing, a jet condenser being operated by a rod running from the walking-beam to the condenser pump, in the engine-room basement.

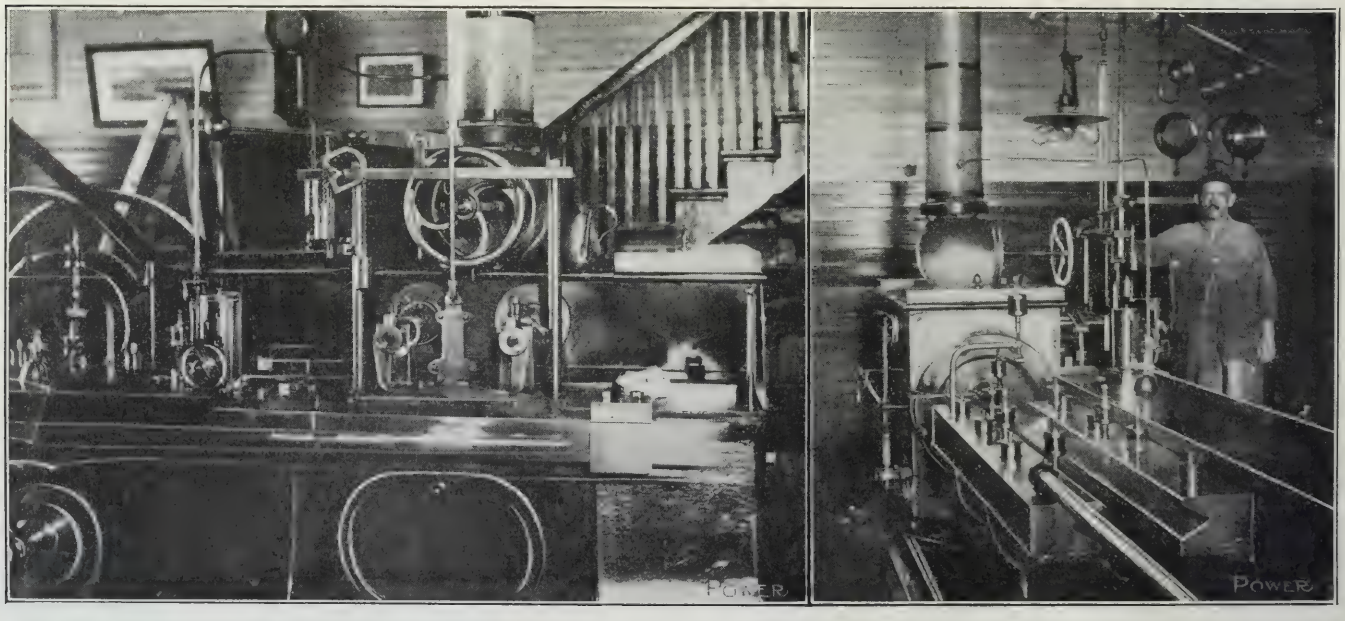


FIG. 3. TWO VIEWS OF ANOTHER OLD GREEN ENGINE

Large Air Compressor Plant at Panama

BY FRED H. COLVIN

One of the largest air-compressor plants in the world is that which supplies compressed air for the work on the Panama Canal. Of the four separate stations, the one at Empire is the largest and the one at Balboa the smallest. The other two are at Las Cascadas and Rio Grande.

The air compressors at Las Cascadas, Empire and Rio Grande deliver their entire output into a 10-in. main, 14 miles long, from which it is drawn off by smaller subsidiary mains to the air drills and other machinery at various points along the canal. The great majority of the air is used in and around Culebra Cut.

The plant at Balboa comprises two Laidlaw-Dunn-Gordon compressors, each capable of delivering 2200 cu.ft. of air into a main four miles long. The two plants thus have 18 miles of air-compressor mains, which is believed to be the longest in existence.

The other three plants contain six Laidlaw-Dunn-Gordon compressors of 2500 cu.ft. capacity each and eight Ingersoll-Rand compressors of the same capacity. The Empire plant has an average total monthly output of over 400,000,000 cu.ft. of free air at 70 deg. F. The largest total monthly output for all the plants, according to the last report, was 794,357,-

608 cu.ft., while the average cost of it for the year was 3.24c. per 1000 cu.ft.

The maintenance of the 14 miles of 10-in. main entails a great deal of trouble, due to the changing character of the soil and the constant jarring of the earth from the daily blasts in Culebra Cut. In 1911, 6300 ft. of the main had to be taken up and relaid, due to its proximity to two slides along the banks of the canal. Patrolling these pipe lines has been found advantageous and a good investment in view of the saving effected in the amount of air lost through strained and leaking joints.

A very close account of the cost of compressing the air is kept month by month, the charges being divided into labor and materials for operating the plants, their maintenance, and any extensions which may be necessary.

From oil-fired boilers an evaporation of 14.53 lb. of water per lb. of oil is obtained. Various tests showed that from 157½ to 168 gal. of oil were equivalent to 1 ton of coal.

The lubricating oil is very carefully accounted for, the performance being given in revolutions of the compressors per gallon. The plant at Rio Grande shows 174,000 revolutions per gallon of valve oil, 177,760 revolutions per gallon of engine oil and 461,065 revolutions per gallon of air cylinder oil, while at the Empire plant the last item is 836,875, or nearly double. The output in compressed air per barrel of fuel oil runs remarkably uniform, the lowest being 42,180 cu.ft. of free air per barrel of fuel oil at the Rio Grande plant and 43,196 cu.ft. for the plant at Las Cascadas.

New Power Development for Connecticut

A dam for power purposes costing $5,000,000 is to be constructed across the Connecticut River at Windsor Locks, Conn., in place of the present dam of the Connecticut River Co. A new $7,000,000 corporation is to be formed, and of this interests of the Connecticut River Co., which owns the present dam and canal, represent about $1,000,000. In addition to the $5,000,000, much more will be expended for an auxiliary steam power plant at tidewater. The new dam will be about 4½ miles below the present dam and is expected to make possible the development of about 35,000 hp.

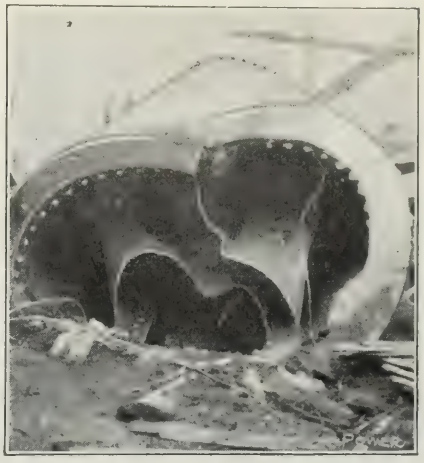
Congress will be asked to grant the franchise to the new corporation, which will probably be called the Northern Connecticut Power Co. That corporation now owns the Enfield Power Co. and supplies electricity for lighting and power from a steam plant. It has the charter right to distribute power to Hartford, Tolland and Litchfield counties and owns also the Connecticut Cable Co., which can operate anywhere in Connecticut.

Webb City Mine Boiler Explosion

Further details of the Webb City, Mo., boiler explosion, Nov. 5 issue, are published herewith.

It was a small vertical boiler and used to run a hoisting engine. It was 30 in. in diameter and 6 ft. high with a double-riveted lap joint. The shell and firebox sheets were ¼ in. thick; there were no stay-bolts, and the fire sheet was riveted at the bottom to a cast-iron mud ring.

The initial rupture was near the mud ring, in the firebox sheet, which was corroded at this point to $\frac{1}{16}$ in. thick. There was no safety valve on the boiler, it having been broken off some two weeks previous and the opening plugged. The boiler contained three gages of water ten minutes before the explosion. The steam gage was located at the throttle of the hoisting engine. The boiler was never inspected, and it is probable that the gage was defective.



EXPLODED BOILER

This is the fourth explosion in this district since the first of the year, in all of which five persons were killed and five badly injured. All of the exploded boilers were old and second-hand, equipped with defective safety apparatus, or none at all.

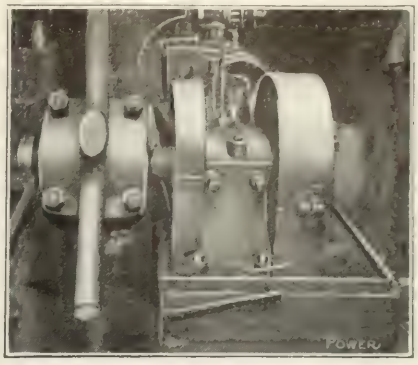
Wauseon Flywheel Explosion

BY OSCAR F. RABBE

An accident occurred at 4:50 a.m., Oct. 31, at the plant of H. S. Williams & Co., Wauseon, Ohio, when a 10-ft. 17-in. flywheel on a Houston, Stanwood & Gamble 10½ and 14½ by 20-in. engine exploded. The wheel weighed about five tons. What was left of it is illustrated herewith. The engine was equipped with a throttling governor.

With the exception of the broken wheel, damage to small piping, walls and roof of the building the loss was small. No one was injured. The wrecked engine was belted to a 60-kw. generator which supplied the commercial and city lighting and at the time was carrying a load of about 25 kw. All lights in circuits were burnt out, due to excessive voltage.

Both engine cylinders were cross-connected so that the engine could be used as a twin as well as a compound unit. One section of the flywheel rim, weighing about 400 lb., tore out the 4-in. cross pipe, went through a belt hole in the wall, hit the wood-lagged pulley on a generator, and deflecting at an angle of about 30 deg. to the left and upward, struck the ceiling and landed on the floor on top of the cross pipe. A section of the rim, weighing approximately 1360 lb., went through the engine-room roof at an angle of about 40 deg. and landed in a field to the east of the plant some 520 ft. from the engine shaft.



EXPLODED FLYWHEEL

A third piece went through the east and west walls of the company's sawmill, and was buried in the ground. All fractured surfaces were clean, showing no flaws except a blow-hole in a rim section, but to all appearances the wheel failure was due solely to overspeed. The flywheel was of two-piece construction with six spokes; the rim joints were bolted with three 1⅝x9½-in. bolts.

A Helping Hand

That much practical good is being accomplished by the international committee of the Young Men's Christian Association is evidenced by its annual report on its work. During the past year 150,000 foreigners, representing 45 nationalities, have been assisted in reaching their friends, helped to secure situations and taught to speak English. The port secretaries in Europe alone met 802 steamers, gave 54,926 cards of introduction to young men, and wrote 4603 letters for them.

The association's work in teaching trades to foreign as well as native members has been highly indorsed by owners and superintendents of shops and factories, and they are anxious to hire men from the association's trade schools.

Bowser Liquid-Measuring Devices

S. F. Bowser & Co., Inc., Fort Wayne, Ind., have recently brought out three types of pipe-line measuring devices for measuring and recording the quantity of any liquid, other than water.

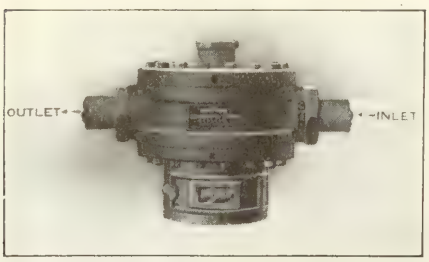


FIG. 1. MEASURE WITH INTEGRATING DIAL

The first type, Fig. 1, measures and records on an integrating dial the number of gallons handled. It is made in sizes for pipes of from ½ to 10 in. in diameter and is applicable in fuel-oil-burning installations for checking up the quantity of oil delivered to the storage tanks, measuring the oil fed to the various burners, etc.

A second type, Fig. 2, is similar in design, except that it has an automatic stop, so arranged that at the option of the operator any predetermined quantity varying from 1 to 1000 gal., can be accurately and rapidly delivered and when this amount has been discharged, the flow is automatically stopped. This device can also be set to duplicate any quantity. When necessary, the measure is steam jacketed to prevent the liquid from freezing.

The remote-control type, Fig. 3, consists of an automatic registering measure and motor-driven rotary pump for handling the liquid. When the predetermined quantity has been delivered, the measure automatically opens the motor circuit and the pump stops instantly. Fig. 4 is a sectional view of the pump showing the principle upon which it is constructed.

An important part of this system is the air separator installed near the measure. It contains a float valve which automatically liberates any air in the pipe and thus prevents it from discharging with the liquid and causing a false reading. This separator contains also a screen of ample area to prevent particles of foreign matter from fouling the measure.

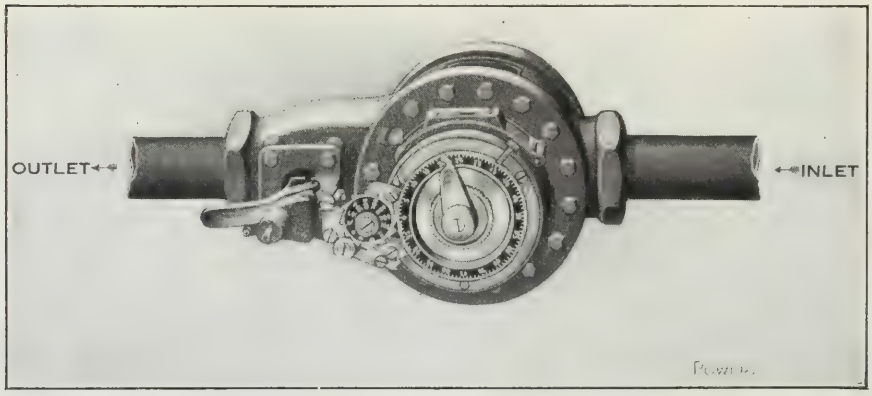


FIG. 2. MEASURE WITH AUTOMATIC STOP

Fatigue of Cast-iron Used with Superheated Steam*

It is now general practice to avoid the use of cast-iron fittings in piping carrying superheated steam, although many such installations are still in use and with no indications of distress in the metal due to the high temperature of the steam.

A case of failure of a cast-iron fitting which formed part of a superheater occurred in the power house of the Union Electric Light & Power Co., of St. Louis. The temperature of the steam in this case was 455 deg. and the temperature of the surrounding gas ran as high as 600 deg. The failure was in the nature of a small crack in a cast-iron outlet, and as there were nearly 5000 similar castings in service under the same conditions, apprehension existed that cast-iron was not a safe metal to use for the purpose. On this account the defective piece was removed and cut up and sample pieces were subjected to careful physical tests. The examination was made in the testing laboratory of Columbia University, and the conclusion reached was that:

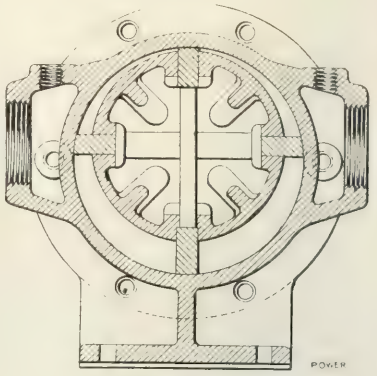


FIG. 4. SECTION THROUGH ROTARY PUMP

> The tensile strength of the cast iron outside the small zone where the crack occurred is quite as high as that of good grades of gray iron, and the condition of service does not seem to have caused a material loss of strength of the metal.
>
> The crack which occurred was due to the segregation of foreign matter in the casting, the area thus affected being small, but in exactly the proper position to receive the stresses which might be induced by the expansion of a pipe in the opening.

Against the earlier experience of the St. Louis company we may instance its later experience and that of various other large companies which were compelled to replace a great number of cast-iron fittings in superheated steam lines because of distortions and failures. A test on pieces cut from the body of a cast-iron valve taken from a superheated steam line of the Commonwealth Edison Co.

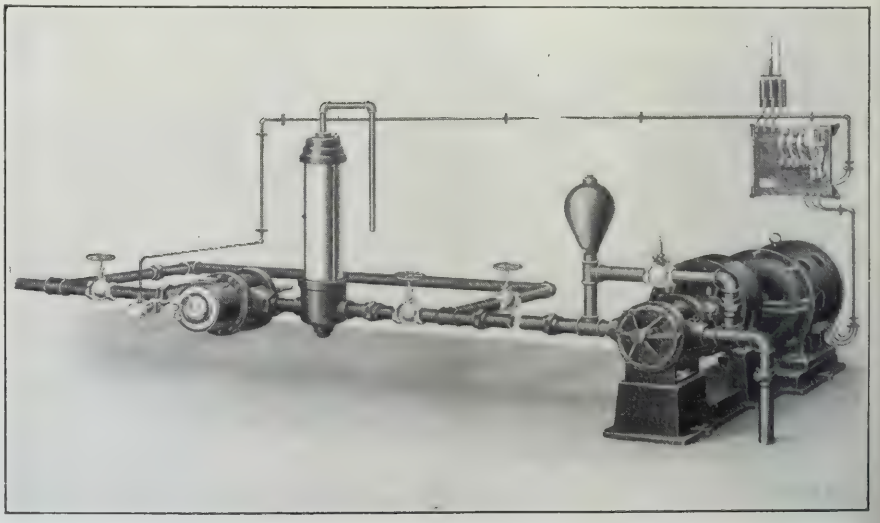


FIG. 3. MOTOR-DRIVEN AUTOMATIC REGISTERING MEASURE

*From the report of the Prime Movers Committee of the National Electric Light Association.

was reported in *The Valve World* for March, 1908, and is in part as follows:

The Crane Co. has been very anxious to obtain something definite on this important subject, and the opportunity for investigation presented itself a few months ago when a 14-in. high-pressure gate valve, which had been in service for four years, was taken out and replaced by a steel valve of similar design. The company, having a complete record of the tests of the iron which was used at the time this valve was made, was in a position to determine accurately what the effect of long-continued superheated steam really was. The salient points are:

A loss of strength in the body of 49 per cent., while the metal in the flanges, not being under quite so high a heat and not directly exposed to the action of the steam, showed a loss of only 33⅓ per cent.

Following is the detailed result:

Test on 14-in. No. 9-E cast-iron gate valve, extra heavy, removed from a superheated steam line.

Steam on line, about August, 1903.

Valve taken out, September, 1907.

Time in service, four years.

Pressure of steam, 200 lb. per square inch.

Temperature of steam, about 590 deg., sometimes a little higher.

Original tensile strength of cast iron, as shown by test bars, 22,400 lb.

Tensile strength of bars cut from body of valve, 12,303 lb., 11,608 lb., 10,610 lb., 12,440 lb.; average strength, 11,740 lb.

Loss of strength in body after four years' service as compared with original test bars, 49 per cent.

Tensile strength of bars cut from the flanges on the valve, 14,900 lb., 15,250 lb.; average, 15,075 lb.

Loss of strength, 33⅓ per cent.

The valve above referred to was 22½ in. long when it was installed in the line, and was $22\frac{13}{16}$ in. long when it was taken out, showing an elongation of $\frac{5}{16}$ in.

The following extract is from a publication issued by the Crane Co. and entitled "The Effect of Superheated Steam on Valves and Fittings":

Cast-iron subjected to superheated steam having a temperature exceeding 500 deg. F.—and we say 500 deg. simply because we know the effect is produced beyond that point and we do not know whether a lower temperature is equally bad—undergoes a slow, constant increase in volume with a consequent loss of strength.

As steel does not appear to increase permanently in volume at temperatures of 500 to 800 deg. F., we may assume that cast iron is so affected on account of the larger volume of elements, such as carbon and silicon, contained in that metal. It may be noted in this connection that cast iron is made up of approximately 92½ per cent. iron and 7½ per cent. of other elements by weight, but the iron is so much heavier than the carbon, silicon, sulphur or phosphorus that when we consider a cube of cast iron by volume we find the elements other than iron, instead of occupying 7½ per cent. of the cube, take up about one-third, hence there is a very large portion of cast-iron which may be affected. Cast steel, on the other hand, contains only about 0.6 per cent. of carbon, silicon, sulphur and phosphorous, an amount altogether too small to have an effect upon the total volume.

Undoubtedly the deterioration of cast-iron in superheated steam lines is due to frequent and rapid changes in temperature within the range of the superheat. For example, suppose a steam line carrying steam having a normal superheat of 150 deg. If, when running at full load, the greater part of the load were suddenly thrown off, the superheat temperature might rise to 200 or 225 deg. If now the entire load were thrown off, the temperature of the steam and of steam piping would fall 200 deg. or more to the temperature incident to the steam pressure.

Suppose, again, while running under normal conditions and with 150 deg. of superheat, water should be carried over from one of the boilers through the superheater and into the steam line. This water would have a temperature 150 deg. less than that of the steam pipe, and the effect on any cast-iron fitting with which it came in contact would be exactly the same as would be produced by pouring water on a hot stove-plate. The metal would warp and its inner surface would shrink and be permeated with cracks, microscopical in size at first, but which would be extended from time to time until the strength of the entire casting was seriously impaired.

For further investigation of this subject reference may be had to the American Society of Mechanical Engineers' *Transactions* for the year 1910, Vol. XXXI, page 989.

The conclusion is that cast-iron fittings in a superheated steam line constitute a hazard, and that the danger increases with the amount of the superheat, and on this account your committee strongly advises against the use of this metal for such purposes.

The *Travelers Standard* is a new monthly journal published by the engineering and inspection division of the Travelers Insurance Co. and the Travelers Indemnity Co., of Hartford, Conn. It will deal with engineering matters of all kinds, but will be mainly concerned with safety engineering as applied to construction work, manufacturing, mining, power generation and transmission, the electrical and chemical industries, and every other form of activity in which machinery or tools are used, or human life and limb are imperiled in productive work. The company says regarding the policy of the standard: "We shall particularly emphasize practical matters, but we shall also deal with theoretical principles to some extent, because it is impossible to be truly 'practical' without having a better knowledge of a subject than can be gained by merely observing facts and combining them crudely by 'rule-of-thumb.'"

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Hen Roost, the popular young fireman at the Pipville featherbed works, says the *Farmer's Advocate*, was some put out on Thanksgiving Day. Hen was corn-feeding a turkey for this festive occasion, but it turkey-trotted for parts unknown the night before. Roost suspicions that the Bulgarians must have slaughtered his "turk" by this time.

Because Thomas Edison, Jr., has declared it his life's ambition to find a substitute for coal, the jokesmiths are poking fun at him. If we had half his enthusiasm—and optimism—we'd drop everything and help him look.

"The Atlanta, Ga., smoke inspector wears a snow-white hat as a smoke detector," says a correspondent. You leave us in grave doubt, sonny, as to the rest of his apparel. Atlanta is a warm town, but we are sure even smoke inspectors must bow to the conventions. Come, now, they must wear something else besides snow-white hats?

Old Doc. Bergonie has discovered that electricity furnishes a means of nourishment for our innards. And just about the time we common peepul have acquired a taste for this kind of diet, some wicked capitalist will have formed a trust in restraint of cheap living, and beat us to it.

It required 200,000 lb. of copper to construct four large brewing kettles of 25,000 gal. capacity each for the Ruppert brewery. Twenty-five thousand gallons of—well, whatever they cook in 'em—is nothing to us, but just think of that large amount of copper!

Sir William Ramsay is said to be working out the details of a "universal primitive sign language." In a manner of speaking, it has Volapuk and Esperanto skinned a mile. We need something of the kind in our operating field, for we have times when no known language seems to have an entirely adequate expression. Some high and primitive sign might turn the trick.

The central station, claims a contemporary, has largely got away from its former "the public be damned" attitude. Some folks are skeptical enough to believe that the central station has frequently got away with more than rightfully belongs to it.

Air Receiver Explodes

By Letson Balliet

On Oct. 30, 1912, the air receiver at the McNamara silver mine, in Tonopah, Nev., exploded, wrecking one end of the compressor room and injuring slightly the night engineer running the hoist. The hoist house stands within 25 ft. of the shaft while on the opposite side of the shaft stands the big mill for treating the ores which are dumped directly from the skips into the bins above the ore breakers. Adjoining the hoist house was a "lean to" shed called the compressor room. The entire structure was of frame covered with corrugated iron. Between the end of the hoist house and compressor room were located two air receivers.

The plant had originally been operated by steam, but had been overhauled and for the last 18 months had been operated by electric power bought from a hydroelectric power company, whose generating plant is in the Sierra Nevada Mountains of California.

A 20¼ and 13¼ by 12-in. Ingersoll-Rand compressor driven by a belt from a 125-hp. motor was uninjured. In May of 1905 a smaller compressor had been installed, with one receiver, and during 1910, the present compressor had been installed, replacing the old one, and a second air receiver connected into the line but the old receiver had been left in use. For a time the steam hoist had been operated by compressed air, but recently the piston rods had been disconnected and a motor had been geared to the intermediate shaft.

The night shift of about 40 men were at work under ground and in the mill, while the hoist engineer was looking after the compressor. The gages had been brought within his vision by removing the partition between the compressor room and the hoist room. The compressor was in sight but the motor was to his right and back of him, while the air receivers were outside the building ahead of the compressor, and between the compressor and the shaft. For some reason the hoist engineer had left his chair, and was about 3 or 4 ft. away from it, when the explosion occurred. In any other spot in the building he would probably have been killed. He was cut by flying débris, the controller for the hoist motor was knocked over and the chair crushed by a piece of the receiver that dropped through the roof. The receiver that exploded was the old one which had been in use since May, 1905. It was 4 ft. in diameter by 16 ft. long and made out of ⅜-in. boiler plate. Longitudinally there were four plates held together by single-riveted lap joints. Both heads were dished and secured with a single row of rivets.

In the explosion the receiver was literally torn in two pieces, similar to the method of opening a can by winding up a strip of the metal on a wire key. A strip 2 ft. wide, badly bent and broken into three or four pieces, was torn out of the middle of the receiver. One head and two tiers of the side plates (about 8 ft. long) was thrown up over the building and dropped through the roof, the strips torn out were thrown in several directions, while one head and about 6 ft. of the receiver moved only a foot or two. The other receiver was blown a distance of 20 ft. and apparently not damaged.

Fig. 1. Scene of the Explosion

Fig. 2. Dotted Line Shows Flight of Receiver

At the time of the explosion the gage pressure was about at the usual point of 100 lb. The receiver was equipped with a safety valve in working order. The compressor was of the piston-inlet type, with a throttled outlet for unloading its low-pressure cylinder at 100 lb. pressure, thus the compressor "unloaded" automatically at about 4 lb. before the safety valve would open.

Not a single rivet had started nor did the seams show any indication of weakness. The grease and dust and noncorrosive paint over them looked exactly as it did before the explosion. One broken edge was almost in the middle of a plate, and continued straight around the cylinder, even where it crossed the lap seam. The other broken edge was about the same in appearance and 2 in. from the row of rivets in the same plate.

Fig. 3. Compressor Room after Explosion

Fig. 4. Pieces of Plate Torn from Receiver

The rivets and seam were perfect, but the broken edges had been torn like a piece of cardboard (not square across but rough and laminated, tearing back ½ in. or more and splitting the sheet).

The 8-ft. end that had been blown away by the explosion was dry and burned and showed no sign of grease on its interior, nor did the pieces that had been torn out, but the 6-ft. end of the receiver that remained practically in its original position, had 2½ in. of grease or half solidified oil on the bottom, as it lay on its side, while the entire inside was lined with a scum of grease ¼ in. thick.

Apparently the grease and oil, which had accumulated for 7½ yr., had formed with the dust a scum on the interior area of the receiver which was too thick to flow to the drain valve. This had dried and baked upon the inside of the receiver. From the fact that the end that remained in place was still coated with grease and the pieces that were blown away by the explosion were devoid of grease and dry and clean, and appeared in spots like they might have been burned, it would appear that the vapor from this grease, held under pressure and the heat of compression, had become ignited, causing an explosion of lubricating-oil gas.

Many mines have air compressors and receivers that are not properly arranged for draining and there are probably other uses of compressed air in which defective conditions exist. It would seem that a manhole, with occasional cleaning and scraping of the interior of the receiver, should be provided as a means of preventing receiver explosions.

N. A. S. E. Educational Program

The educational program of the National Association of Stationary Engineers will include five lines of work, which will be begun as soon as practicable.

For lecture work among subordinate associations, lantern slides will be loaned by the national educational committee free of charge under certain conditions to be announced shortly. The subjects to be covered are steam; combustion and smoke prevention; steam turbines; boilers and superheaters; indicator diagrams, and gas engines. From 20 to 40 slides will be provided for each subject.

A catalog describing the pictures and giving information which will be helpful to anyone preparing lectures on the subjects in question will be sent to association secretaries as soon as the lists are complete. Three or four sets of slides will be ready for use by Jan. 1. Members having slides are requested to send duplicates to the Advisory Board, N. A. S. E., Box 81, Back Bay, Boston, Mass. In this way the board will have a large collection available.

The board during the coming year intends establishing a question department relating to plant operation and any technical work in which members may wish to be informed; these will be answered confidentially. The board is not anxious to waste its time over puzzles or trick questions, but will endeavor to be of practical assistance to members needing operating information.

As a means of encouraging educational work, subordinate associations will be offered prizes for holding the most instructive single meeting between Nov. 1 and May 1. To compete, an association must send a complete report of the meeting, signed by the president and secretary, stating the number of persons present and taking part in the discussion, and giving the discussion in full. The award will be based entirely upon these reports. Four prizes are offered of steam-engine indicator outfits of equal value, which will be awarded after May 1, 1913.

New Haven Meeting of A. S. M. E.

On Nov. 13 the New Haven, Conn., branch of the American Society of Mechanical Engineers held an afternoon and evening session in the lecture room of the Mason Laboratory of Mechanical Engineering. E. S. Cooley was chairman of the afternoon meeting and Prof. L. P. Breckenridge of the evening session.

Papers on the transportation of material in manufacturing plants were presented by C. F. Bennett, of the Stanley Works, New Britain, and by Herbert L. Seward, of the mechanical engineering department of the Sheffield Scientific School, Yale University. The second paper was illustrated by lantern slides. The papers were discussed by Arthur Brewer, superintendent of the Bridgeport Brass Co.; C. F. Hutchins, Sprague Electric Co., and Mr. Goldstein, of the Link-Belt Co. The remarks of the two last speakers were illustrated. C. W. Rice and Prof. Breckenridge also discussed the papers.

Written discussions were sent in by C. D. Rice, superintendent of the Underwood Typewriter Co., and Harwood Frost, of the Brown Portable Elevator Co.

The general subject for the evening session was "Industrial Education." Calvin W. Rice, secretary of the society, made a short address on the work of local branches of the American Society of Mechanical Engineers. This was followed by addresses and discussion on the general subject with especial reference to its present development in Connecticut. The work now being done by the state was described by F. J. Trinder, principal of the New Britain Trade School.

Both meetings were well attended and much interest in the subjects presented was manifested.

OBITUARY

Clement A. Griscom

On Nov. 10, Clement A. Griscom, one of the founders and the first president of the Society of Naval Architects and Marine Engineers, died at his home in Philadelphia, Penn.

Mr. Griscom was born in Philadelphia, Mar. 15, 1841, the son of Dr. John D. Griscom, a prominent Philadelphia physician. He received his early education in public and private schools. In 1857 he became a clerk for Peter Wright & Sons, shipping merchants, of Philadelphia. In 1863 he was admitted to partnership at the age of 22.

Mr. Griscom was one of the first in this country to recognize the growing ascendency of steam over sails for commercial purposes. He was elected president of the International Navigation Co. in 1888 and in 1902 was associated with J. P. Morgan in forming the International Mercantile Marine Co.

PERSONAL

W. R. Addicks, vice-president of the Consolidated Gas Co., of New York, has been elected president of the American Gas Institute.

R. D. Tomlinson has resigned his position as manager of the condenser and steam-engine department of the Allis-Chalmers Co., to become vice-president and general manager of the Grangers Lime & Marble Co., of Danbury, Conn. This will bring Mr. Tomlinson back to the scenes of his early life. Born and educated in Connecticut, he came to New York with the John F. Bushnell Co., became power-plant engineer for the Interborough Rapid Transit Co. in some of its earlier stages, leaving that position to become manager of the condensing department of the Allis-Chalmers Co., where he was later given the management of the steam-engine department also. He is a member of the American Society of Mechanical Engineers, past president of the National Association of Stationary Engineers, and one of the founders of the Institute of Operating Engineers, and if he carries into this new connection the same interest and intelligently applied endeavor which have marked his course as an engineer, the success which all his friends wish for him will be assured.

Twenty-four thousand men, as compared with 17,600 last year, are now employed by the Westinghouse companies. Their business is close to record levels.

Moments with the Ad. Editor

Here's a little problem in advertising for you.

Supposing a friend should ask your opinion of Sapolio, Elgin or Waltham Watches, Steinway Pianos, Yale Locks, Tiffany Jewelry, Royal Baking Powder, Pear's Soap, Quaker Oats, Eastman Kodaks, Victor Talking Machines or Uneeda Biscuits?

You may never have used any of them or even seen half of them, but the chances are 10 to 1 you'd say, "Why, sure, those articles are all right—you'll make no mistake in buying any of them," or words to that effect.

Now the question is, on what is your opinion based? That's the problem we're getting at.

And again it's 10 to 1 that your answer will be, "Well, I know all about them because I have read so many advertisements about them for so many years—and I know they are *good* because they have been so extensively advertised."

All of which reasoning is almost but not quite correct.

In reality you must turn your reason right around.

Those articles are not *good* because they are so well-known, but *they are well-known because they are so good.*

That's the *real* answer to the problem although the final result is the same to you, the buyer.

In other words, it's the quality and satisfaction-to-the-buyer in those goods that have made it highly profitable for the manufacturers to spend hundreds of thousands of dollars in advertising them to you.

For there is no quicker way on earth to get rid of a fortune than by advertising poor, unsatisfactory goods—

And there's no *better* investment than advertising *good, high quality products.*

Advertisers realize this and so do publishers—

And therein lies the basic reason why advertising protects your pocketbook.

POWER

Vol. 36 NEW YORK, DECEMBER 3, 1912 No. 23

Know Your Power Costs

AS WE have pointed out repeatedly, one of the most vital necessities in power plant operation is knowing what the costs are from month to month and year to year. Every engineer in charge of a plant, no matter how small it may be, should have cost data at his finger tips, for his own satisfaction, if not for his actual salvation. It is important to know just where the money goes and for what.

We actually have heard of men—calling themselves engineers—who reasoned something like this: "Why should I bother my head about coal bills and operation costs? I don't pay the bills. I order only as much coal as I need; I don't throw any away. What more can I do? What difference would it make if I did keep track of what I burn?"

Other men argue along this line: "I know it is a good thing to keep track of the costs, but how can I do it when the boss will not put in automatic scales, recording water meters, steam-flow meters, recording and integrating wattmeters, etc.?"

If such men do not want to get into a rut and stay there, probably even if they merely hope to retain their present positions, they must change their attitude and *get busy*. The first requisite is to know what you are doing; the next, to try to discover some way to improve the operation. And having made the improvement, you want the figures in black and white to *prove* that you have been "up and doing."

The man who does not try simply because he thinks he can't accomplish anything unless he has an elaborate assortment of scales, meters, etc., is probably sincere enough in his intentions but a little shy of initiative. While he may not be able to get a complete or accurate analysis of the operating costs without a suitable equipment of testing and metering apparatus, he can get a good and fairly complete set of cost data by exercising a little thought and ingenuity.

In any event, he can get *some* information if he will but *try*.

We earnestly invite the attention of all operating engineers to the article on "Power Costs in a Manufacturing Plant" on page 814 of this issue. It is one of the best articles of its type we have received in a long time. Thoroughly practical, it gives an idea of what can be done by any resourceful man, no matter how scant his testing equipment may be.

True, the costs per kilowatt-hour and per thousand feet of air probably are not very exact, yet no one can maintain that the figures do not answer every need under the existing circumstances. They show the total yearly expense, the approximate unit costs for the purposes of comparing one year's operation with that of another, and, above all, they show convincingly that it was cheaper to maintain a private plant than to purchase current from an outside source at the prevailing contract rates.

What one man has done others can do also!

Power Costs in a Manufacturing Plant

By H. J. Mistele

How the economy of a new plant was increased by cleaning and improving the circulating-water system, stopping a careless waste of engine oil and by purchasing the coal which would evaporate the most water for the least money under the existing conditions.

Description of the system of records employed. Four years operating costs and a comparison of them with the cost of purchased current.

The present power plant of the Falk Co., of Milwaukee, Wis., steel founders and machinists, was completed early in the spring of 1907. It is brick and concrete, 100 ft. square and 50 ft. high. A dividing wall separates the engine room from the boiler room. As may be seen in Fig. 1, the engine-room floor is 10 ft. above the ground and boiler-room floor level, making a light and airy basement for the auxiliary machinery. The chimney, of reinforced concrete, is 180 ft. high and 8 ft. in inside diameter. The coal bunker, also of reinforced concrete, faces the boilers and the bottom inclines toward them so that the coal falls by gravity on the firing floor, as shown in Fig. 2. Hopper-bottom coal cars are hauled up an inclined track onto the trestle over the bunker with an electric hoist and the coal is emptied directly into the bunker.

In the boiler room are five 300-hp. Wickes vertical water-tube boilers which originally were hand fired but now have Swift stokers. Steam is generated at 150 lb. gage pressure. The boiler room also contains two Burnham duplex outside-packed plunger feed pumps 14x8x16 in. in size; one 6x4x6-in. Prescott duplex pump for operating the turbine cleaner, and one Crawley open feed-water heater

Fig. 1. Power Plant of the Falk Co., Milwaukee, Wis.

of 1000 hp. rated capacity. The four Tomlinson barometric condensers which serve the main units are also in the boiler room.

Engine-room Equipment

Fig. 3 is a view in the engine room which contains two electric-generating and two air-compressing sets. The main generator, of 550 kw. capacity, is driven by a 20 and 42 by 42-in. Allis-Chalmers horizontal cross-compound Corliss engine, running at 100 r.p.m. For noon-hour service and at other times when the load is light, a 125-kw. auxiliary generator, driven by an 18x24-in. Allis-Chalmers vertical Corliss engine running at 125 r.p.m., is used. Direct current is generated at 250 volts.

The main air compressor is a horizontal two-stage cross-compound unit with 18- and 30-in. air cylinders, 18- and 36-in. steam cylinders and a stroke of 42 in., with a rated capacity of 2500 cu.ft. of air per minute when running at 75 r.p.m. The auxiliary compressor is a single-stage unit with 22-in. cylinders and a 36-in. stroke, rated at 1400 cu.ft. per minute at 90 r.p.m. The air is compressed to 80 lb. gage. Both units were built by the Allis-Chalmers Co.

In the basement, as shown in Fig. 4, are two 10-in. horizontal double-suction centrifugal pumps of 10,000-gal.-per-min. rated capacity, driven by a 6x6-in. American Blower Co. vertical engine running at 250 r.p.m., used for supplying river water to the condensers. The mean static head from the suction well to the pumps is 13 ft. and from the pumps to the condensers, 35 ft. Another pump of the same type, but 6 in. in size, is used for pumping sludge from the well. An 18x 10x12-in. Prescott duplex underwriters' fire pump and a 10x7x12-in. Prescott duplex pump for general service are also in the basement.

Condenser Trouble

The writer took charge of the plant about three months after it was started. The first thing that attracted his attention was the fact that but 18 or 20 in. of vacuum was being carried. At the first opportunity the circulating-water pumps were taken down and examined. The parts were found choked with everything imaginable from fish to stones. The water is taken from a well that is supplied from a shallow river. After cleaning the pumps the 9-in. suction pipes were taken up and the strainers examined. The bottom of each pipe was fitted with a bell-shaped casting across the mouth of which a screen was fastened. This had become clogged and the effect of the vacuum had forced the screen part way up into the bell, admitting all the refuse to the pump. The bells were removed and sliding screens were substituted, which have eight times the area

Fig. 2. Part of the Boiler Room

of the former screens and which can be lifted and cleaned properly whenever they become choked. This settled the trouble with the condensing system and thereafter there was no trouble in maintaining 26 or 27 in. of vacuum.

Excessive Lubrication Cost

Unreasonably high lubricating bills were the occasion of the next investigation. All the bearings are lubricated by a gravity system and the consumption of engine oil ran as high as three barrels per month. The joints of the sheet-iron drip pans and crank housings were tightened and cemented with a plastic metal cement. The oil pumps, in the basement, are set in a pan and formerly a drain pipe led from this to the sump. It was found that the pump rods were leaking badly and the constant drip soon carried away a barrel of oil to the sump. The oil end of the pump tray was properly piped so that all the drip from the rods drained back into the return reservoir instead of to the sump. After the system had been put in order the oil consumption dropped from three barrels per month to four barrels per year.

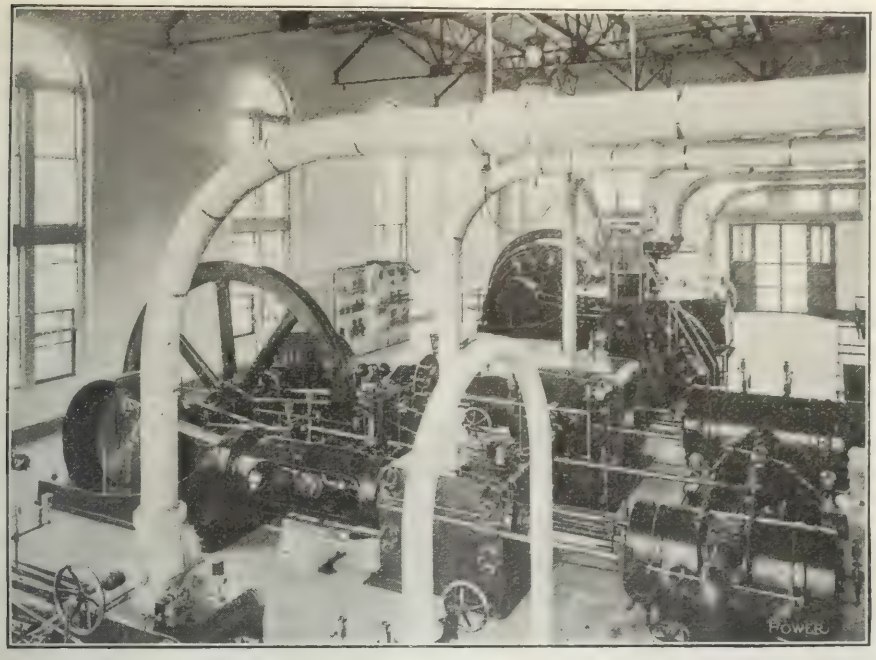


Fig. 3. A View in the Engine Room

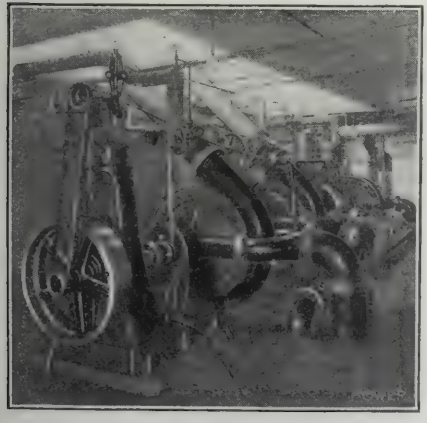


Fig. 4. The Circulating Water Pumps in the Basement

Studying the Fuel Problem

During the fall of 1907 the fuel question was taken up. It was attacked in the most practical way as the sole object was to ascertain what coal would evaporate 1000 lb. of water the cheapest under the conditions existing in the plant, regardless of the number of heat units it contained. One of the boilers was equipped with a hot-water meter and the usual appliances for making the necessary observations. The coal being used at the time was Pittsburgh mine-run, costing $3.20 per ton in the bunkers. Five tests were made with this coal. The results showed that the average cost of evaporating 1000 lb. of water from and at 212 deg. F. was 18.24c.

The next grade tested was Youghiogheny screenings costing $2.75 per ton. Six tests were conducted with this coal and the average cost of evaporating 1000 lb. of water from and at 212 deg. F. was found to be 16.84c., indicating a saving of 7.7 per cent. over the Pittsburgh mine-run coal. Consequently, Youghiogheny screenings were used during 1908, except a few carloads of other coals that were accepted for test but from which no better results were obtained.

Late in the fall of 1908, Hocking screenings at $2.35 per ton were tried. Three tests were conducted, which showed that the average cost of evaporating 1000 lb. of water from and at 212 deg. F. was 16.5c. This coal was used all during 1909 in the spring of which year a reduction of 25c. per ton was granted, making the cost $2.10 per ton. This price prevailed for one year. Five tests were made with the $2.10 coal and the average cost of evaporating 1000 lb. of water from and at 212 deg. was found to be 14.6c., showing a saving of 13.3 per cent. over Youghiogheny screenings and 20 per cent. over the Pittsburgh mine-run. In the spring of 1910 the cost per ton went back to $2.35 and has remained there ever since.

In the fall of 1911 the question came up as to whether the use of mechanical stokers would result in greater economy. One of the boilers was fitted with a Swift stoker and four tests were made, followed by two tests with hand firing. The results indicated an improvement in economy of 7.5 per cent. in favor of the stoker. As a result all of the boilers were equipped with stokers.

THE FALK COMPANY
POWER HOUSE DAILY REPORT.

For Twenty-four Hours commencing 6 P. M. ... Day of ...

EQUIPMENT	Started	Stopped	Hours Run	Rev.	Watt Meters A. M.	Watt Meters P. M.	K. W. Generated Day	K. W. Generated Night	TOTAL	GENERAL REMARKS
Generator No. 1										
Generator No. 2										
Generator No. 3										
COMPRESSORS					Gallons A. M.	Gallons P. M.	Fuel Day	Fuel Night	TOTAL	
Air Cooling No. 1										
Air Cooling No. 2										
CENTRIFUGAL PUMP										
Pump No. 1										
Pump No. 2										
Pump No. 3										
BOILER FEED PUMPS			Days Run		Gallons A. M.	Gallons P. M.	Duplex Strokes Day	Duplex Strokes Night	TOTAL	
Pump No. 1										
Pump No. 2										
Fire Pump										
BOILERS	In	Out	Days Run	When Washed	Coal Received Gross Tons	Coal Received Net Tons	Tonnage	Ash and Refuse	Hours Steaming	
Boiler No. 1										
Boiler No. 2										
Boiler No. 3										
Boiler No. 4										Engineer
Boiler No. 5										

Fig. 5. Daily Report Sheet (Actual Size 14x8½ in.)

All of the foregoing tests were conducted under everyday working conditions and when the boilers were operating considerably below rating. The results show what can be accomplished by

experimenting with different grades of coal. Several years ago the writer tested six different grades of coal in another plant and under different conditions and got just the opposite results. The highest priced coal yielded the most economical results.

System of Records

The power-plant records are kept on forms, shown in Figs. 5 and 6, which are filed in a loose-leaf book. The readings are taken at 7 a.m. and 6 p.m., 11 hr. constituting the day run and 13 hr. the night run. Records are kept of each night and day run together with the total for each 24 hr. The number of cubic feet of air delivered is estimated from the revolution counter readings. The daily reports are on yellow paper and the monthly on pink, facilitating their distinction in the loose-leaf file. All repairs or important occurrences are entered in the daily reports.

At the end of the year the monthly reports are totaled and averaged and the figures are arranged as shown in Table 1. In reducing the quantity of air compressed to its equivalent in kilowatt-hours it is estimated that 0.105 kw. is required to compress 1 cu.ft. of air per minute. This theoretical figure is only approximately correct, but answers for comparing one year's work with another.

Besides supplying current and compressed air, the power plant furnishes high-pressure steam for heating throughout the works, and for operating various steam hammers, sump pumps, oil pumps, etc. The use of live steam for heating was considered more economical than exhaust steam because foundries do not require heating except for short intervals during extremely cold weather and the greater economy secured with condensing engines more than offsets the relatively small loss in the present case due to the use of live steam.

Analysis of Yearly Costs

A large amount of coal is required for maintaining steam on the works for power purposes because of the great area covered and consequently the great loss due to condensation.

In Table 1, all of the coal consumed is charged against the power and air output, making the cost seem higher than it should. This is done because the plant is not arranged so that a close record can conveniently be kept of the steam distribution.

This power plant is the second the company has had to build within 10 years because of the increase in its business. When the present equipment was first put into operation the load was less than 50 per cent. of its rated capacity and the actual cost of current and compressed air was relatively high because of this inefficient loading. By 1910 the current and air output had greatly increased; hence the machines operated at or near rating for longer periods and the actual economy was much better. Then, too, as the heating load remained nearly the same, and as the quantity of steam required for power in the works increased but slightly, the outside steam charged against unit current and air output was much less, resulting in a better looking cost figure. The third factor in reducing the unit cost was the use of cheaper coal and the installation of mechanical stokers.

It will be noticed that the cost figures for 1911 show an increase of 10 per cent. in unit cost. Part of this increase is chargeable to the increase in the cost of coal per ton; another part is due to greater heating demands during January, February and March, and, finally, an extra pump used for a sand slusher was operated for the greater part of the year, for which the power plant has received no credit in the report.

When the last two boilers were installed in September, 1911, one was so arranged that it could be isolated and employed solely for supplying steam to the works. A seven-day test was then conducted to determine the amounts of steam required for this work. It was found that the average weight of coal consumed for this purpose was 8700 lb. per 24 hr. Deducting this from the weight of coal consumed for power and air, the approximate weight of coal consumed per kilowatt-hour under average conditions was estimated to be 4.35 lb. Table 2 is based on these figures.

The first column gives the estimated weight of coal consumed for various purposes. The second column gives the price of the coal based on the average cost per ton throughout the year, including switching charges. The fourth column gives the cost of labor. This item was estimated as follows:

The salaries of the three firemen and half that of the chief engineer were

THE FALK COMPANY.
POWER HOUSE MONTHLY REPORT

Month of 19

	GENERAL REMARKS
Total K. W. H. Generated	
Average per Day of 24 Hours (Days)	
Average per Day of 11 Hours	
Average per Night of 13 Hours	
Average Amperes per Day	
Average Amperes per Night	
Percentage above below last Month	
Total Cubic Feet of free Air delivered	
Average per Day of 24 Hours (Days)	
Average per Day of 11 Hours	
Average per Night of 13 Hours	
Total Revolutions	
Percentage above below last Month	
Coal in Bunker on first of Month Tons	
Coal Received during Month Tons	
Coal Consumed during Month	
Coal in Bunker at end of Month	

Engineer

Fig. 6. Monthly Report Sheet (Actual Size 14x18½ in.)

TABLE 1. SUMMARY OF POWER COSTS

	1908	1909	1910	1911
Total current to swbd., kw.-hr	836,400	1,057,600	2,082,000	1,950,300
Maximum	(Oct.) 86,200	(Dec.) 132,900	(Sept.) 206,800	(Oct.) 196,300
Minimum	(Feb.) 53,900	(May) 65,600	(Feb.) 134,000	(Feb.) 128,500
Average (12 months)	69,700	88,133	173,500	162,525
Increase over previous year, per cent		26	97	(Decrease) 6.33
Air delivered, cu.ft.	191,563,100	270,946,100	398,710,500	368,911,600
Maximum	(Aug.) 19,466,500	(Dec.) 28,498,000	(Aug.) 41,920,500	(Oct.) 36,467,800
Minimum	Jan.) 10,262,000	(Jan.) 15,684,100	(Dec.) 24,176,700	(Feb.) 24,400,000
Average (12 months)	15,963,600	22,578,840	32,225,875	30,742,600
Increase over previous year, per cent		42	47	(Decrease) 7.5
Air equivalent in kw.-hr.	333,450	471,626	694,022	642,150
Total output expressed in kw.-hr	1,169,850	1,529,226	2,776,022	2,592,450
Increase over previous year, per cent		31	81	(Decrease) 6.6
Total coal consumption, tons	5,108	6,400	9,153	8,966
Cost of coal	$15,275	$14,193	$20,741	$21,494
Coal consumed per kw.-hr., lb.	8.7	8.37	6.59	6.92
Cost of coal per kw.-hr., ct.	1.3	0.928	0.747	0.829
Total cost of labor	$4,470	$4,590	$5,430	$5,625
Cost of labor per kw.-hr., ct.	0.38	0.3	0.195	0.217
Cost of oil and waste	$350	$340	$390	$375
Cost of oil and waste per kw.-hr., ct.	0.03	0.02	0.014	0.014
Total operating cost	$20,095	$19,123	$26,561	$27,495
Total cost per kw.-hr., ct.	1.71	1.25	0.956	1.06
Reduction over previous year, per cent		27	24	(Increase) 10

charged against the total coal consumption, and the labor cost for supplying steam to the works and for heating is based on this figure. The cost of labor charged against power and air combined equals the cost of labor per ton of coal fired times the number of tons used for power and air plus the salaries of the two assistant engineers plus half that of the chief engineer. From this the individual cost of labor for power and air is estimated from the ratio of coal used for each purpose to the total quantity used for both purposes. In the fifth column the cost of oil and waste is charged against power and air.

TABLE 2. DISTRIBUTION OF COSTS FOR 1911

	1 Coal Distribution tons	2 Cost of Coal at $2.397 per ton	3 Coal Distribution in per cent.	4 Labor Cost Distribution	5 Total Cost Distribution	6 Total Cost Distribution in per cent.
Power supplied to main works	2890.65	$6930	32.2	$2304	$9426	34.3
Power supplied to branch works	1351.25	3239	15.1	1078	4406	16.0
Air supplied to main works	1396.60	3348	15.6	1113	4555	16.6
Steam supplied to main works	1587.75	3806	17.7	540	4346	15.8
Steam for heating	1739.75	4171	19.4	590	4762	17.3
Totals	8966	$21,494	100	$5625	$27,495	100

Saving Effected

Table 3 compares the cost of manufacturing the current with the cost of purchasing it at prevailing contract rates. From this it will be seen that the average purchase cost per kilowatt-hour during 1911 would have been 1.636c., whereas the actual cost of production was but 1.06c., even when the cost of heating and supplying steam to the works was charged against the power item. The maximum demand charge of the public-service company is based on a 15-min. period whereas the peak given in Table 3 is for an entire day run of 11 hr. The peaks for short intervals often run fully 100 kw. higher than those given in the table and hence the monthly demand charges would probably be $200 or more higher than those given, or upward of $2400 more per year.

In estimating the total actual saving, the first, second and third items in column 5, Table 2, were added, giving a total of $18,387. This, subtracted from the total estimated cost of purchased current, gives $42,408 — $18,387 = $24,021 in favor of the power plant. The plant complete cost $160,000, it is 5 years old and practically as good as new. If 10 per cent. is allowed for depreciation, insurance, taxes and general upkeep, there would still be $24,021 $16,000 = $8021 remaining, or 5 per cent. on the original investment. And this, after charging the power plant with everything except the cost of coal and labor for heating and supplying steam to the works.

During the first two years the showing was not quite so good for self-evident reasons but it will be conceded that it was sufficiently satisfactory when all factors are taken into consideration.

Details of Operation

Every car of coal is washed before being hauled over the bunkers. The bunkers are graduated and at the first of the month the coal is measured up and the amount recorded on the monthly report. It might be mentioned that generally there are between 30 and 60 days in the summer when it is necessary to use Youghiogheny screenings instead of Hocking, due to the fact that there is no Hocking in the market.

The 550-kw. generator operated 11 hr. per day until September, 1909, and the 125-kw. generator operated 13 hr. per night until that date. Since that time the 550-kw. has operated 24 hr. per day for 6 days a week and the 125-kw. has operated Sundays and Sunday nights. The compressor operates 7 nights and 6 days per week. During the lunch hours and between shifts and on Saturdays and Sundays, with practically no load, the operating conditions are very unfavorable from an economical standpoint.

Maintenance and Repairs

The question of maintenance and repairs has not been mentioned in the reports because they are so small that the costs would not have had any material effect on the yearly costs. There are 42 stuffing-boxes on the engines and during the five years' operation the packing on one valve stem has been renewed. One new set of trip steels has been put on, two new sets of brushes have been put on the generators, the small generator commutator has been turned off twice, one rocker-arm and a reach rod on the compressor have been renewed, two new shafts and sets of bushings have been obtained for the centrifugal pumps, three new tubes have been put in the boilers, the brick parts of the furnaces have been renewed once, the boiler-feed pump plungers have been trued up twice and the plungers have been packed once a year. This, with the ordinary valve-stem packing required throughout the house, covers all renewals and extra supplies.

TABLE 3. DISTRIBUTION OF LOAD FOR 1911

	Current Generated kw.-hr.	Max. Output in 11 hr. kw.-hr.	Air Output cu.ft.	Air Equivalent kw.-hr.	Max. Air Output in 11 hr. cu.ft.	Electrical Peak Load kw.	Air Peak Load equivalent kw.	Total Peak Load kw.	Central Station Rates: Max. Demand Charge	Current Charge	Total
						for 11 hr.	for 11 hr.	for 11 hr.			
January	151,000	4900	25,650,000	44,646	1,040,000	466	172	638	$1,306	$2,057 46	$3,363 46
February	128,500	4100	24,400,000	42,471	957,700	390	155	545	1,140	1,810 71	2,950 71
March	155,000	4300	33,630,000	58,536	1,044,600	409	173	582	1,194	2,236 36	3,430 36
April	156,500	4700	30,002,900	52,223	1,027,000	447	170	617	1,264	2,188 23	3,452 23
May	150,500	4700	28,591,600	49,766	961,800	447	160	607	1,244	2,103 66	3,347 66
June	147,000	4300	30,403,700	52,921	1,037,300	390	172	562	1,154	2,100 21	3,254 21
July	152,500	4900	31,589,000	54,985	1,140,400	466	189	655	1,340	2,175 85	3,515 85
August	181,500	4600	35,044,500	60,998	1,116,300	438	185	623	1,276	2,525 98	3,801 98
September	186,500	4700	32,438,400	56,462	1,411,700	447	234	681	1,392	2,530 62	3,922 62
October	196,300	4800	36,467,800	63,476	1,129,100	457	187	644	1,318	2,698 76	4,016 76
November	179,000	4900	30,359,200	52,843	1,037,300	466	171	637	1,304	2,419 43	3,723 43
December	166,000	5000	30,334,500	52,800	1,078,500	476	179	655	1,340	2,289 00	3,629 00
Totals	1,950,300		368,911,600	642,127					$15,272	$27,136 27	$42,408 27

McEwen Flexible Steel Coupling

A new form of cast-steel flexible coupling is being introduced by the McEwen Bros., Wellsville, N. Y. This coupling has a small diameter and mass, and consequently small inertia.

Key pins extend through the shafts and are set at right angles. The coupling will connect two shafts that are considerably out of line and is free of noise or tendency toward serious wear. Very little clearance is required for removing any part.

The coupling is suitable for use under high speeds and severe conditions such as reversible motor drive for machine tools, blowers, rotary pumps, motors, generators, turbines and line-shaft connections, and on machinery of any speeds.

Tom Hunter, Hoisting Engineer

By Warren O. Rogers

In which more pump rooms are visited and Hunter explains the construction and operation of the jet condenser.

After further discussion regarding the Goyne pump we went past the shaft to a second pump room, where the cat was awaiting us, the rat evidently having escaped. Here we found a 26, 45 and 14 by 48-in. duplex compound outside-packed pump working against a head and suction lift of 608 ft., with a capacity of 2600 gal. of mine water per minute.

This pump room was similar to the other. The floor was of concrete and the walls were of cement instead of half concrete and pipe, but the ceiling had I-beams across it supporting discarded pipe over which earth had been filled in. A view in the pump room showing the ceiling construction is given in Fig. 1.

In the cement side walls of the room, lockers had been built, with metal doors, an improvement over the other pump room, which was supplied with iron chests for keeping small supplies. These chests and lockers were to minimize fire risks and also to protect the supplies from dust and injury.

At another mine I later saw a storeroom which had been constructed handy to the pump room. It was made with concrete floor and walls, with ventilating windows in the end wall. The roof was of the natural slate formation. Spare pumps, parts, packing, etc., were kept for immediate use, and it was the pump runner's business to keep the supplies on hand, never allowing the stock to run out, for when a mine pump requires new valves, packing, etc., there is no time to send to distant cities for supplies. Fig. 2 gives an idea of the room.

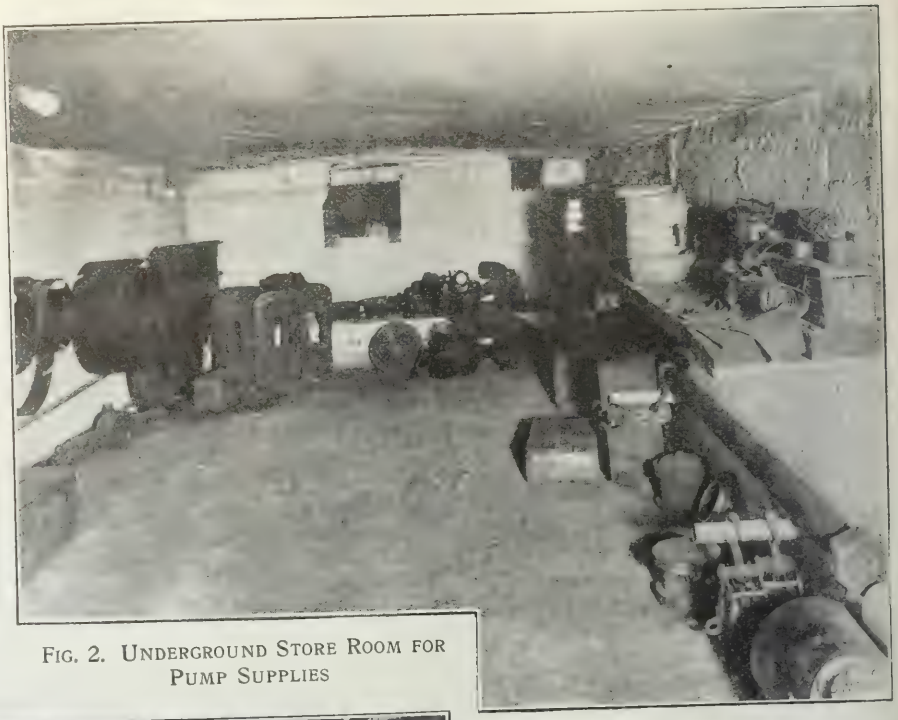


FIG. 2. UNDERGROUND STORE ROOM FOR PUMP SUPPLIES

All pump rooms are not so well laid out as those I had visited, and I later saw one, Fig. 3, that was dirty, hot and congested. The chief machinist was engaged at the time in installing a small pump, under extremely unfavorable conditions, as the room was restricted.

Reaching the rear end of the pump room we came to the 12 and 18 by 18-in. Jeanesville jet condenser, the gage of which registered 25 in. of vacuum.

"I thought you said the surface condenser was the most economical and we find jet condensers are used with all of the mine steam pumps we have seen. How do you account for that?" I asked. "Don't mine people go in for economy?"

Hunter grinned as if amused. "You think you have me there, don't you? Well, a surface condenser is more economical than a jet condenser where the water is reasonably clean and free from acid or alkali. It is adapted for mine work where it can be piped to the suction line to the pump, and the incoming water to the pump used for condensing purposes. This gives a large amount of cooling water that is rapidly moved by the pump and cuts out the necessity of a circulating pump."

"Are they difficult to operate?" I asked, watching the jerking motion of the air pump.

"No, they are simple in operation, and the vacuum pump uses a small quantity of steam as compared with the independent surface condenser. The trouble is that acid in the mine water attacks the metal and causes a heap of trouble. Sometimes I think the water gets worse each year.

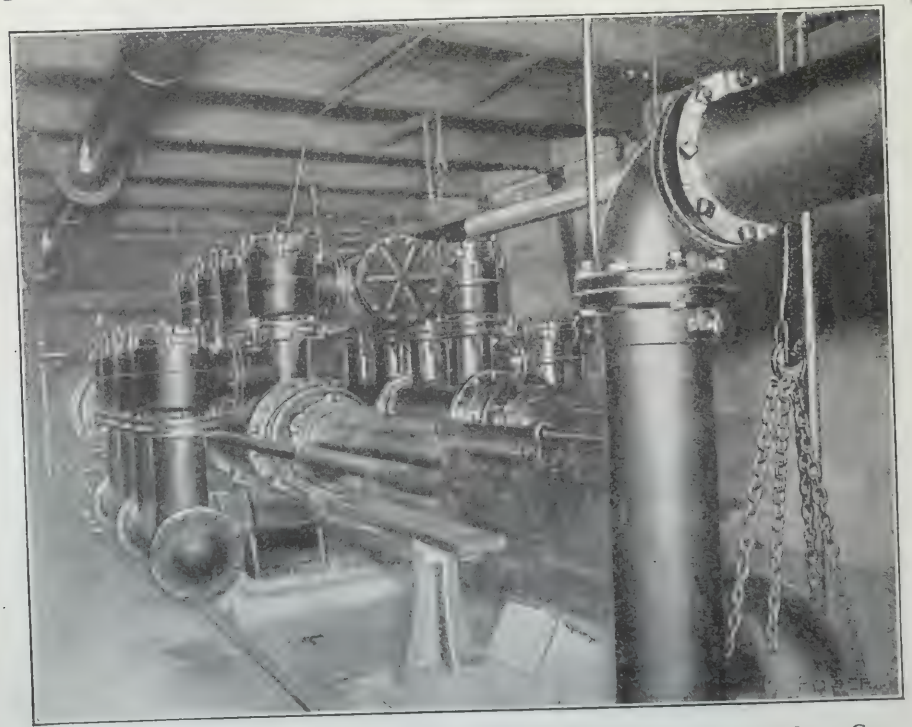


FIG. 1. A PUMP ROOM IN THE NO. 2 COLLIERY OF THE SUSQUEHANNA COAL CO.

If the water is dirty the tubes are pretty apt to become coated, which reduces the heat transference through them."

"Well, won't the acid water attack the jet condenser just as readily?"

"No, for the reason that the water cylinder is bushed with a heavy phosphor-bronze lining, where acid or alkali water is used for condensing. Furthermore, the jet condenser makes a simple and reliable machine for use in a mine where skilled men are not employed.

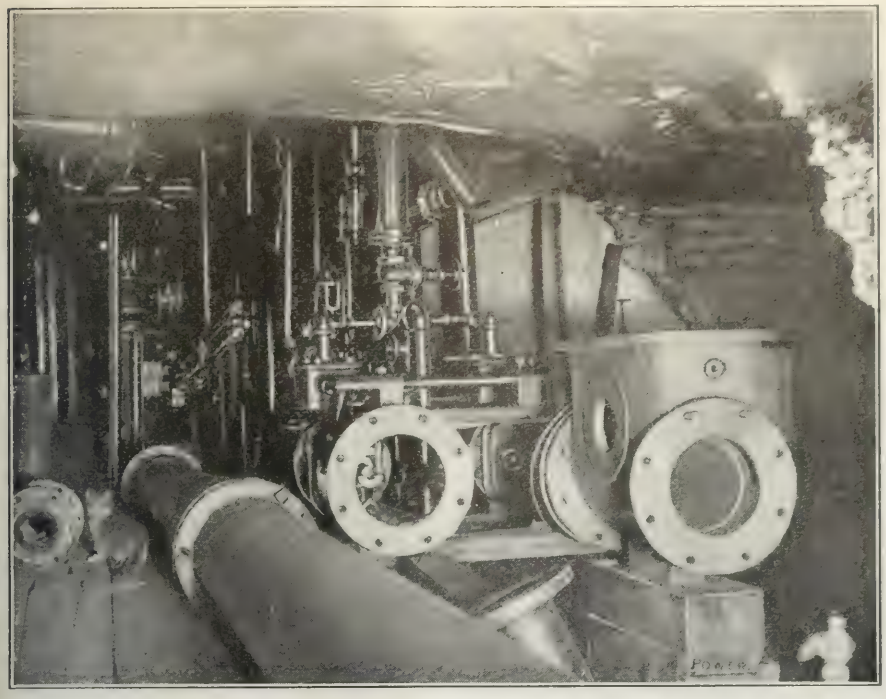


FIG. 3. INSTALLING A PUMP IN RESTRICTED SURROUNDINGS

All valves in the water end are generally made of rubber, and seat on phosphor-bronze seats under curved phosphor-bronze guards. This construction makes a good exhaust valve, and valve springs are eliminated. In fact, all important details about the condenser are made of phosphor-bronze.

"The steam end is fitted with the ordinary duplex valve motion with modifications which insure a full piston stroke." A sectional view of the condenser under discussion is shown in Fig. 4.

"Seems to be a mighty small machine to handle all of the exhaust from that large pump; how does it do it?"

"The principles involved and the method of operating jet condensers are the same in all makes. It consists of a steam-driven pump piston which removes the water in the chamber *A*." Hunter designated the position on the sketch shown in Fig. 4. "The steam and water valves act exactly as in an ordinary direct-acting steam pump and the only difference is that the jet condenser has a condensing chamber attached to it instead of an air chamber. The condensing chamber has an inlet for exhaust steam, and an outlet for the condensing water."

Referring to the sketch, Hunter pointed out how the exhaust steam is condensed. "To condense steam," said he, "it must be cooled and the best available agent is the water from the mine sump. The condensing chamber has a cone valve *B*, and the outlet of the elbow is piped to the sump. The exhaust steam from the steam end of the condenser is piped to the condenser chamber, and the discharge of the condensing water and condensed steam is out through the outlet *C*."

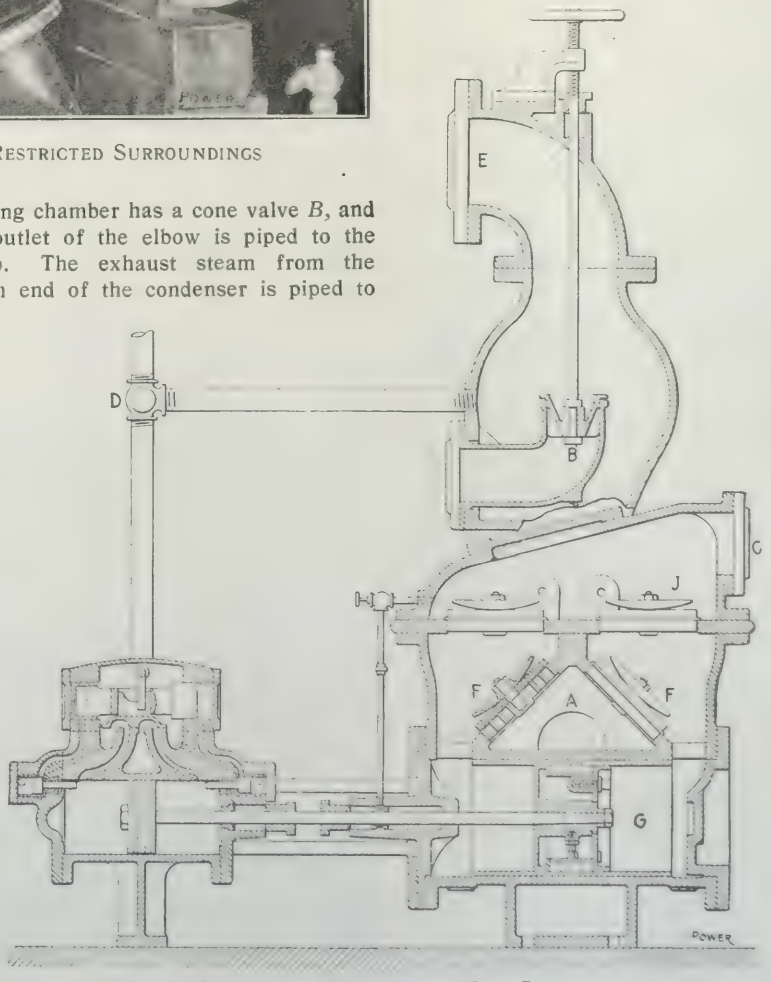


FIG. 4. SECTION OF JEANESVILLE JET CONDENSER

"I should think the pump would get flooded with the combined steam and condensing water, for the volume of steam from the mine pump must be enormous as compared with the size of the condensing chamber."

"One would assume such to be the case, but you see 1 cu.ft. of water evaporated into steam will have a volume of approximately 1600 cu.ft. Reversing conditions and each 1600 cu.ft. of steam when condensed will occupy but 1 cu.ft. and weighs 62.4 lb., or an equivalent of 7½ gal. Between 20 and 30 times the weight of steam is usually allowed for condensing water, so for each cubic foot of steam condensed the air pump would have to handle 225 gal. of cooling water.

"This condenser is capable of handling 700 gal. of injection water at 70 deg. temperature and will condense 14,000 lb. of steam per hour. If the water is above 70 deg. temperature more will be required to do the work. Do you get me?"

"Sure, and now that you have told me about everything except what I wanted to know, tell me how the thing works."

"All right," replied Hunter with a laugh, "but if I had not told you how the condenser was made you would have asked, so I would have had to cover the ground anyway.

"Before starting a jet condenser, the

drain cocks on the steam cylinder are opened and the three-way cock *D* in the exhaust pipe is turned to exhaust to the atmosphere. Then steam is turned on, the condenser is started slowly and the drip cocks are closed after the steam cylinder has been cleared of water.

"As soon as the condenser has taken water in through the cone valve, which is open, the steam from the pump exhaust is turned into the condensing chamber. The condenser is always started before the main mine pump, and after the latter is running smoothly the cone valve *B* is throttled down to give just enough water to maintain a vacuum of 25 or 26 in. The stroke of the piston is adjusted by opening or closing the cushion valve in the steam cylinder."

"Then how do you know when the condenser is running right?"

"The condenser water piston should run away from the water, about 1½ in. at the beginning of the stroke, which will be indicated by the jumping forward motion of the piston during the stroke. This will give the best vacuum and prevent backing the water into the low-pressure cylinder of the mine pump. The hot well discharge should be at a temperature of between 100 and 110 deg. F."

"Now, with the pump and condenser in operation, what happens? The steam from the mine pump enters the condenser chamber at *E*, and a spray of cold water is passing up through the cone valve *B*. The hot steam striking the cold water is condensed and passed through an outside connection to the chamber *A* and out through the valves *F* to the water cylinder, as the piston is moved from one to the other. If the water piston were moving toward the end *G* of the cylinder the water would be forced out through the valve *J* and discharge *C*. Water would, at the same time, pass from the chamber *A* to the opposite end of the cylinder. On the return stroke the water passes up through the other discharge valve. That is all there is to it."

"Well, as most pump pistons leak, why don't air leak into the water cylinder?"

"For the reason the stuffing-box is water-sealed by water from the discharge chamber. If the piston packing does get to leaking the water will be drawn into the cylinder instead of air."

After Hunter made these last explanations we left the pump room and ascended to the surface, to find that night had settled down and the yard, except for a light here and there, was as black as the depths of the mine below.

Cooling Ponds for Condensing Water*

The tests described were made to determine how large a reservoir is necessary to allow water from a condenser to cool sufficiently for reuse. The author's conclusion is that with engines having a water rate of 15 lb. and a vacuum of 26 in. a reservoir having a surface of 120 sq. ft. per hp. would be ample in the northern part of the United States.

*Paper presented at the Cleveland meeting of the American Society of Mechanical Engineers.

The tests described in this paper were made with the idea of determining if possible, a fairly reliable factor by the use of which the necessary size of a cooling reservoir for condenser water could be predetermined for any assumed power and weather conditions.

In 1908, while designing the mill for the Crescent Portland Cement Co. at Wampum, Penn., it was found necessary to install a cooling device for the condensing water from the engines. The mill stands on a level plateau about 110 ft. above the Beaver River and to pump water from the river for condensers meant a considerable expense for power. On the side of the plateau farthest from the river was a natural depression alongside the Erie division of the Pennsylvania R.R. which offered a desirable location for a reservoir (Fig. 1). After a diligent search, no data were found as to how large a reservoir had to be to give the necessary cooling effect to the water, so that it might be used continuously. A dam, however, was built 275 ft. long and 18 ft. high, of skeleton reinforced concrete construction which would impound about 6½ acres of water (Fig. 2).

Three tests were made during the year 1911, each of one week's duration, to determine the heat radiation from the surface of the reservoir. These tests were made in May when the temperature was moderate, in July when the temperature was high and in November when the temperature was low. Readings were taken of the temperature of the river water, the intake water to the power house, the tail water from the condenser and of the air. The vacuum, the power, the amount of water pumped from the river to the reservoir, and the rainfall were also recorded. These data were taken by James Toler, chief engineer of the Crescent plant, and by the writer's assistant, John E. Mason. Unfortunately, the temperature of the rain was not taken, but it has been assumed the same as the temperature of the air, so that the figures for that are only approximate, but a difference of 10 deg. in the temperature of the rain would not alter the result by more than 0.2 per cent.

The engines used in the power house are three Ball & Wood compound-condensing engines, 26 in. by 52 in. by 36 in.

The air compressors are two compound Laidlaw-Dunn-Gordon compressors, the steam cylinders of which are 12 in. and 21 in. in diameter respectively, the air cylinders 11½ in. and 24 in. respectively, the stroke 24 in. The steam from all of these is condensed in an Alberger barometric condenser equipped with a dry-vacuum pump and the circulating water supplied by a direct-connected centrifugal pump driven by a synchronous motor; an engine-driven cir-

FIG. 1. VIEW OF THE CRESCENT PORTLAND CEMENT CO. PLANT SHOWING RESERVOIR

culating pump and a second dry-vacuum pump are installed as reserve.

The water from the condenser flows about 50 ft. through a tile pipe into the east side of the reservoir about 100 ft. from the north end. A dike was built south of this inlet to the reservoir ex-

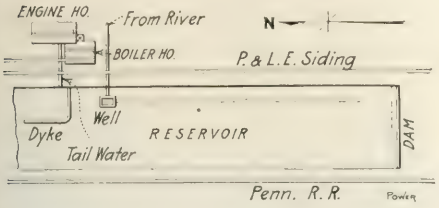


FIG. 2. DIAGRAM SHOWING 6½-ACRE RESERVOIR AND CONNECTIONS

tending about 50 ft. toward the center of the reservoir and then north nearly to the north end. This compels the circulating water to flow up and around the end of this dike and down toward the dam, a distance of about 1100 ft., before it reaches the intake to the power house.

Water to supply the reservoir is pumped from the river by a centrifugal pump directly connected to a 150-hp. motor. The pump delivers an average of 2324.25 gal. of water per minute and it is found that about 8 hr. pumping per week is sufficient to make up for the evaporation and seepage in the reservoir. The water from the river pump is delivered into an open well situated on top of the plateau where the mill is located. It flows by gravity from this well through a 24-in. tile pipe into another concrete well located in the reservoir about 800 ft. north of the dam. From this well it flows through a No. 24 tile pipe laid on the bottom of the reservoir to within 6 ft. of the dam. The intake water for the boiler feed and condenser is drawn from this well so that the water must flow up through the concrete tile pipe from the bottom of the reservoir at its deepest point. While water is being pumped from the river, there is a flow south through this tile pipe and when not pumping, the water is drawn in the opposite direction through it.

The average horsepower developed in the engines during the 21 days of these tests was 2446. Incidentally, it may be stated that there was an average of 3062 bbl. of cement manufactured per day during the same test, or 1 bbl. of cement for 0.8 hp.

The results of the tests during these three weeks are as uniform as could be expected; the heat lost per square foot of surface being slightly less per 1 deg. difference of temperature in cold weather than in warm weather. This difference is undoubtedly due to an increased amount of evaporation during the warmer period.

The average humidity for the different weeks during which the tests were made was 58.5 per cent. in May, 62.3 per cent. in July and 71.2 per cent. in November.

It appears from the tests that under ordinary conditions, in the northern part of the United States with engines using 15 lb. of water per horsepower-hour and a vacuum of 26 in., a reservoir having a surface of 120 sq.ft. per hp. would be ample for cooling and condensing water.

TEST OF HEAT RADIATION FROM SURFACE OF CONDENSER WATER RESERVOIR AT PLANT OF CRESCENT PORTLAND CEMENT CO., WAMPUM, PA.

Area of reservoir, 288,000 sq.ft.; average depth, 5.36 ft.; capacity 1,543,680 cu.ft.=96,480,000 lb.

Date of Tests	Week ending May 7, 1911	Week ending July 12, 1911	Week ending Nov. 27, 1911
Amount of water pumped from river, lb.	10,458,956	29,050,875	4,648,140
Average temperature of river water, deg. F.	57.5	77	36
Average temperature of intake to power house, deg. F.	72.75	91.43	61.71
Average temperature of tail water from condenser, deg. F.	101.36	129.43	90.71
Average temperature of reservoir, deg. F.	87.05	110	76.71
Average temperature of air, deg. F.	51	78.43	33.3
Average difference of temperature between water and air, deg. F.	36.05	31.57	43.41
Change in temperature of reservoir during test, deg. F.	0.25	7	2
Three 26 and 52 by 36 in. compound condensing engines, hp.-hr.	397,807	411,320	426,936
Two 12 and 21 in., 11½ and 24 by 24-in. air compressors, hp.-hr.	46,536	46,536	46,536
Average vacuum at 30 in. barometer, in.	26.4	22.2	26.6
Average water consumed by engines, per hp. per hr., lb.	14.46	15.64	14.41
Average water consumed by compressors per hp. per hr., lb.	18.85	20.12	18.83
Steam condensed by engines, lb.	5,752,289	6,433,045	6,145,148
Steam condensed by compressors, lb.	877,204	936,314	876,273
Latent heat of steam condensed, lb.	1024.7	1007.1	1026
Heat delivered to reservoir by engines, b.t.u.	5,894,370,000	6,478,720,000	6,304,922,000
Heat delivered to reservoir by compressors, b.t.u.	898,871,000	942,961,000	899,056,000
Heat to raise river water to average temperature of reservoir, b.t.u.	309,062,000	958,679,000	189,226,000
Heat given up or retained in reservoir during test, b.t.u.	24,120,000	675,360,000	192,960,000
Heat reduction in reservoir due to rain, b.t.u.	21,630,000	56,700,000	46,200,000
Heat absorbed by air and evaporation during seven days, b.t.u.	6,438,429,000	5,730,942,000	6,564,509,000
Heat absorbed by air and evaporation per sq.ft. of surface, seven days, b.t.u.	22,356	19,899	23,495
Heat absorbed by air and evaporation per sq.ft. per hr., b.t.u.	133.1	118.4	139.8
Heat absorbed by air and evaporation per sq.ft. per hr., per 1 deg. difference b.t.u.	3.69	3.71	3.22

The United States, Great Britain and Germany produce more than 80 per cent. of the world's total supply of coal.

Nugent Shaft Oiler

The latest oiling device brought out by William W. Nugent & Co., 220 to 222 West Huron St., Chicago, Ill., is shown herewith. It consists of a pump, can and a nozzle long enough to reach the shaft hanger to be oiled. The operator con-

NUGENT HAND OILER

trols the amount of oil used and may deliver a drop or a teaspoonful, or such quantity as may be necessary.

This oiler does away with the necessity of using a ladder, saves time and eliminates the danger of accidents to the man caring for the bearings.

Beware the blandishments of the scientific management expert who can look at the tongue of your wagon, feel the pulsation of your boiler-feed line, take the temperature under your collar, and prescribe offhand a sure-cure system, whose principal principle is "change everything."—*Chained Lightning.*

Keys and How to Remove Them

By George Rice

A man tried to remove a steel key from a crankpin and shaft by striking the arm with a hammer. He had previously tried to remove the key by the common method of driving it back, but had failed. He had also poured oil on the parts and tried other means to free the obstinate key. Then he lost his temper. It does not do to get impatient with keys, as they are pretty sure to hold tightly when driven correctly for that is what is expected of them. Nevertheless, this man sought to remove the key by banging at the parts with the hammer, but he had not delivered many blows when the cast-iron arm broke as shown at *A*, Fig. 1.

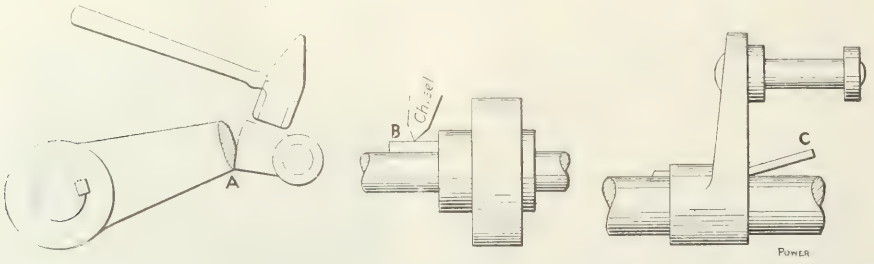


Fig. 1. Removing Keys

Another man used the sharp edge of a cold chisel on the part of the key extending beyond the hub, as at *B*. Consequently the edge of the cold chisel was sunk well into the steel and the key was ruined when it was finally removed.

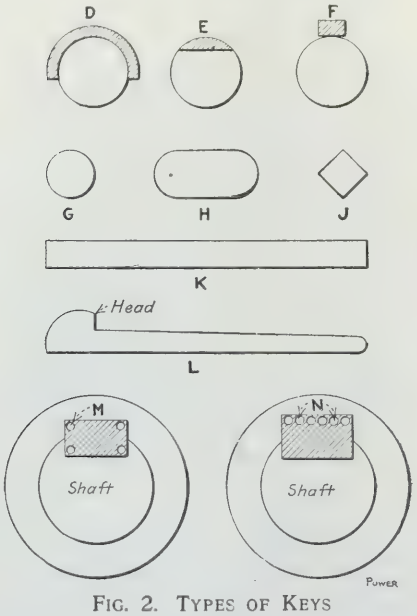


Fig. 2. Types of Keys

A common method of freeing refractory keys is to drive them through with a like key from the other side, as shown at *C*. This method is often used and is a good way of driving out stubborn keys when it is possible to get at the sides of the work. It will, however, be necessary to use a driving key of smaller dimensions than the keyseat; otherwise the driving key will bind so tightly in the seat that there will be trouble in getting it free.

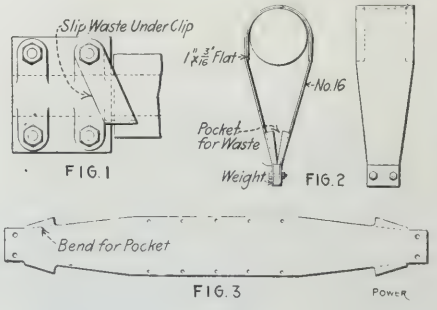
Among the numerous types of keys in use is the half-moon-shaped key that covers about one-half of the shaft, as shown at *D*, Fig. 2. This key when once set is quite hard to remove. Then there is the key made to fit on a flat section of the shaft, like *E*, also the bothersome form of key *F* that is calculated to bind so tightly in the seat of the hub that the frictional contact with the shaft will cause the hub to hold firmly in place.

Still other forms are the round key *G*, the extended oval key *H* and the diamond-shaped key *J*. If the key has a head, *L*, the chances are that it can be pried from its seat with a cold chisel. If it has no head, *K*, and cannot be otherwise removed, drill it out by drilling the corner holes as at *M*. Instead of drilling holes at the corners of the key, holes are sometimes made along the upper side of the key as at *N*. The key is then easily removed.

Keeping Stray Oil Off Belts

By James Clark

In a plant having a number of pumps for high-pressure service, considerable trouble was experienced from the oil which would work out of the bearings and follow the shaft to the wheel hub to be thrown off by centrifugal force and land on the inside of the wheel rim. From there it would creep to the outside and get on the belts. Oil would also work out between the tight and loose pulleys and get on the belts in the same manner.



To overcome the trouble caused by the oil from the bearings, some clips of No. 16 sheet steel were made and fastened between the two nuts which held the bearing cap, as in Fig. 1. The end of the clip was bent for holding a small bunch of waste against the end of the hub to continually wipe off the oil.

To take care of the oil from the loose pulley some oil catchers were made as shown in Fig. 2; Fig. 3 is a development in the flat. These were made of No. 16 sheet steel. The curved end which fits the hub was bent to a radius of about ½ in. larger than the hub and fitted with a strip of 1 x 1/16 in. flat on each side. These strips are to prevent the catcher from coming in contact with the hub at the point where the oil works out; otherwise the oil would creep between the catcher and the hub to the spokes and down to the belts. The width of the curved end should be about ½ in. less than the distance between the spokes of the tight and loose pulleys, and the length from the center of the shaft to the end about 12 to 18 in., depending on the size of the pulleys. The other end of the catcher is fitted with a weight to prevent it from turning with the wheels; it also has a pocket to hold the waste. These devices are very simple to make and they completely cured the oil trouble.

City Smoke Ordinances and Smoke Abatement

The Bureau of Mines has just issued Bulletin 49, entitled "City Smoke Ordinances and Smoke Abatement," by Samuel B. Flagg. Some space on factors affecting smoke conditions in cities and the status of smoke abatement work in the United States introduce the pamphlet.

The exact form that the ordinance of any particular city should take can be decided upon only after a study has been made of existing local conditions. Certain features are desirable, however, and certain others are undesirable, and in the comment in the bulletin on the city ordinances some of the good and bad features are noted. Two specimen forms are presented, one adapted to the average conditions in a city of 200,000 population, and the other for a city of smaller size, ranging from 50,000 to 200,000 inhabitants. For the information of persons wishing to know the exact phraseology of some of the smoke-abatement ordinances, and especially those containing features of particular interest, the smoke-abatement ordinances of the cities of Chicago, Pittsburgh, Des Moines, Milwaukee and Los Angeles are printed at length, as well as the State act that applies to the city of Boston. Of these the Chicago ordinance is perhaps the most comprehensive and most complete of those in effect in the large cities of the United States. Its requirements apply particularly to the needs of a great industrial center where coals high in volatile matter are the chief available fuel.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

The Electric Motor

The operation of an electric motor is due to the fact that a wire extending across a magnetic field will tend to move sidewise through the magnetic field if an electric current be passed through it. Reference to Fig. 92 will help the reader to understand this statement. If a current be passed through the wire *C* in one direction, it will move upward across the magnetic lines of force, unless restrained;

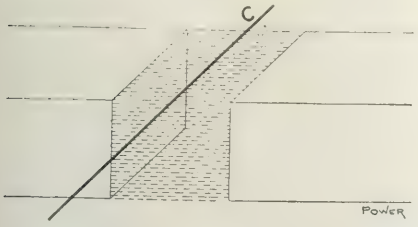


Fig. 92

if the current be passed through the wire in the opposite direction, the wire will tend to move downward.

Just why this is true has never been fully explained with scientific accuracy. One explanation, which is sufficient for ordinary purposes, is that magnetic lines of force under certain conditions act like rubber strings; when they are deflected from their natural path they try to pull back into that path, and thereby exert a force upon whatever deflects them. Moreover, if magnetic material

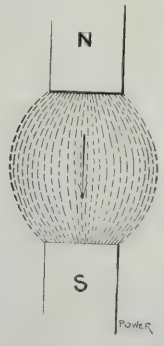


Fig. 93. Normal Flux

or another magnetic flux be located within the natural path of a set of magnetic lines of force, the latter will try to pull the material or the other flux into such a position as to reduce the "reluctance" (magnetic resistance) of their path to the lowest possible amount. Both of these properties enter into the explanation of the movement of a current-carrying conductor across a magnetic field.

Reaction between Magnetic Fluxes

Fig. 93 represents the passage of magnetic flux from one pole to the other of

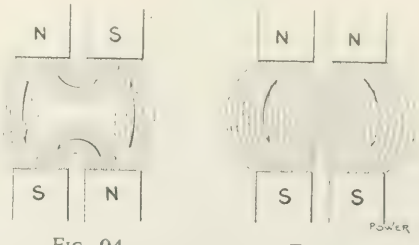


Fig. 94 Fig. 95

Reactions between Two Magnetic Fluxes

a simple magnet (ignoring stray lines and leakage fluxes). If another magnet be located near the first one with its "north" pole alongside the other's "south" pole, the fluxes of the two magnets will unite to a large extent and the distribution of the fluxes will be somewhat as indicated in Fig. 94. Under this condition the two fluxes will try to pull the magnets together, because this would shorten the magnetic paths.

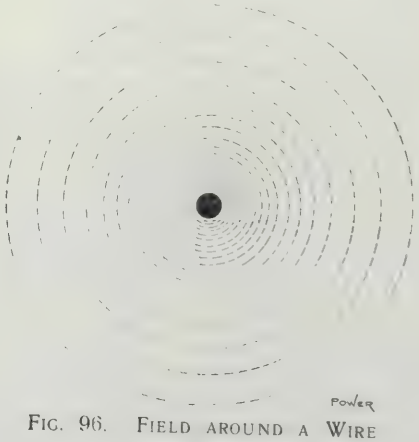


Fig. 96. Field around a Wire

If the second magnet be turned around so that the two "north" poles are alongside each other and the two "south" poles similarly placed, then the fluxes will oppose each other and the distribution will be somewhat like that represented in Fig. 95. The crowding of the fluxes in the space between their fields will tend to push the magnets farther apart, because that would remedy the deflection and shorten the magnetic paths.

Figs. 96 to 102 illustrate the elementary application of these principles to electric-motor conditions. As explained in an early lesson on the electromagnet, passing an electric current through a conductor produces a magnetic flux around the wire, as indicated in Fig. 96. Fig. 97 represents a conductor placed near the field of a magnet and carrying a current in such a direction that the circular flux has the direction indicated by the arrow. The circular flux, therefore, agrees with the magnet's flux where the two mingle, and the density in this part of the field is greatly increased. This crowds the

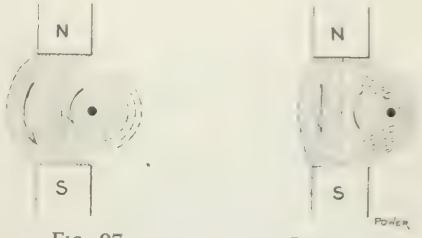


Fig. 97 Fig. 98

Field Distortion by Current-carrying Wire

magnet's flux over to the left, and the lines of force, acting like rubber strings, tend to push the wire horizontally to the right and thereby restore the natural path, by the shortest route, between the magnet poles.

Fig. 98 illustrates the opposite condi-

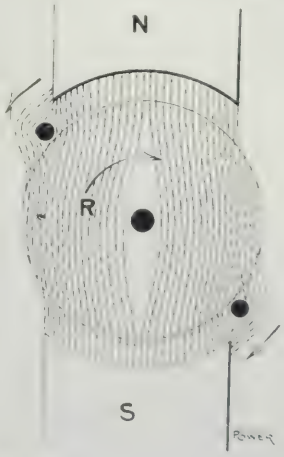


Fig. 99. Rotary Pull of Armature Wires

tion. The current in the conductor is in the reverse direction and the flux therefore circles around the wire in a clockwise direction. This forces some of the flux from the magnet out of the normal field and around the outside of the wire, and the tendency here is to pull the wire

further into the field because that will relieve the distortion of the main flux and shorten its path.

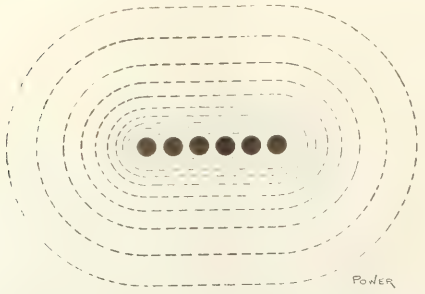


FIG. 100. FIELD AROUND GROUP OF WIRES

If the two wires be put on an armature core and are located as shown in Fig. 99, with current passing through them so as to produce the circulation of magnetic flux indicated by the small arrows, the

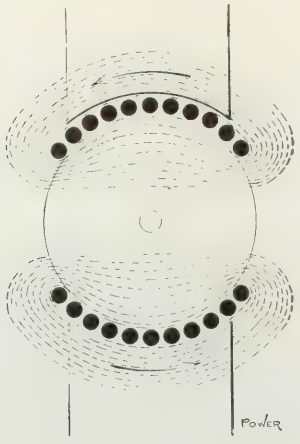


FIG. 101. FLUX DUE TO ARMATURE CONDUCTORS

condition represented in Fig. 98 will be produced in duplicate and the armature will be pulled in the direction shown by the arrow *R*. The step from this arrange-

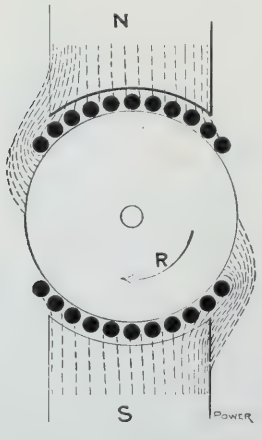


FIG. 102. FIELD DISTORTION BY ARMATURE CONDUCTORS

ment to a complete winding requires a little more elementary discussion.

If several wires be placed side by side and currents be passed through them in the same direction, a magnetic flux will be set up surrounding the whole group, as indicated in Fig. 100. With two such

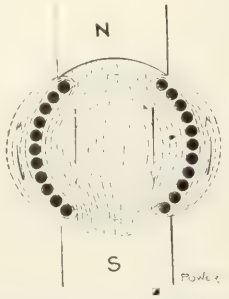


FIG. 103. POSITION OF MAGNETIC STABILITY

groups placed on opposite sides of an armature core and currents passing through the two groups in opposite directions a condition about like that illustrated in Fig. 101 will be obtained. So long as there is no flux set up by the field magnet the armature will remain stationary. If, however, the field magnet be energized so as to produce the polarity indicated in Fig. 102, the field flux will be distorted by the armature flux some-

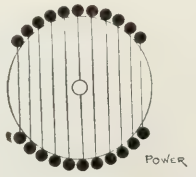


FIG. 104

what as shown and the armature will be pulled around clockwise until it reaches the position indicated in Fig. 103, where the armature flux can combine with the field flux without distortion.

In order to pass current through the conductors around the armature core in Figs. 101 to 103 in the manner described, they would naturally be connected to-

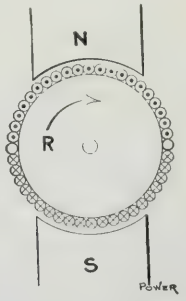


FIG. 105. FLOW OF CURRENT IN ARMATURE WIRES

gether, somewhat as indicated in Fig. 104, and the same current passed through the whole coil. With such an arrangement, however, the armature would turn no farther than the position shown in Fig. 103. To correct this defect, the armature core is provided with conductors all the way around the periphery and these are connected to a device called a "commutator" in such a way that the current passes through all the conductors on one side of the armature always in the same direction and through those on the opposite side always in the opposite direction, as indicated in Fig. 105, where the crosses signify current flowing away from the reader and the dots current flowing toward him. The action of the commutator will be explained fully in a future lesson.

TORQUE OR ROTARY EFFORT

The tendency of a motor armature to rotate under the influence of the magnetic fluxes is called "torque," which is measured in pound-feet (in English units); that is, pounds of "rotary" pull multiplied by the radius (in feet) at which the pull is exerted. For example, if the pull at the rim of an armature 12 in. in diameter were 100 lb., the torque would be

$$100\ (lb.) \times \tfrac{1}{2}\ (ft.) = 50\ lb.\text{-}ft.$$

If the torque is known, the rotary pull

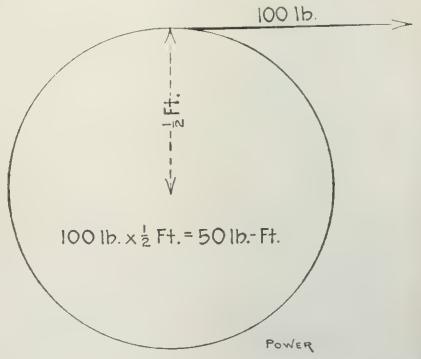


FIG. 106. TORQUE DIAGRAM

at the wires can be calculated from that and the armature radius, thus:

$$\frac{50\ lb.\text{-}ft.}{\frac{1}{2}\ ft.} = 100\ lb.$$

pull at the edge; see Fig. 106. This means the total pull of all the wires around the armature. If there were 100 wires, the average pull exerted at each wire would be 1 lb. The pull on the wires under the pole faces is greater than that on the wires outside, but that is not of much practical importance; it is the total rotary effort or torque that concerns the operator.

The torque of an armature depends on the number of armature wires, the current in each wire and the quantity of magnetic flux supplied by the field magnet. Multiplying the total flux passing between the armature and all of the magnet poles by the number of wires and by the current per wire, and dividing the final product by 852,230,000 gives the torque in pound-feet tending to rotate the armature. Expressed as a formula:

$$\frac{Total\ flux \times wires \times current}{852{,}230{,}000} = torque$$

in pound-feet, *not the pull in pounds.*

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gas Engines at the Central Furnaces

By A. D. Williams

The plant of the American Steel & Wire Co., at Cleveland, Ohio, is known as the "Central Furnaces" and consists of four blast furnaces, the latest of which *D* was completed about 15 months ago. Power for the three earlier furnaces was furnished by a steam plant containing seven blowing engines and six engine-driven direct-current generators. In addition to this four gas-engine driven units, op ing on blast-furnace gas, supplied alternating current to the Cuyahoga and the Newburgh works of the company. This gas-engine installation has now been in operation about five years. With the addition of furnace *D*, however, two large gas-blowing engines—the largest in America—were installed to operate upon the gas from this furnace.

Fig. 1. Showing Blast Stoves and Gas-cleaning System of Furnace *D*

The general arrangement of furnace *D* is shown in Fig. 3. It is designed for an output of about 500 tons of pig metal per day, the entire output being run into ladle cars for transportation to the pig-casting machine or to the steel mills. The pig machine is used only for Sunday metal. Some of the ladle cars used can be seen in Fig. 1.

Gas-cleaning System

The furnace gases are taken off by four bleeders 3 ft. 9 in. in diameter connecting with four downcomers at a point about 18 ft. above the top of the furnace. (See Fig. 3.) On each side of the furnace the two downcomers unite in a single one and the two large downcomers connect at opposite sides of a primary dust-catcher 30 ft. in diameter by about 50 ft. high. This, together with the centrifugal dust-catchers, are located over a track and the collected flue dust is dumped into cars. This dust forms a part of the regular furnace burden.

From the primary dust-catcher the gas passes out at the top through a riser which makes a sharp turn into an 8-ft. main leading to the large centrifugal dust-catcher. Up to this point the gas passages are lined with 9 in. of firebrick as the temperature of the gases leaving the furnace is from 300 to 400 deg. F. There are one 14-ft. and two 12-ft. centrifugal dust-catchers arranged in series. A few furnaces have been equipped with a secondary dust-catcher, but the use of four is a new departure. They work on the same principle as a steam separator, differing only in size. The gas enters the upper part of these chambers on one side, the gas main forming a tangent to the annular passage. This imparts a whirling motion to the gas which then passes down through the annular passage and turns sharply upward through the inner shell, out the top and down to the tangential entrance of the next. The dust falls down the sides of the shell into the dust chamber, which is protected by a conical cover from the direct current of gas. This conical cover also serves to direct the gas upward to the outlet of the dust-catcher. The construction of this apparatus is shown in Fig. 2. At the bottom there is a counterweighted bell valve with an inclosing spout to concentrate the stream of dust. A 2-in. spray pipe in the spout serves both to cool the dust and keep it from being blown all over.

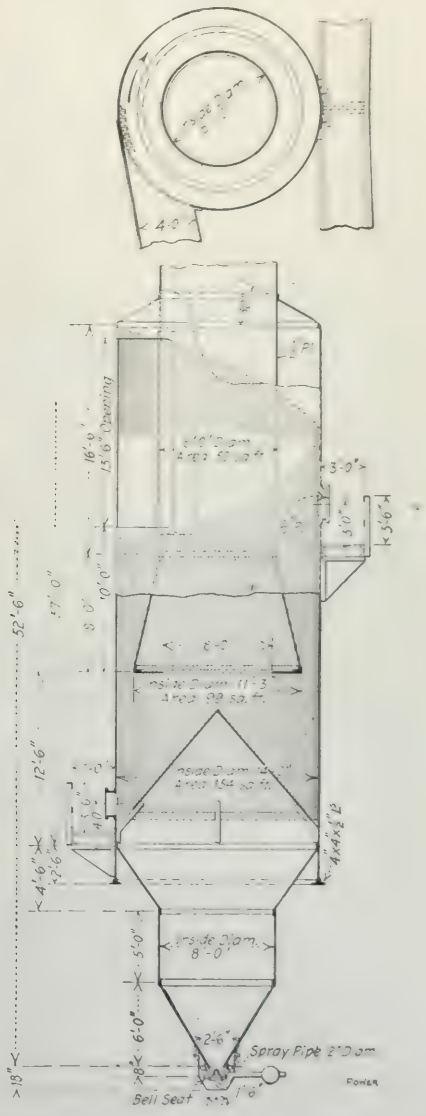


Fig. 2. Secondary Centrifugal Dust Catcher

After leaving the last 12-ft. dust-catcher the gas passes through a goggle valve which permits closing furnace *D* off from the gas system. The valve connections at this point consists of four

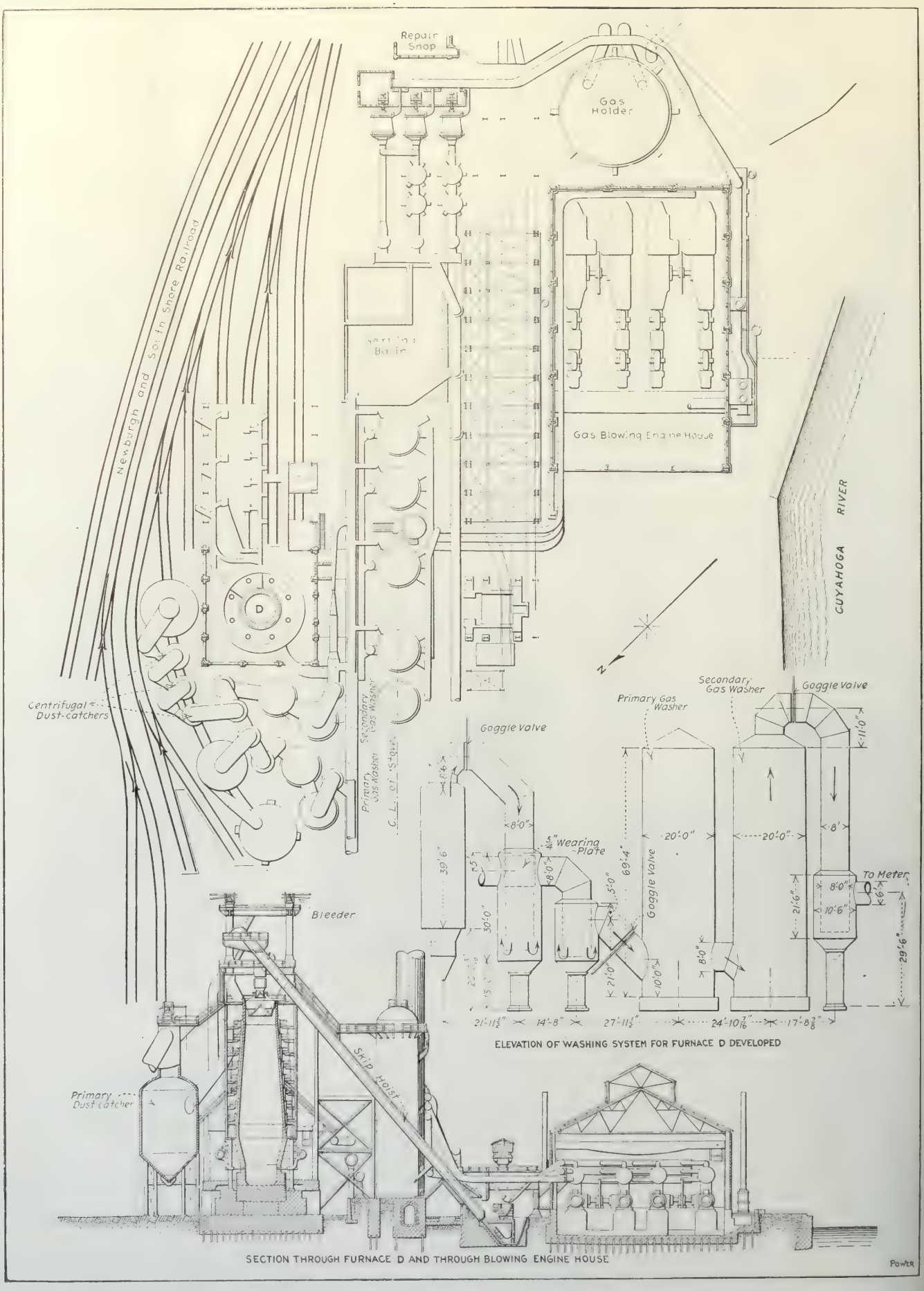


FIG. 3. PLAN AND SECTIONAL ELEVATION OF FURNACE *D* AND BLOWING EQUIPMENT AND DEVELOPED ELEVATION OF WASHING SYSTEM

water-seal valves and five goggle valves, so arranged that either furnace *C* or *D* can be cut off from the gas system, permitting either of the two sets of primary and secondary gas washers to be cut out for cleaning or repairs and the flow of gas from both furnaces can pass through one set of washers, or one set of washers may serve one furnace.

The primary gas washer, Fig. 4, is 20 ft. in diameter by 69 ft. high. The gas enters at the bottom and passes up an annular passage through a number of conical baffle-plates over which the wash water flows. At the top of the washer the gas turns downward through a central passage leading to the gas outlet. The wash water enters at the top, falling into a small distributing basin from which it flows on to the distributing cone through an opening around the bottom of the basin and by overflowing its edges. The distributing cone directs the sheet of water onto the baffle-plates over which it flows from baffle to baffle, falling through the ascending stream of gas. The water is then carried off through a trap or gooseneck and flows to a settling basin from which the flue dust can be recovered.

The secondary gas washer, Fig. 5, is the same overall size as the primary washer. The gas enters near the bottom and flows upward through a series of internal and external baffling cones and wire-net diaphragms and passes out at the top. Wash water is fed into the top by 32 inlets which turn horizontally in a trough encircling the top of the washer. The water overflowing the edge of the trough runs down an inverted cone and over a vertical wire-netting diaphragm through which the gas passes to a cone suspended from the inverted cone, thence over a second netting diaphragm, and so on, until it finally reaches the bottom and passes out through a trap to the settling basin. There are ten wire-netting diaphragms in this washer giving a large amount of washing surface.

After leaving the secondary washer the gas passes through a 48-in. venturi meter, arranged with a bypass for emergency use. Part of the gas is drawn off, after passing through the meter, for

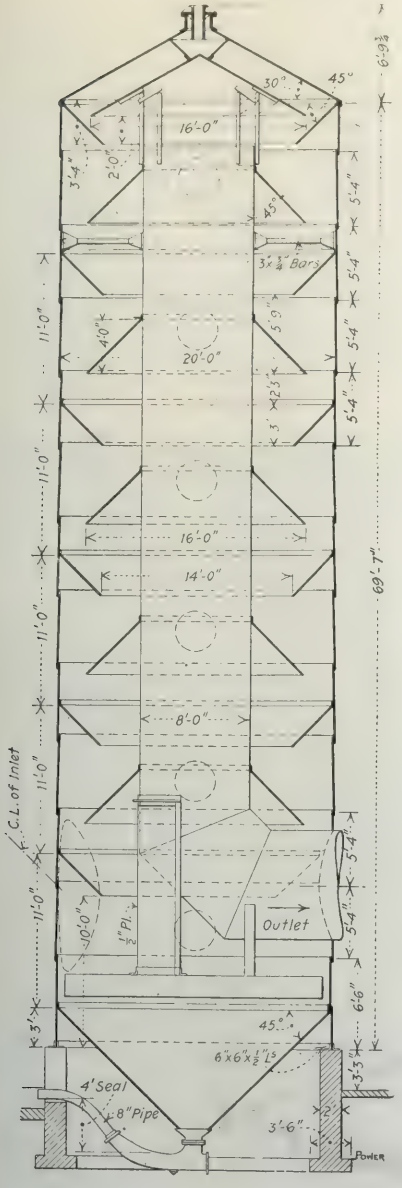


FIG. 4. PRIMARY GAS WASHER

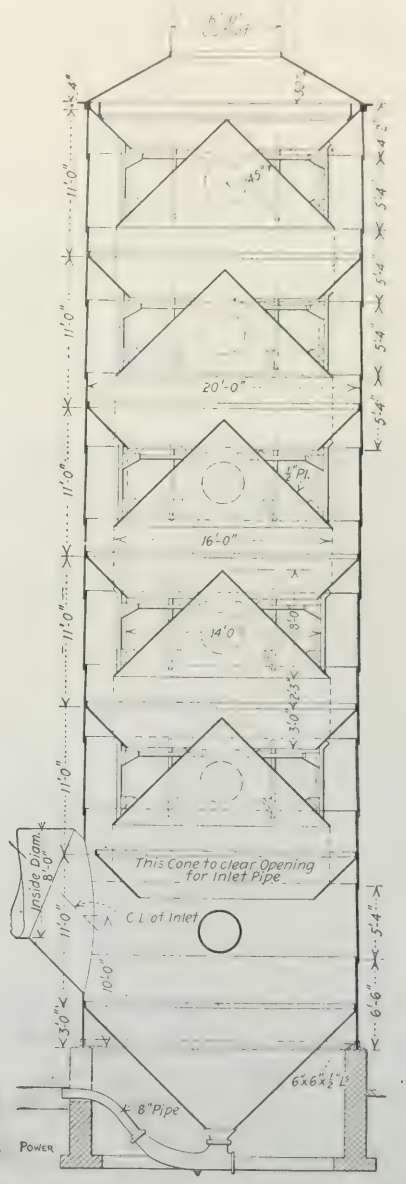


FIG. 5. SECONDARY GAS WASHER

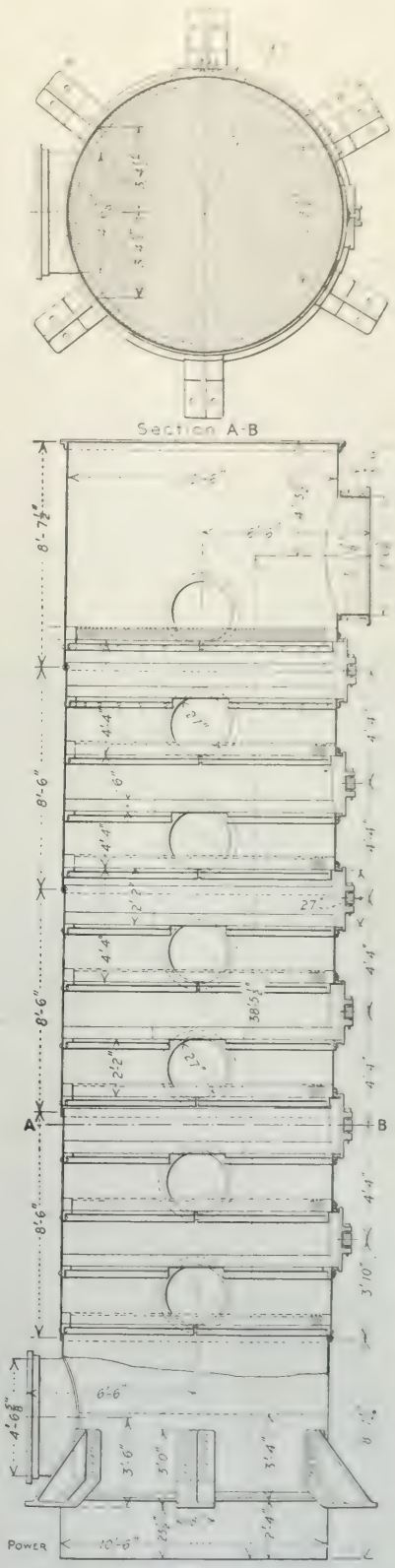


FIG. 6. ZSCHOCKKE GAS WASHERS

heating the hot blast stoves, and the remainder passes to a manifold thence through a water seal and a goggle valve to the baffle washer. This portion of the gas system is shown in Fig. 1. The gas enters at the bottom and passes up an annular passage obstructed by internal and external cone baffles with ¼-in. mesh screens in the openings between the baffles. It then passes down through the central gas passage to the outlet of the washer. Wash water enters at the top and falls into a distributing basin similar in construction to that used in the primary gas washer.

From the baffle washer the gas passes to the Zschockke washers, shown in Fig. 6; these are 10 ft. 6 in. in diameter and 43 ft. high. The gas enters at the bottom and passes up through a series of 13 wooden grilles, kept wet by the descending water, and leaves the washer near the top. Water enters through 12 distributing nozzles and leaves at the bottom. The wooden grilles can be removed through the clean-out doors.

Gas from the Zschockke washers goes to the Theissen washers, which are of the usual construction. These are operated by 150-hp. direct-current motors, but it is the intention to change to alternating-current motors as the constant load characteristics of the Theissen washers is particularly favorable for alternating-current operation. They receive the gas at a pressure of 11 to 16 in. of water and build it up to the holder pressure at which it flows to the gas engines.

The holder is 55 ft. in diameter with a 14-ft. lift and under normal conditions is kept full. From the holder the gas flows to the gas engines in the alternating-current power house and to the gas-blowing engines. The line to each engine house contains a 60-in. venturi meter which measures and records the amount of gas used.

The successful operation of the gas engines in this plant is largely due to the ample gas-cleaning equipment. Table 1 shows the average conditions of the gas for one month, during which period the furnace was working with Mesabe ore and flue dust comprising nearly 72 per cent. of the burden. Mesabe ore is extremely fine and frequently gives trouble in the furnace, even when there are no gas engines to be considered. In this case the gas carried 1.333 grains of dust per cubic foot at the entrance to the clean gas washer and 0.0185 grains of dust in the engine gas. The dust in the gas leaving the secondary dust-catchers was so fine that all passed through a 100-mesh sieve and the scum on the waste water from the gas washers passed through a 200-mesh sieve. This scum had the following composition: coke, 33.33 per cent.; silica, 15.02 per cent.; iron, 47.33 per cent. The dust taken from the gas engines had the following composition:

	Inlet.	Cylinder head.
Si	40.68	35.35
Fe_2O_3	22.03	18.70
Al_2O_3	5.70	9.50
CaO	10.21	7.68
MgO	6.40	4.20
Coke	12.34	23.48

TABLE 1. AVERAGE ANALYSIS OF GAS FOR ONE MONTH

	CO_2	O_2	CO	H	CH_4	N
Raw gas, per-cent	12.7	0.36	26.3	2.7	0.22	57.5
Clean gas, per-cent	12.7	0.36	26.2	2.8	0.22	57.6
Fine gas, per-cent	13.0	0.40	26.9	2.9	0.22	57

Atmospheric temperature	34 degrees
Barometer	29.66 in.
Grains of moisture per cu. ft. in air	2.41
B. t. u. in engine gas, computed	101.1
B. t. u. in engine gas, by calorimeter	93.8
Flue dust in grains per cubic foot in clean gas washer	1.333
before baffle washer	0.980
after baffle washer	0.361
percentage removed and efficiency	58.45
after Zschockke washer	0.222
percentage removed	13.80
efficiency	31.66
after Theissen washer	0.0185
percentage removed	24.45
efficiency	90.47
in engine gas	0.0185
Moisture in grains per cubic foot in clean gas	26.65
Moisture in grains per cubic foot in engine gas	0.528
Clean gas temperature	162 degrees
Furnace burden, mesabe ore, percent	67.53
Furnace burden, flue dust, percent	4.34

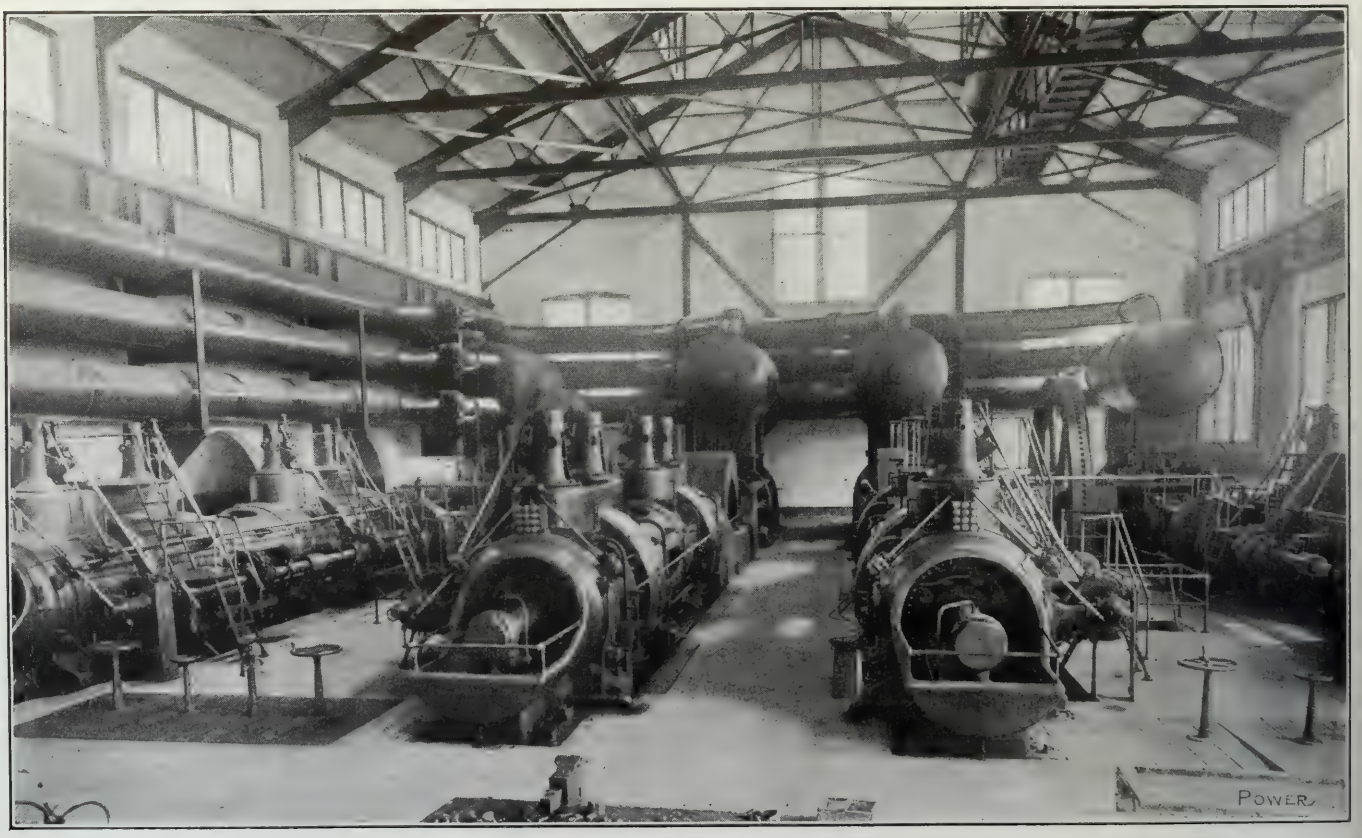


FIG. 7. GAS-DRIVEN BLOWING ENGINES AT CENTRAL FURNACES

BLOWING ENGINES

In Fig. 7 are shown the gas-driven blowing engines which consist of two twin tandem double-acting engines built by the Allis-Chalmers Co. having 44-in. gas cylinders and 60-in. stroke with air tubs of the Slick type, 80 in. in diameter. These engines have 23-ft. flywheels built

up of eight segments and weighing 137,000 lb. They are designed to operate against a normal blast pressure of 15 to 22 lb. and a maximum pressure of 28 lb. Each has a capacity at 60 r.p.m. of 40,000 cu.ft. of free air per min. and at 75 r.p.m. will deliver 50,000 cu.ft. per min.

A clearance of 2 per cent. is provided in the air tubs. The air-inlet ports have an area of 19.5 per cent. of the piston area and are arranged around the periphery of the tub. The outlet valves have an area of 17 per cent. of the piston area. There are four 18-in. valves in each end of the tub actuated by eccentrics on the main shaft of the engine; see Fig. 8. These valves open automatically as soon as the pressure in the cylinder exceeds the blast pressure and are closed positively before the air-inlet valve starts to open.

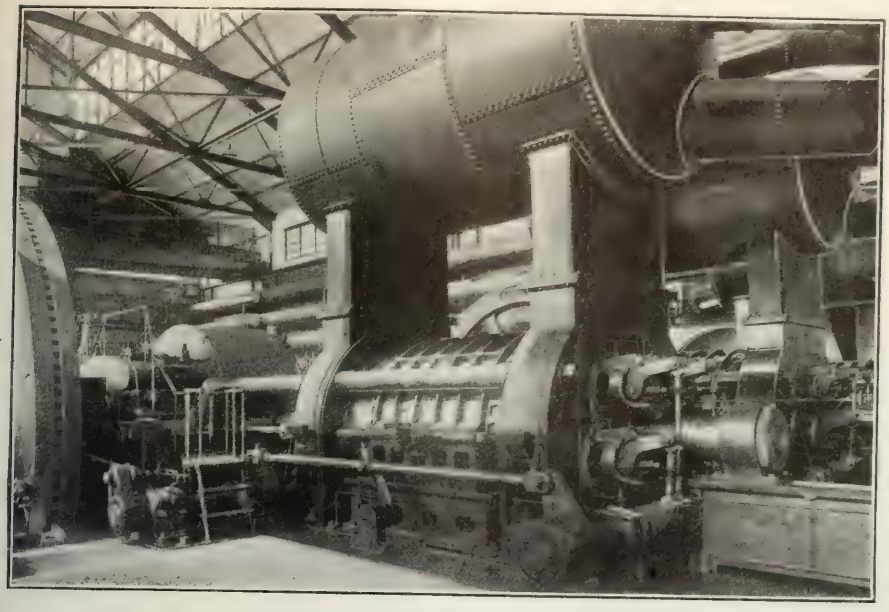


FIG. 8. AIR RECEIVER AND COMPRESSOR OR TUB

Above each air tub is a receiver of 1000 cu.ft. capacity from which the cold blast mains are taken off. There are three of these mains, one leading to furnace *D*, one to furnace *C* and one to furnace *B*, with gate valves operated by pendant hand chains from the floor. The cold blast mains are shown along the wall in Fig. 7. The arrangement of these mains permits the engines to blow any one of the three furnaces while a blast main from the steam-blowing engines to furnace *D* is provided for use when the gas engines are not in operation. The cold blast main connects with the four stoves in which the blast is heated and from which it passes to the hot blast main, thence to the bustle pipe from which it is taken off to the tuyere for the furnace.

The gas from the holder is brought to the engines by a 60-in. main which tapers to 48 in., a 24-in. branch being taken off to each side of the engines. From each side of the engines an independent 30-in. exhaust line leads to a muffler outside the building. These mufflers consist of a hollow cylinder 5 ft. in diameter by 12 ft. high with a 30-in. escape pipe about 50 ft. high.

Compressed air is used for starting the engines and a small compressed-air barring cylinder is provided for turning the engine over slowly. Air for this purpose is supplied by a two-stage air compressor driven through a Morse chain by a 50-hp. direct-current motor. The compressor has an automatic pressure con-

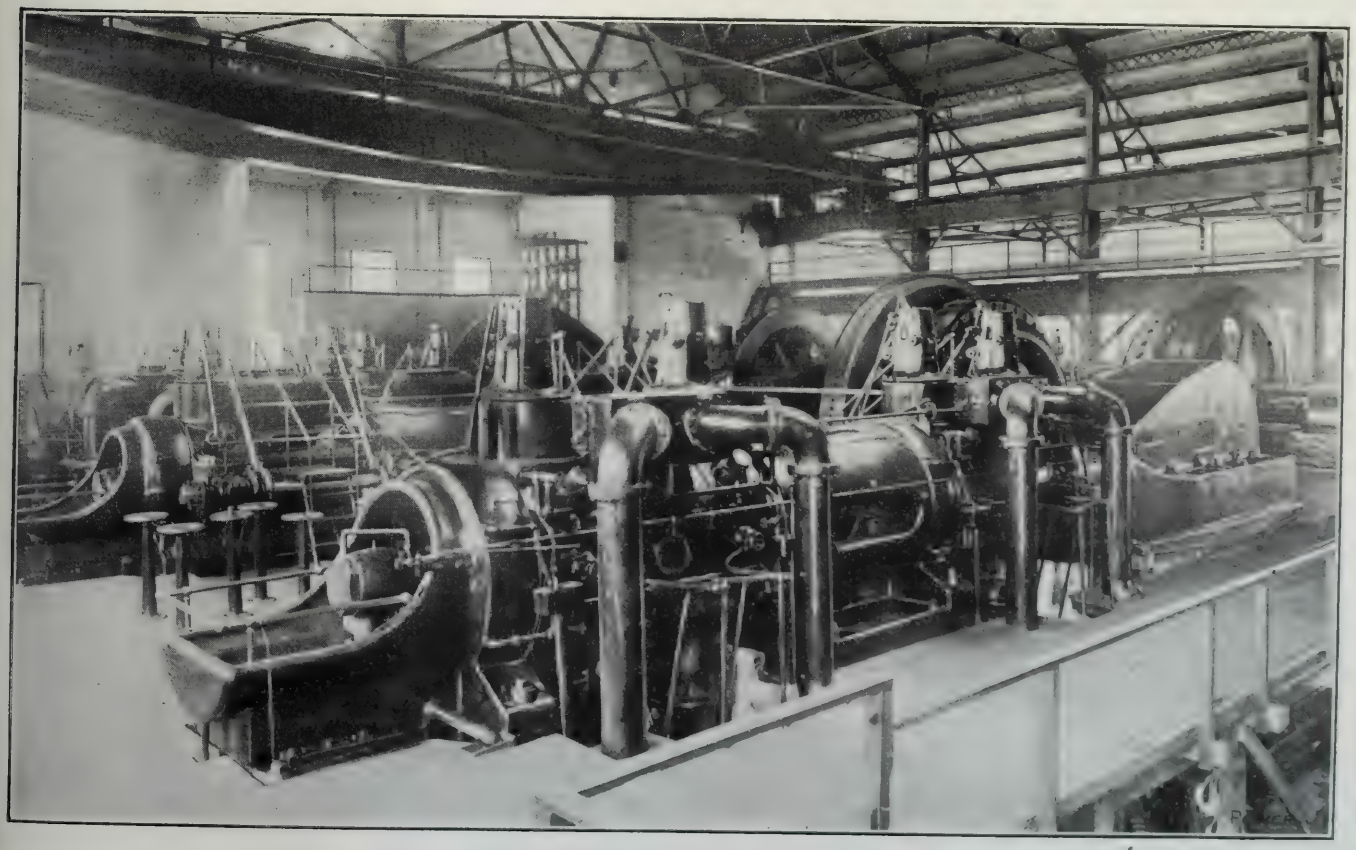


FIG. 9. GAS ENGINES DRIVING ALTERNATING CURRENT GENERATORS

troller so that the 250-cu.ft. air receiver is always kept full, and it has sufficient capacity to fill the tank in one minute.

A duplex ignition system is provided, there being two igniters in each end of each cylinder with a telltale lamp connected in series with it. A telltale lamp is also connected across the ignition circuits in such a manner that any break or current failure is immediately indicated and localized. The ignition current is supplied by a small motor-generator set.

The cooling water for the engines is drawn from the regular service supply of the furnaces upon which pressure is maintained by an 18-ft. water tower 101 ft. high.

Forced-feed lubrication is used, the oiling system being supplied from tanks supported in the roof trusses of the building and maintaining a static head of approximately 40 ft. Automatic feeders regulate the amount supplied to each bearing and the waste oil flows to tanks in the basement from which it is returned by pumps to the pressure tanks. The system is a duplex one, two pressure tanks being provided, with valve connections arranged to permit of shifting from one supply to the other. The pressure tanks hold about 200 gal. each.

These engines are close to the river and heavy piling was required in constructing the foundation. The site of the foundations was first inclosed with steel sheet piling and excavated, after which wooden piles were driven at close centers over the entire area with their tops cut off below the ground water level. The bottom of the excavation was carried below the point of cutoff for the piling a sufficient depth to give room for a cinder fill around the upper portion of the piles below the concrete mat which incloses the tops of the piles, making a raft upon which the piers supporting the engines were constructed.

Alternating-current Power Plant

The alternating-current power plant; see Fig. 9, contains four generating sets, each consisting of an 1880-hp. 34x42-in. twin tandem double-acting Allis-Chalmers gas engine direct-connected to a 1000-kw. generator. These engines have 18-ft. flywheels, each weighing 90,000 lb., in addition to the flywheel capacity of the rotating generator fields. There are two main bearings 23x42 in., each of which weighs 80,000 lb., the frames supporting these being the heaviest single pieces in the engines. The total weight of each unit is 740,000 lb.

The compressed-air starting equipment consists of a 4½x4½-in. electrically driven compressor drawing its supply at 90 lb. pressure from the furnace-air power lines and compressing it to 200 lb. in two 520-cu.ft. storage receivers. Two 1-kw. motor-generator sets convert the 220-volt direct-current supplied by the direct-current power plant to 90 volts for the ignition circuits.

The alternators are of the revolving-field type with 28 poles and deliver three-phase current at 25 cycles, 13,200 volts and 51.5 amperes per lead. Direct current for field excitation is obtained from either one of a pair of 100-kw., 250-volt, steam-driven units in the direct-current power house.

The lubricating system in this power house is the same as that in the gas-blowing engine house.

Practically the entire load on this plant consists of motors and the power factor is rarely better than 85 per cent., averaging 80 per cent. and occasionally dropping to 70 per cent. The total day load averages 3000 kw. and the night load 2000 kw. The gas consumption averages 237 cu.ft. per kw.-hr., and the oil consumption for the cylinders is 0.42 gal. per 1000 kw.-hr., and for other lubrication 0.22 gal. per 1000 kw.-hr.

The cylinder oil is recovered and used as engine oil. There have been no interruptions of service due to the gas engines, all service interruptions thus far being due to the opening of the circuit-breakers or to trouble on the line.

As previously stated, the alternating current is supplied to the Cuyahoga works and the Newburgh wire works of this company. Some of this energy will be used at the furnaces when the improvements now under way are completed, as it is the intention to operate some of the machinery upon which the load is nearly constant by alternating-current motors. The alternating-current plant can be tied in with the direct-current plant through a reversible motor-generator set for supplying direct current at times when the demand for it is in excess of the generating capacity and the direct-current plant can carry the alternating-current load at times of light demand.

CORRESPONDENCE

Ignitor Points

In looking over the numerous gas-engine journals, I noticed several advertisements, calling attention to the merits of different kinds of composition metals for gas-engine ignitor points, of the make-and-break type.

I have tried several, and found them good, bad and indifferent; my old standby is machine steel. I have used it for several years on a 600-hp. and a 300-hp. engine with good success; also it is much cheaper.

We keep all the old electrodes, cutting them up in the lathe to suitable points. The holes in the electrodes are then reamed slightly taper, and made standard. We can then make up a quantity of points at a time, and keep them in stock. It takes but a few seconds to punch out old points and replace them with new ones, when the old points become too short.

The points are made about ⅜ in. in diameter on the working face, and as before stated, are made slightly tapered, so they rivet up to a good tight fit in the electrodes, insuring good contacts.

On some ignitors, we make fixed electrodes with no point; just machining the exposed end round but eccentric with the shank, so that a slight turn of the electrode will take up wear and keep the contacts at proper distance.

Again, nearly all gas engines of the better class have bronze bushings in the ignitor body in the outside end, so as to be able to take up wear of the movable electrode. We bore out the ignitor bodies, so that the bushings can be made of ¼-in. brass pipe. These bushings will last a long time. The outside of the bushing is machined to a nice driving fit, the inside is bored out slightly small, and after driving the bushing in place, can be reamed true from the opposite end, and with a small drill the oil hole is put through. To remove the old bushing, tap out with suitable size tap, using a capscrew, nut and washer to draw it.

C. A. Stephenson.

Elko, Nev.

Rating of Gas Producers

The rating of gas producers in horsepower has been abandoned by most manufacturers, as there is no definite relation between the capacity of a producer and the power developed by an internal-combustion engine. The character of the fuel and the gas requirements of the engines affect the rating of the producers.

The rating of producers by the diameter of their fuel bed and by the rate of burning the fuel has been adopted by some. This is the same as the rating of producers by the pounds of coal burned per hour and depends on the gasification of 10 to 15 lb. of coal per square foot of fuel bed.

While the method of rating producers by the fuel burned per hour is much superior to the horsepower rating, it is also somewhat indefinite. The commercial efficiency of a producer, or the ratio of the B.t.u. in the gas to that in the coal will vary within wide limits. Thus with a good anthracite coal 70 or more cubic feet of gas can be made per pound of coal and the commercial efficiency should be about 80 per cent., while with a low-grade bituminous coal it is usually less than 50 per cent.

A better method of rating producers seems to be in B.t.u. per hour. Thus a producer rated at 1,000,000 B.t.u. should be capable of supplying an engine of 100 b.hp. if the gas requirement of the engine is 10,000 B.t.u. per b.hp.-hr.

A. A. Potter.

Manhattan, Kan.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Standardization of Ammonia Fittings

By R. P. Kehoe

This question is now being discussed by manufacturers with the idea of adopting a uniform standard as has been attempted with steam fittings. The movement was started last May at a meeting of the American Association of Refrigeration and a paper was read setting forth the advantages of standardizing dimensions and bolt centers of ammonia fittings. This paper appeared on page 741 of the May 21 issue of Power.

It is claimed that owing to the large variation in the different makes of such fittings, great inconvenience is experienced by users. It is further claimed that manufacturers also find this variation an obstacle in selling their goods to plants already fitted up with a competitor's fittings. The first claim is more important than the second, as the convenience of buyers should be a primary consideration in the betterment of conditions.

There are undoubtedly good reasons for such a step, but there are also equally good reasons in opposition to it. The requirements in the manufacture of ammonia fittings, the existing preference on the part of owners and operators of refrigerating and ice-making plants, and in fact all the conditions bearing on the subject, are entirely different from those surrounding the standardization of steam fittings. In the latter case the field is almost unlimited and the demand for ammonia fittings is a very small fraction in comparison.

The inconvenience now experienced by users is really of small extent. There are not more than half a dozen prominent makers and some of them carry a stock of fittings in various parts of the country. The use of refrigerating and ice-making plants will never be such that it will pay to maintain a stock in smaller cities and towns in which there are only one or two plants in operation.

For example, consider the State of Pennsylvania. There are hundreds of plants spread all over the entire state, but for many years to come the demand will not justify a large stock of ammonia fittings except in Philadelphia and Pittsburgh. Perhaps some may disagree with this statement; if so, let other cities such as Harrisburg, Reading and Altoona be included. The owner of a plant in Erie will still be able to buy fittings in Chicago or Milwaukee with little or no disadvantage.

When changes are being contemplated there is usually a little time to prepare and obtain any necessary fittings. If a sudden demand arises and a certain fitting is required immediately, the expense for a telegram and expressage is a small matter, as this contingency occurs so seldom.

Machine parts, especially those subject to constant wear and motion, often break and show unexpected signs of giving out. These must be obtained far more quickly than ordinary fittings and yet hundreds of owners located a long distance from any manufacturer suffer little inconvenience even from this source.

In the United States there are about 5000 plants of from 25 to 2000 tons or more daily refrigerating capacity. There are many thousands of smaller plants. For 30 or 40 yr. there will be a demand for existing sizes from these plants in spite of the most earnest efforts to promote a standard. Some manufacturers will no doubt find it to their interest to foster this demand. In any event it will be necessary to continue the production of present sizes to an almost indefinite period. In the meantime it is very probable that new methods of making joints will be introduced and all the effort spent on standardization will be wasted.

Since a maker's individuality will be lost in the production of a universal standard of fittings, the tendency will be to make these as cheaply as possible. High-grade material may receive some credit, but in the general run of trade, the cheapest material will usually have the same consideration. It is difficult to determine the quality of cast iron, semi-steel, etc., at a glance and only a careful comparison may discover the difference.

As already stated, the production of ammonia fittings at present is almost entirely from the factories of half a dozen large manufacturers of ice-making and refrigerating machines. These fittings have been developed by years of experience and the closest application to a line of work which is known to be very special and exceedingly difficult to master. The flanges, fittings and valves form a very important part of every equipment. Practically all refrigerating work, whether it be in an ice-making tank or in the auditorium of a public building is performed by ammonia or some other medium circulating in coils of pipe.

Each refrigerating system must therefore have many fittings and these are supplied and installed by the makers under strong guarantees. Such an equipment must be carefully put together and nothing but proper fittings can be used with success. Some of the best grades of fittings have been in use for 20 or 30 years and show no sign of wear. Ammonia will not corrode iron or steel and if such pipe work is painted occasionally it will not wear out. Very seldom are ammonia lines altered or broken into, and in fact blanked openings are usually provided for future additions to avoid the necessity of making alterations in existing mains.

The present movement has not been instituted by ice-machine manufacturers, but by manufacturers of steam fittings. Like all other manufacturers when "standardization" is suggested, they concede its advantages and will endeavor to agree upon a proper basis. It is a good thing, but can be carried to an extreme. The result in the present case is very doubtful. Since manufacturers of ice machines will find it inadvisable to give up their present standards, it will probably mean still another set of standards for different types of fittings, which instead of simplifying conditions will add to the variety. In this event the stock on hand of each individual type and make of fittings will no doubt be proportionately reduced. The user will then be less able to count on quick deliveries than he is at present.

It should also be borne in mind that about 80 or 90 per cent. of ammonia fittings are installed or used in constructing new equipment by the manufacturers themselves. New plants and the largest part of alterations and additions are contracted for to be made up and completed by the builders. Only a comparatively small part of such work is put together by users.

Manufacturers are compelled to guarantee their apparatus, piping and fittings for at least a year. Needless to say, every part is usually of ample strength for the purpose. The reputation of the makers depends on the durability and satisfactory operation of their plants. They cannot afford to take the chance of furnishing flimsy fittings. The material may be a cheap grade and the design may be inferior to others, but the dimensions must be ample to provide sufficient strength for at least a reasonable period of wear. In the effort to overcome com-

petition, fittings may be made as cheaply as possible, but ice-machine builders must be careful to sustain their reputation and poor fittings may lose the sale of large plants.

The claim that builders find it difficult to sell their goods where a competitor's fittings are in use, has little foundation. In fact, large users as a rule have no objection to different makes of fittings in the same plant. If it is a consideration it is usually a minor one. When additions are made the connections to the system are a small detail and the new equipment can have entirely different fittings without inconvenience.

Standardization has done much in developing many lines of manufacture and is almost always embraced as a great benefit. Manufacturers hesitate to oppose it because their motives may be questioned, but the fact remains that in this instance a good principle may be adopted where it can avail nothing.

On the whole, when everything is considered, the question of standardization in the present instance offers no benefit to user or manufacturer. The former would probably have a larger variety to contend with and slower deliveries, while the latter would suffer a heavy expense for patterns and the necessity of tying up more capital in the stock of present standards and also a new one. The only gainers would be manufacturers of steam fittings who desire to enter the field. This may be considered an advantage which might introduce cheaper fittings, but the characteristics and improvements of ammonia fittings can only be developed by those manufacturers who are making a special business of ice-making and refrigerating machinery and are compelled to meet the changing requirements from day to day.

CORRESPONDENCE

Making Calcium Chloride Brine

Mr. Anderson's article on preparing calcium-chloride brine, page 682, Nov. 5 issue, is interesting inasmuch as he goes to such immense trouble to keep impurities out of the brine. I have used calcium-chloride brine at times and never have I seen a case in any of the packing houses, where this brine is used, where so much care was taken for so little gain. We used a 6x12x4-ft. heavy-plank brine tank. The suction pipe from the brine pump entered over the side near one end, passing down to within 8 in. of the bottom, while the discharge pipe from the pump entered over the top of the tank near the opposite end, with an elbow attached to deflect the flow downward. The brine was made by piling the calcium chloride in great blocks in the tank under the discharge pipe, allowing the discharge from the pump to circulate over the calcium and gradually dissolve it. The finer particles are quickly dissolved and the specific gravity when up to 1400 or 1500 per hydrometer is strong enough so that the brine will withstand a temperature of 5 to 8 deg. below zero without freezing. Where an open tank is used the calcium brine will lose its strength very rapidly, and unless some calcium is constantly added or allowed to slowly dissolve in the tank there is danger, in case of an accident to the apparatus, of the brine pipes in the cooler freezing. The greater density of the brine the better, as a much lower temperature can be maintained in the refrigerators, if so desired.

I have not noticed any deposit in the brine tank which came from the calcium, although some scale, due to the corrosion of the pipes, accumulates. A suitable strainer on the end of the suction line will prevent this sediment from being drawn through to the pump.

Calcium chloride is not a very pleasant substance to get on the hands, and Mr. Anderson must expend much labor in preparing the brine as described. I have heard of some plants boiling a small quantity of calcium where it was desired to put the refrigerating apparatus into service without delay. I think that Mr. Anderson's system requires an excessive expenditure of time and labor. If the system above described is adopted it will save much hard work and possibly a freeze-up in case of accident.

R. A. CULTRA.

Cámbridge, Mass.

Long Springs on Discharge Valves

In the plant where I am employed there are two steam-driven ammonia compressors. The efficiency of the machines had been gradually falling off and, although the load due to expansion of business had been increasing, this condition could not be held as the cause. The double-pipe coolers were carefully overhauled and the internal pipes renewed as they were heavily coated with scale and rust and badly pitted. Naturally we looked for better results, but the machines still failed to "deliver the goods." About this time a new condition developed. Machine No. 1 would do about the same work as before, but machine No. 2 "laid down" altogether. At first it was surmised that the machine had been allowed to become unduly heated and had probably developed a scored or cut cylinder. To relieve any doubt on this point the cylinder was opened up and the valves reground, although they looked to be in good shape. Trials for clearance between the collar on the valve stem and the guide web on the valve cage showed no effect. A caliper showed the cylinder to be a very scant $^{1}/_{64}$ in. larger in the center than at the ends. The rings were in good shape and the clearance all right.

The machine was put together again and cut in on the system, but so far no betterment of conditions could be noticed. In fact, No. 2 machine was in all appearances the same as before the overhaul. Meanwhile No. 1 was steadily failing. This time the chief made the discovery that the needles on the suction or low-pressure gages seemed to be acting in a very sluggish manner. This started another investigation, and eventually the facts discovered were these: The springs on the head, or discharge valves were just long enough to allow the valve-box cover to be pulled up snug, leaving the spindle spring almost, but not quite, coil to coil, thus restricting the discharge-valve opening to a very limited area. When the springs had been shortened $^{3}/_{8}$ to $^{1}/_{2}$ in. and the machine cut in again, the way that the brine began to fall in temperature can only be appreciated by those who have had similar troubles. The same trouble was found in No. 1 machine and immediately remedied. Trial runs showed that the machines had been restored to normal operation. The above case indicates that it is a good plan to inspect the springs on all machines. It may save a lot of trouble.

H. J. BAKE.

Victoria, West, B. C., Can.

Practice and Science

At the meeting of the Cold Storage and Ice Association of Great Britain, the president, J. T. Milton, in his opening address referred to the statement of Sir Alfred Ewing that the steam engine in its early stages of development owed little or nothing to theory. In mechanical refrigeration the case is different. At every step of the process we find the influence of scientific ideas. No part of it has taken place, except under the direct guidance of physical science. The success in producing low temperature for commercial uses has been rendered possible only by the combination of scientific knowledge and the mechanical skill and experience of the practical engineer. It is the engineer with high scientific knowledge and great practical expeence to whom we must look for the further progress that will surely come. cannot be too well recognized that a theoretical knowledge of all the principles involved in refrigeration and in the preservation of foodstuffs dealt with by cold storage, must add largely to the usefulness and value of those whose business it is to give the unremitting attention required to successfully deal with the veryday duty of the refrigerating engineer.

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POWER

State Boiler Inspection

Nothing could show more forcibly the need for stringent laws governing the operation of steam boilers, than the many cases of criminal carelessness, negligence and ignorance, brought to light since the new boiler-inspection law in Ohio went into effect.

This law made it possible to inspect many boilers that were never inspected by anyone, competent or otherwise, and revealed conditions that were startling, to say the least.

It exposed many plants where boilers were in charge of either grossly ignorant men, who knew absolutely nothing about the explosive energy in boilers under pressure, or criminally careless men, who, though otherwise competent, were operating under conditions that they knew were violating all laws of safety.

If the public only knew in how many plants dangerous conditions are found and corrected by the inspectors there would be no peace for our legislators until all states had adequate boiler-inspection laws. Massachusetts and Ohio are the only states having such laws and enforcing them, although there are some cities in other states having their own.

Here is one example of what is being found by the inspectors:

A so called high-pressure heating plant carried from 40 to 60 pounds of steam pressure, used a reducing valve and returned the condensation with a Bundy trap. It contained an 80-horsepower boiler, with a three-inch lever safety valve. The owner of this plant operated the boiler and lived next door. To heat his own house he reduced the discharge of the safety valve from three inches to one inch and connected to the house-heating system. When he wanted steam in the house he held the safety valve open by putting a stick under the lever.

This man took pride in showing the inspector the advantages of using this method for getting just the right amount of steam to heat his house in any kind of weather.

Another example is one of criminal carelessness on the part of a competent engineer. This happened in a manufacturing plant having a boiler 60 inches in diameter of $\frac{5}{16}$-inch plate, with double lap seams, $\frac{3}{4}$-inch rivets on two-inch pitch, 67.5 per cent. joint efficiency, 25-foot grates, and a $1\frac{1}{2}$-inch pop safety valve set to blow at 100 pounds. The bursting pressure was 360 pounds, making the factor of safety 3.6. The boiler was twenty years old.

The engineer, when asked why he operated the boiler at that pressure and with a safety valve entirely too small, knowing the danger of operating under those conditions, advised the inspector that he had the best job in town and that, if he made a kick, he would get fired, so he preferred to take a chance and keep the job.

It is needless to say that at both of these plants the proper remedy was applied to safeguard the public from "the fools that are with us." What might have happened had there been no inspection can only be conjectured.

The Logical Unit of Power

Although the unit of the mechanical rate of doing work established by James Watt (the horsepower) is generally accepted as being the equivalent of 550 foot-pounds per second, regardless of locality, the actual work represented by one foot-pound really varies a trifle with the latitude and altitude; Watt's horsepower, therefore, is not a stable unit. For example, 550 foot-pounds at 45 degrees latitude and sea level represents the same actual work as 551¼ foot-pounds at the equator or 549¾ foot-pounds a mile above sea level at 60 degrees latitude; the difference being due to the variation of the acceleration due to gravity, or as it is commonly expressed, the attraction of gravity.

These slight discrepancies are not of great importance in ordinary power-plant work, but they are a nuisance when it comes to translating between mechanical and electrical units, because the watt

is the same at all latitudes and altitudes.

The United States Bureau of Standards has therefore adopted a fixed value of exactly 746 watts for the equivalent of one horsepower and put the mechanical equivalent on the sliding scale where it belongs. The mechanical equivalent of 746 watts at sea level and 30 degrees becomes 551.01 foot-pounds per second; at 45 degrees it is 550.28 and at 60 degrees 549.55 foot-pounds per second. At an altitude of 5000 feet the equivalent of a horsepower becomes 551.92 foot-pounds per second over the equator; 551.18 at 30 degrees latitude; 550.45 at 45 degrees, and 549.71 foot-pounds per second at 60 degrees.

The bureau advocates the kilowatt as the standard unit of power*; and its argument therefor is reasonable. Unfortunately, however, pounds and feet are tangible, physical units with which everybody is familiar, whereas the kilowatt seems rather theoretical or even imaginary to the nonscientific mind. One cannot see a volt or an ampere. In reality the kilowatt is more fundamental in character than the horsepower, because it is based on physical phenomena which have the same values at all points within man's reach. Scientifically, therefore, the kilowatt is the logical unit of power and practically it is just as easy of application as the horsepower. Moreover, its adoption would greatly simplify the conversion of mechanical into electrical terms and *vice versa*, which is of considerable importance in these days of heavy-duty electrical machinery and long-distance power transmission.

Pipe Lines in Power Plants

Pipe lines have been appropriately called the arteries of the plant. In large plants, especially those of office buildings, the pipe lines are very numerous.

Because serious leaks anywhere in line damage the building and material near them and also waste the medium which the line conveys, it is highly important that those responsible for the care of hte lines know where any particular line goes to, what it conveys and where the stop valves are located. Further, they should know where to quickly find the stop valve nearest the leak so as not to interrupt the supply to more rooms or apparatus than necessary.

In office-building plants, as in all others, the engineer is looked upon as a bureau of information of anything relative to the light, heat and power. The operating engineers usually have little to do with repairs to pipe lines throughout the buildings, this work being done by the "fitters." But when the latter fail to find a particular line the engineer is consulted—"he ought to know," is the universal conclusion. He should, therefore, take keen interest in the pipe lines, the changes made and those proposed.

Unfortunately, the engineers on the night watches are frequently handicapped in this respect. Changes are made in important lines during the day without their knowledge. This leads to serious mistakes and trouble, which would have been avoided if a written notice of the change had been posted where all might read it.

It is difficult, indeed, to get thoroughly acquainted with the pipe lines in the cellar or in one part of the cellar, but to acquire the ability to distinguish the various lines in the rooms or shafts of a many-storied building requires time and study and there is no short cut.

The chief engineer should insist that the engineers, oilers, pipe fitters, plumbers, etc., under him study the lines, and in addition he should select one "fitter," a man of good memory and judgment to specialize himself in this work. The man should be a steady worker and one likely to remain in the company's employ indefinitely. To assure this his wages should be commensurate with the value of his services. There should be no hesitancy in giving him such, as he must be occasionally called out of bed or kept on duty Sundays to direct emergency repairs. The responsibility for the care of, and the recording of alterations to lines should also be given this man.

When new work is being done, those in the plant should go over the ground with the steam fitter or plumber and make notes of the location of the valves they control. There is no better way of "learning the lines."

All of the foregoing should be done in addition to studying and keeping up-to-date the usual directories of the pipe lines. The latter should be posted where all may have access to the information which they contain.

*Circular No. 34, recently issued.

World-beating Discoveries

Every little while some genius proposes to revolutionize power production by applying an entirely new principle. Shoemakers render ashes combustible by sprinkling on them a magic "dope." Drygoods clerks solve the "problem" of perpetual motion, and book dealers increase the efficiency of a machine to over 100 per cent. by introducing a train of gears.

That engineers have not solved these problems is because they have become so steeped in the dogma of their profession that they are blind to new principles.

Now a company has been organized to produce liquid carbon dioxide from the products of combustion from lime-kilns, power-plant furnaces, etc. The gases of combustion are taken from the stack, cooled and washed and then separated in a centrifugal machine; the different gases separating because of their different specific gravities. The carbon dioxide is then liquified and sold primarily for use as a refrigerant.

This scheme can be extended to extracting free nitrogen and oxygen from the air. Without doubt also these "centrifugal separators" will be installed in every power plant, superseding the slow and costly Orsat apparatus.

An engineer who was consulted thought that gases when brought together would diffuse until intimately mixed and that their subsequent separation would be extremely difficult. The company regard this man as "bright enough but very bull headed." In coal mines carbon dioxide issuing from the roof drops to the floor, showing that gases separate because of the difference in their specific gravities. How the human race has managed to exist in the stratum of carbon dioxide that must envelope the earth (if the foregoing reasoning is correct), is a mystery.

The improvement which recent years have effected in the efficiency of the air compressor, is brought into sharp relief by the statement in a paper recently presented by George Barr to the Manchester (England) Association of Engineers, in which he mentions an old compressor which was replaced by a new one of four times the capacity, which it nevertheless took only the same amount of steam to drive.

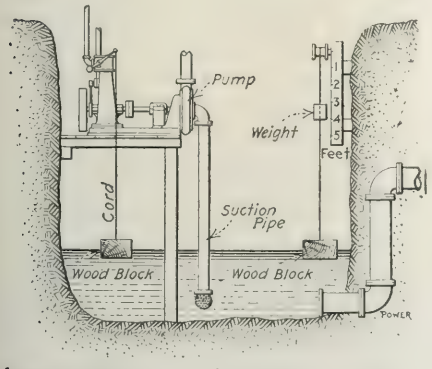
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Automatic Control of Engine Driven Pump

Recently I ran a small engine driving a centrifugal pump used to remove water from excavations in which concrete work was being done. The engine and pump were on a platform in a very small sump pit. As the pit was always wet and difficult of access and as I worked nights, having to start the pump frequently to prevent flooding the newly laid concrete was very troublesome.

To avoid this inconvenience I bolted



ARRANGEMENT FOR AUTOMATIC CONTROL OF ENGINE CONNECTED TO SUMP PUMP

a lever to the handwheel of the throttle valve, put a weight on one end of the lever and a cable and block of wood on the other end. The wooden block floated in the water as shown. As the water rose, the throttle opened and did not close until the proper amount had been pumped out. I also provided a telltale that consisted of an ordinary large spool, cord and float as shown at the right of the illustration.

J. H. KELLY.

New York City.

Troubles with Superheated Steam

The troubles experienced in high-pressure plants operating compound engines with superheated steam have generally been blamed on the grade of cylinder oil used, the character of the iron in the cylinders or the piston rings.

I have had charge of several Corliss engines, of the vertical type, carrying steam at 150 lb. pressure and 120 deg. superheat that showed as high as $\frac{1}{16}$-in. wear in the rings in 24 hr., and the walls and pistons looked black and badly scored. A cylinder oil of a high grade, 600 deg. flash point, was used, but no amount of it would help matters. Graphite in various forms was used with no signs of relief.

After considerable studying it was finally decided that the trouble might be due to superheated steam getting in behind the rings when the valves opened, thereby causing an excess pressure. A set of rings, built in three sections with coil springs behind them, was made and a groove $\frac{1}{4}$ in. wide and $\frac{1}{4}$ in. deep turned in the face. Four $\frac{1}{4}$-in. holes were drilled in the bottom of the groove, through the ring, in each one of these three sections so that they could not become steam-bound. A set of rings was run in one engine for ten days and the cylinder opened and examined thoroughly. The walls and piston had smoothed up practically as good as new, and the rings showed very little wear. All the wearing surfaces had a good polish and the old troubles had disappeared.

All the engines were then equipped with these rings.

F. J. RAVLIN.

Chicago, Ill.

Cleaning Inside and Neglecting Outside of Boilers

In a boiler plant the parts most likely to be neglected are the external surfaces of the drum and tubes and the baffle walls of the boiler. It is comparatively easy to clean the scale off the inside of the drum and bore it out of the tubes, but another matter to get into the firebox and back chamber and, with a piece of pipe with a hook bent on the end, rattle the soot and ashes from the outside of the tubes. According to Professor Abady, a coating of $\frac{1}{16}$ in. of soot on the outside of the tube will cause a loss of fuel of 15 per cent., $\frac{1}{8}$ in., a loss of 32 per cent., etc. The loss due to defective baffle walls can hardly be estimated, but it is very apparent that a defective baffle wall is expensive.

Last summer and again this summer the writer was called to inspect a battery of several water-tube boilers in an isolated plant in charge of an engineer supposed to be an expert. The inside of the drums and tubes were free from scale and could not have been in better condition, but on the outside of the tubes the soot and ashes were heaped up to a peak, and in some places were packed solid between the tubes. The tubes were blown two or three times a week, but the blower was not very effective. My practice when cleaning a boiler is to thoroughly clean the external surfaces first and stop up any air leaks that may appear, then go after the inside.

Economizers are fitted up with some form of an automatic scraper which keeps the tubes free from soot, and it seems to me that this idea could be applied with good results to the tubes of water-tube boilers, although boilers with vertical baffles would require two or three sets or scrapers. It is hard and slow to replace the baffle brick in some water-tube boilers, even with the recently invented tools, but I have found from experience that it pays in the end to be as particular in keeping the outside clean as the inside. Also that strict attention should be given to air leaks in the brickwork and around the doors. Lack of complete combustion has much to do with the amount of soot deposited on the tubes. When combustion is complete much less soot will collect on the tubes and a saving is made both in an increase of heat units due to more complete combustion, and also to more water evaporated per pound of combustible due to the absence of the insulating coating of soot.

J. C. HAWKINS.

Hyattsville, Md.

Unloading Coal on Bonus

In some plants the chief engineer finds it rather expensive to unload coal from cars by hand and transport it by wheelbarrows to the coal pile or pit. To those having charge of such work the following should prove interesting:

The laborers unloading coal from cars usually work slowly and the cost per ton of unloading is often very great. The cost per ton generally can be reduced by unloading the coal on a bonus system by which the men are amply compensated for the extra effort exerted in their work.

In some places the car is simply opened at the bottom and the coal dropped into the coal pit, in others the coal has to be shoveled into wheelbarrows and wheeled to the coal pit. It was in one of the latter places, where the coal had to be wheeled 20 ft. from the car to the pit, that the bonus system was tried very successfully.

Before the men were put to work on a bonus it took three laborers from a day and a half to two days to unload a 50-

ton car. The cost to the company was about $10, or 20c. per ton, the men receiving 18c. per hr.

Under the bonus system two men were given the job of unloading a 50-ton car. They shoveled out the coal and wheeled it 20 ft. to the coal pit in a total time of 19 hr. or about 23 min. per ton. The cost of unloading the coal was reduced to 14c. per ton and the men averaged about 30c. per hr.

This speaks well for the bonus system when laborers can increase their pay from 18c. to 30c. per hour.

W. L. Myles.

Irvington, N. J.

Tank Collapses Under 14-in. Vacuum

As nearly as my recollections serve me, there was 14 in. vacuum on that tank the last time I passed it, the same being there by the order of Boss Kemist, who said the tank was guaranteed for 200 lb.

We were trying to force oil through the filters that were fast becoming plugged with waste, dirt, etc., and as they emptied into the oil tank, we were assisting the flow by applying the vacuum machine to the tank. At 10 in. vacuum the chief walked away in disgust, and at something above 14 in. there was a roar and the tank, one of the "common garden variety" type, closed up like a paper bag. The boss collapsed nearly as badly as the tank, because he was so persistent in his demand for a greater vacuum.

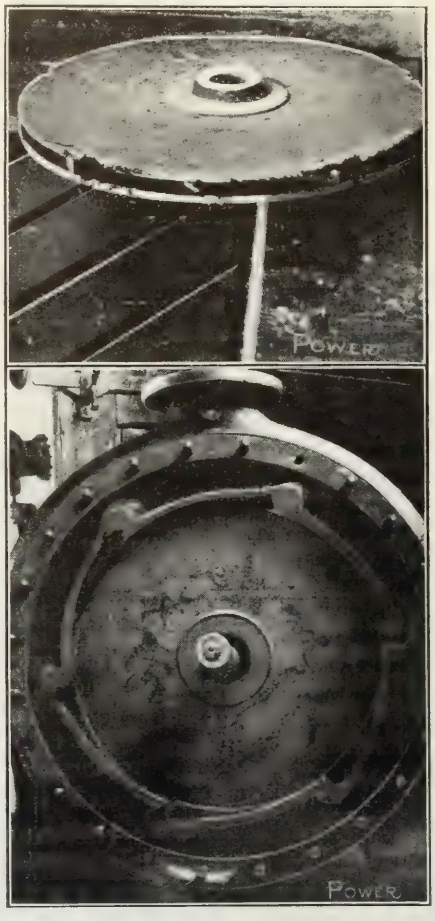
Richard Phoenix.

New York City.

Corrosion of Pump

The illustrations show the corrosion of a 3-in. two-stage turbine pump, which we believe is due to electrolysis. This is one of two pumps which are taking water from a receiving cistern and delivering it into a storage tank of 3,000,000-gal. capacity, the high level in which is 164 ft. above the base of the pumps.

The water in the cistern is delivered from several deep wells by air lifts and double-acting deep-well pumps. Both of these turbine pumps are motor-driven, the motors being supplied with direct current at 240 volts, and they either run together, discharging into the same line, or separately, according to the demand. The motor driving the pump these photographs show, has for some time past been taking considerably more current than the motor driving the other pump, and just before the pump was shut down it was discovered that there was considerable current leaking where the leads entered the motor frame, which explained in a measure the much higher current consumption of this pump. The result is but vaguely indicated in the photographs. Both the runner and the entrance to the cross-over water connections into the second stage of the pump were badly pitted. I am unable to explain just why this action was confined to the part of the pump mentioned, but perhaps others may.



Showing Corrosion of Parts of Centrifugal Pump

Both of these pumps at times handle water which is very salty and which also contains some sulphur, but the other pump shows no such corrosion as does the one illustrated.

L. B. Lent.

Detroit, Mich.

Think It Over

The following actual happening illustrates better than anything else the value of a thorough system of power-plant records.

In a New England city of some 300,000 inhabitants, in the heart of the business district and within two blocks of each other are two department stores. Until recently each ran its own heat, light and power plant.

The engineer of one had been aware for some time that efforts were being made and persuasive arguments used by the central-station representatives to allow them to furnish light and power. As a matter of fact both engineers knew their companies were being approached.

The first engineer was called into the office and told of the central-station solicitor's offer and was shown the figures. "If you will wait a moment or two Mr. Blank, I will show you some figures that I think will interest you more than these." The engineer came back with the plant records on which were figured everything entering into the cost of production of a kilowatt-hour. Initial cost of installation, floor space, interest and depreciation, coal, oil, waste, lamp renewals, wages, and even removal of ashes. The central-station man nearly collapsed when the total showed a kilowatt-hour for a price far below his offer.

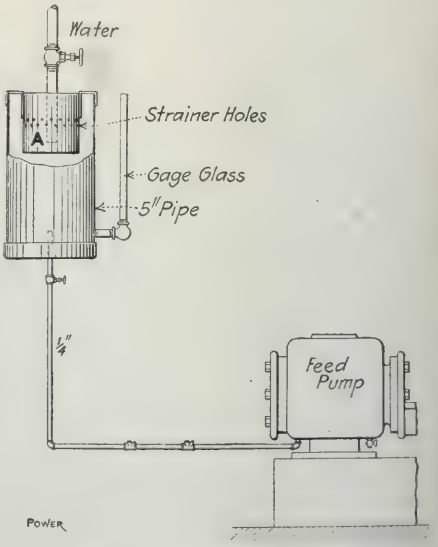
Neither he nor any of his kind have ever been seen around that plant since. I do not know what the engineer of the other plant has been doing, but at last report he was looking for a job, because that same solicitor got his firm's signature to a contract on July 1. The reason is obvious.

R. H. Lyman.

Moosup, Conn.

Boiler Compound Feeder

The illustration shows an inexpensive apparatus for continuous feeding of boiler compound in connection with a pump drawing water from an open heater or other source under less than 5 lb. pressure.



Boiler Compound Feeder

The operation is as follows: Place the compound in powder or brick form in the bucket *A*, and turn on the water slowly (hot or cold). This will dissovle the compound and the liquid will overflow through small holes near the top of the bucket. When the tank is full open the valve in the ¼-in. pipe enough to empty the contents in 12 or 24 hr. as desired.

The ¼-in. pipe connects through two

checks into the drain opening on the water cylinder of the feed pump. Each time the plunger makes a suction stroke it draws in a small amount of compound and this amount increases with the speed of the pump.

If the water comes to the pump under too much pressure tighten the spring on the suction valve of that chamber, which will make the pump draw harder on the feeder. It is necessary to use two check valves *B* on account of grit getting under the valves, then one will usually be clean and hold back the water on the discharge stroke which would otherwise flood the feeder. Dry compound can be added at any time without stopping the feed and the usual mixing pail, funnel and spilling of compound are avoided.

A. KARREMAN, JR.

Grand Rapids, Mich.

Card Index System

Many ideas have been set forth in POWER relative to filing articles so they can be found readily when wishing to refer to them. While many of these ideas were good, none of them appealed to me as exactly what I wanted, consequently I tried, and am using one of my own. I drew up a sample card size 3x5 in. and had 1000 cards printed like those shown, for which I paid $1.60.

All matter pertaining to one subject,

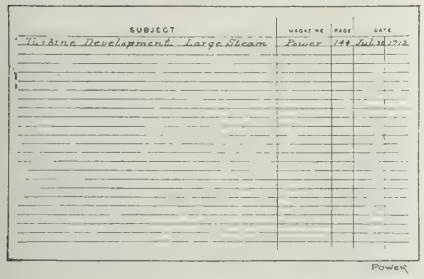


FIG. 1. SUBJECT CARD

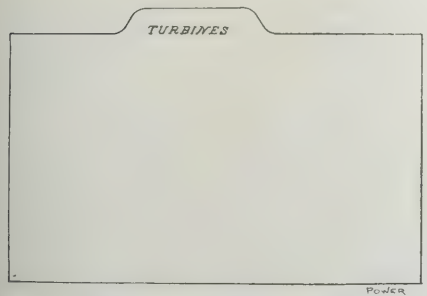


FIG. 2. INDEXED CARD

as turbines, is put on one card, as in Fig. 1. These cards are not indexed, but another set, Fig. 2, is used, which is blank except the index. These cards are all kept in a box made purposely for them, the index cards being placed directly in front of all cards on that subject. It is therefore easy to find the articles on any particular subject by referring to the date and page of the magazine mentioned on the card, no matter how long it has been printed.

I do not cut the magazines, but separate the reading matter from the advertising sections, and bind the former into books of convenient size, so that the page numbers are continuous. I have found this index a great time saver.

L. E. THOMPSON.

Medford, Mass.

A Knock to Locate

I have charge of a 60-hp. suction gas producer plant, the engine of which is belt-connected to a 30-kw., 2300-volt generator. The engine has a knock which I am not able to locate and would like to hear from interested readers in regard to the cause and remedy. The engine has a throttling governor that controls the mixture. The engine knocks while coming up to speed, but runs very quietly until a very light load is thrown on, then the engine knocks in one cylinder only, but this knock is in step with every other revolution of the engine. As the load increases, the knock becomes irregular up to about one-half load, after which the engine runs very quietly.

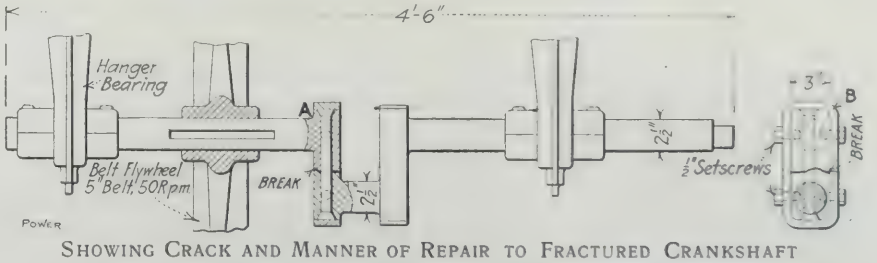
The crankshaft is in alignment and all bearings are taken up to a good working fit. The spark has about 15 deg. advance. The fuel used in the producer is coke.

P. F. ROBNETT.

Chouteau, Mont.

Air Pump Crankshaft Repair

The shaft in question drove an air pump through a long connecting-rod and being single acting, the badly arranged bearings and belt flywheel caused much unnecessary stressing. The break showed a bad flaw in the forging and the fracture was prematurely opened as the flywheel, which was originally keyed to the shaft with a flat key, worked up against the crank web at *A*. The first intimation of trouble occurred when the flywheel reached the floor 10 ft. below, accompanied by the pieces of crankshaft and the shaft hangers.



SHOWING CRACK AND MANNER OF REPAIR TO FRACTURED CRANKSHAFT

A repair was at once effected by clamping the two parts of the shaft together firmly and after setting them between lathe centers, getting them as true as possible, drilling a 1-in. diameter hole down through the web, a tool steel pin was driven in lightly and headed over cold. The sharp corners of the crank web were well rounded off and an oblong ring *B* of 2x⅝-in. iron was shrunk on. Setscrews were used to prevent the ring shifting.

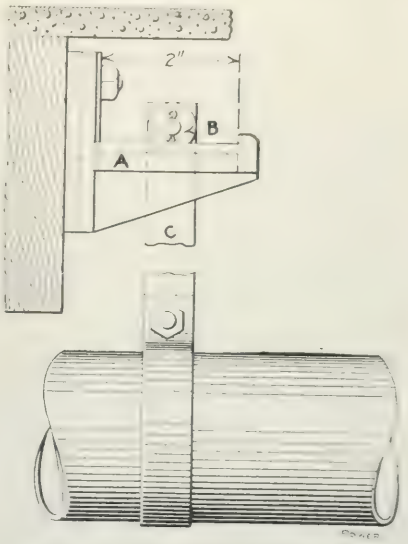
The hangers meanwhile having been set up, the shaft was replaced, and, although it was slightly out of true, it ran day and night for a month until a new shaft arrived.

J. McA. HOWDEN.

Melbourne, Vic., Australia.

Expansion Pipe Hanger Bracket

While erecting an exhaust heating system in a modern concrete building, I had occasion to connect an 8-in. pipe line



EXPANSION PIPE HANGER BRACKET

about 175 ft. long running under beams at right angles to the pipe.

To suspend the line, taking care of expansion and contraction and avoiding bulky brackets or spools, I designed the expansion hanger bracket shown. The bracket *A* is a casting weighing 4½ lb., cast without cores; the rollers *B* have a travel of 2 in., and the suspension bar *C* also has a ½-in. lateral movement in a slot. This size bracket is suitable for 6-in. to 12-in. pipe with the hangers spaced according to the size of the pipe. The design of the bracket can be modified for pipe lines to run parallel to the beams.

GEORGE G. LENNIG.

Philadelphia, Penn.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Smoke Prevention with Automatic Steam Jets

In the Sept. 3 issue in an article under the above heading, J. A. Sweitzer referred to my article published in the Feb. 20 issue and took exception to my statement: "The rate of combustion depends on draft." He followed with this remark: "This is a very orthodox statement; if coal were a pure carbon it would be strictly true and with anthracite coal the rate of combustion does depend almost entirely upon the draft, but the statement emphatically does not apply to soft coal and especially where it is hand fired."

First of all the rate of combustion is the amount of coal burned per unit area of grate in a unit of time. In practice a unit of time is taken as one hour and a unit of area as one square foot of grate. The amount of coal which a fireman can shovel into a furnace is not limited by his physical ability or the coal on hand but by the amount he can burn economically. To burn coal economically requires a fixed amount of oxygen which has to be obtained from the atmospheric air and this oxygen must be uniformly distributed throughout the fuel bed, and the operation requires force. The force required to deliver air to the fuel bed and to distribute it is derived from the draft.

Draft is a pressure difference between the outside and inside of the stack; this pressure difference is created by heated air entering into the stack and as heated air has a greater volume than an equal weight of cold air, it weighs less for the same volume. This difference in weight causes the flow of fresh air from outside into the furnace and thence into the stack. The pressure difference, not the vacuum, does the work; therefore the rate at which air is delivered and distributed through the fuel bed and thence carried into and out of the stack is proportional to the pressure difference. The greater the pressure difference the more air will be delivered and better distributed and, therefore, more coal can be consumed. Oxidation of fuel becomes easy if the desired impinge of air against the fuel can be had and intimate mixture of volatiles and air can be obtained by arches or arches and steam jets combined. Every fireman knows this and regulates his draft accordingly, and no amount of sophism will convince him that he should close the damper if he wanted more steam or to burn more coal. From this it can be seen that the rate of combustion in any case, regardless of what the fuel may be, depends entirely upon the draft.

Further in the same article Mr. Sweitzer has quoted from the excellent work of the Bureau of Mines, Bulletin No. 26, describing how distillation takes place and what the products of distillation are, he concludes as follows: "From this discussion it should be clear that it is not the rate of combustion which depends on draft as Mr. Misostow states, but on the rate of driving the combustibles out of the fuel, an altogether different matter."

To burn coal it must be broken into, first, gas and fixed carbon. Carbon enters into combustion in very small particles, gases are broken into their elements—hydrogen and carbon. Hydrogen first liberates the carbon and enters into combustion with the oxygen and liberated carbon enters into combustion in its turn and forms CO or CO_2. Distillation being the preparatory stage in the combustion of coal, naturally bears a direct ratio to the rate of combustion, but it cannot be substituted for the rate of combustion, which means consumption of all fuel. The rate of distillation in coke and gas making, for instance, is entirely different from the rate of combustion and it has a very indirect relation to it.

In the same bulletin, quoted by Mr. Sweitzer on page 367, under the heading "Relation of pressure drop to rate of combustion and evaporation," concludes: "Thus one may logically conclude that the rate of combustion and the capacity of the boiler will vary with the difference of pressure between the furnace and the uptake." The difference of pressure referred to is draft.

Steam Jets

The difference between automatic steam jets and common steam jets is that one is shut off automatically and the second is hand operated. Mechanism which starts steam jets is sometimes actuated by fire-doors and closing is accomplished by some arrangement of weight and dashpot; in fact the most ingenous one I ever saw was operated by a damper regulator. The advantage claimed for automatic steam jets is that they save steam, that is, they use steam only when required while the common steam jets are turned on all the time; however, so far as their respective work is concerned one is no better than the other when properly installed.

Automatic steam jets are generally patented and the patentee naturally expects to be compensated, whereas common steam jets can be installed by any engineer. The drawback of the automatic steam jets is that they become autocratic and refuse to work, are more complicated, and as the fireman does not throw on the same amount of coal at every charge one adjustment is not satisfactory for all fire conditions. However, they may be all right, but most of the time they give trouble.

H. Misostow.

Chicago, Ill.

[The last sentence of Mr. Misostow's letter would lead one to believe that all automatic steam jets for draft production were subject to frequent failures of proper operation. This is not wholly true, for with proper care, the attendant need experience little trouble with the jets.

The chief trouble with automatic steam jets is the plugging of the steam nozzles with the alkaline impurities carried by the steam from the boiler. This trouble is easily overcome by pushing a wire into the nozzle while the steam is blowing through; the pressure will then blow out the particles as they are broken away by the wire.

When the nozzles are placed in the soot-pit and the latter is allowed to fill with soot or the sliding door of the pit is closed, the air supply is so reduced as to make the jets ineffective. There is no excuse for such troubles, however, as reasonable attention to duty would obviate them.—Editor.]

Comment on Pressure Chart

Although it is a little late, I cannot refrain from commenting on the steam chart shown by Mr. Seymour in the Aug. 13 issue.

The steam pressure varies from 120 to 130 lb., which is too much. If he cannot make the damper regulator work better than it does, I advise him to discontinue its use altogether and regulate the damper by hand. Steam charts that do not record a pressure variation of more than 2½ lb. in 24 hr. are common. Mr. Seymour should be able to obtain such charts. Again, the chart shows that the damper is closed about half the time.

I wonder what a CO_2 recorder would show if applied to his boilers. He should cut out some of the boilers or reduce the grate surface in them until his firemen will have to work harder to keep up steam if he wants to obtain economical boiler operation. He says the fires are cleaned three times in a 24-hr. run. I believe that. It merely shows that he has too many boilers in service; otherwise the fires would have to be cleaned oftener to keep up steam. I imagine that the fires must get quite thick after an eight-hour run, and that the amount of air that would get through the very thick fuel bed would hardly be enough to support good combustion.

If he will clean his fires at more frequent intervals, installing grates that will admit of easy cleaning, he can bank the 250-hp. boiler and have it standing by for emergencies and save coal. I mentioned that his dampers, and probably also the fan, were shut off half the time. The rest of the time more heat was being generated than the boilers could absorb and the gases were going to the stack at too high a temperature.

W. T. MEINZER.

Brooklyn, N. Y.

Unsafe Boilers

The article under this heading in a recent issue reminded me of some unsafe boilers I saw recently. We had loaned a pulsometer pump to a firm that was sinking its quarry deeper. After they had used the pump for a day or two, they called up the office and wanted a man sent up to start it, as they were unable to do so.

I went the next day and found the pump connected to a small vertical boiler of about 40 hp. No provision had been made for a throttle valve near the pump and no extra valves were in stock, but I found an old one on a water tank and connected it into the pipe. The steam line was in accord with the rest of the equipment, being old, rusty and leaking at every joint.

The fireman informed me that the boiler could carry 125 lb. and as the safety valve was blowing off slightly I started the pump, which, after a few strokes stopped.

Thinking something was wrong with the valves I opened the pump, but found everything in good condition. I was then informed that the steam had fallen to 70 lb., so telling the fireman to get a good fire going, I walked over to a small locomotive-type boiler that furnished steam for two drills just to see what it looked like. It was very rusty and as I approached I saw it was leaking at the girth seam where the shell joined the firebox.

There was no steam gage or water glass and I asked the fireman who came over to put on fresh coal: "What pressure have you got on this boiler?" "Me don't know," he answered, "Bime bye she pop," and sure enough it did as soon as a drill was stopped.

Opening the fire-door disclosed a bad bag in the crown-sheet. The joints around the stay-bolts as well as the flues were leaking and in very poor condition generally. How they ever kept a good fire in the furnace was a mystery to me, even though the blower was working to its full capacity.

The lower try-cock showed water when opened but the upper ones were tightly plugged. This aroused my curiosity as to the condition of the rest of the equipment and I walked back to the boiler that supplied steam for the pump. The fireman had a fire that was white hot and that filled the firebox half way up to the flue sheet, but the gage registered only 75 lb. The gage also showed that it was about ready for the scrap heap. The dial glass was gone and also half the pointer. The gage looked as though it had been damaged by fire.

After the fireman hit the gage a couple of cracks with a stick the half pointer jumped up to 105 lb. There was no water glass on this boiler and the try-cocks looked as though they had been but very little used. Trying the lower one and holding a shovel under it, showed no water and as I backed away I asked him where the water was.

"Oh, it's there," said he as he turned on the injector. Water appeared in about three minutes and he considered he had brought the water level up very quickly. The firebox of this boiler was bagged and blistered all around the sides and leaked in two places from what appeared to be cracks in the sheet. Near the door the sheet was red hot on the outside, showing it to be covered with scale up to the door at least.

These boilers were never washed and seldom blown out, it being too much trouble to fill them again, I was informed. After trying the pump once more and reducing the pressure so much that the pump would not run, I went back to the office and reported that the pump was too large for the boilers and started for home, thankful to get away from the place alive and whole.

These boilers are within 100 ft. of possibly twenty-five men, who are working in the quarry, and within a stone's throw of many dwelling houses. Since my visit a boiler maker told me that he had been sent to put a patch on the vertical boiler, but had told the owner that a new firebox was needed as it was impossible to repair the old one. Whether this has been done since I do not know, but these boilers are in operation at this writing and the boiler maker said the pop safety valve on the locomotive-type boiler had become entirely inoperative while he was there, but that they still continued to fire the boiler as usual.

C. L. MALLERY.

Myerstown, Penn.

Phosphorus for Flue Gas Analysis

In a recent issue, L. A. Farrell asks about the use of this substance for the absorption of oxygen. It has the advantage that it will absorb oxygen (theoretically) almost indefinitely and with considerable rapidity. It suffers from the serious disadvantage that there is a small quantity of ozone formed at the same time which oxidizes any carbon monoxide to carbon dioxide, rendering it soluble; also unless kept in the dark it becomes covered with a leathery skin which effectually stops further absorption. Its most serious disadvantage is that it is readily inflammable. If its container be cracked, allowing the water to run out, it will spontaneously take fire, causing the destruction of its container and running down onto the wooden case, setting it on fire. Finally, it is not easily prepared in the small sticks necessary for absorption, and must be handled rapidly and with care to prevent its spontaneous ignition. All these considerations have led me to prefer the slower and more unsatisfactory copper solution.

A. H. GILL.

Boston, Mass.

Regarding the use of phosphorus as a reagent in flue-gas analysis, there is a point that has not been clearly brought out in the discussion. It is that the action of the phosphorus soon becomes slow and then ceases altogether. This characteristic makes it necessary to test the action frequently by introducing a sample of air into the phosphorus pipette when the prompt appearance of the characteristic white fumes shows the material in good condition.

Should the action be unsatisfactory, it is not necessary to throw the remaining phosphorus away or even remove it from the absorption pipette, but only to wash it with water, which can be done by changing the water once or twice.

The solid phosphorus pentoxide P_2O_5 fumes formed in the reaction in the pipette form metaphosphoric acid HPO_3 as they are absorbed by the water, which acid acts to protect the phosphorus from the oxygen.

In handling phosphorus it should always be remembered that it spontaneously ignites in air and that its burns are exceedingly deep and painful. For these reasons it must be stored under water and must never be touched with the fingers in handling.

G. W. MUNRO.

LaFayette, Ind.

Who Is Responsible?

The question in the above heading which appeared on the title page of POWER for Oct. 22 is worthy of considerable thought. In the same issue appear accounts of two boiler-tube accidents in which one man was killed and four others were badly injured. In the previous issue an account also appeared of a bursting boiler-tube accident in which one man was scalded to death and another so badly scalded and burned by flying cinders from the furnace that he was not expected to live, and the engineer died of heart failure due to the same accident.

If Germany or any other country has a law to guard against such injuries and loss of life I say by all means let us enact a similar law. Let us adopt all the essential parts of the German law and add enough to it to meet the conditions under which we are working. I know many readers of POWER are in a position to easily convince their employers of the great protection from injury to have some sort of a locking arrangement to prevent the furnace door from flying open when a tube lets go. Let readers relate their experiences along this line so that we may all work for the same cause and protection. When it has been proved that inward swinging fire-doors and other boiler adjuncts are a success, then I think it will be easy to convince the law makers of the good to be derived from the enactment of a law to make their use compulsory.

A. LAMARINE.

New Bedford, Mass.

Air Compressor Efficiencies

The very excellent article on "Air Compressor Efficiencies," by E. M. Ivens, in POWER of Oct. 15, affords an opportunity for discussing the subject and for submitting certain definitions, differing somewhat from those given, as used in the steam and gas engineering department of the University of Wisconsin.

The definition of mechanical efficiency as the ratio of the air horsepower plus the jacket horsepower divided by the indicated horsepower of the steam cylinder cannot be accepted as correct, for this would indicate little advantage in water jacketing other than that of securing lower temperatures, when it is well known that by approaching the isothermal line in compression the actual horsepower required is reduced. The heat given to the jacket represents energy furnished by the moving piston, but this energy is included in the area of the air diagram. In the case of isothermal compression the heat energy given to the jacket is exactly equal to the area of the diagram if clearance is disregarded, while in actual compression the area of the air diagram is equal to the heat given to the jacket plus the additional internal energy given to the air during compression. This point of view is apparent if the fundamental energy laws dealing with compressed air are considered. The energy stored up in compressed air is a direct function of the temperature, and with isothermal compression there is no more energy in the compressed air than in the free air, both being of the same temperature. The only difference is that in the case of the compressed air the energy is available because a drop of pressure—and, therefore, of temperature is possible, while with free air this is not true.

In compressing air isothermally energy is expended in merely changing the energy in free air from an unavailable to an available form, and the work of the air-compressor card shows the energy that is expended to make the energy in air available. It has no reference whatever to the energy in the air.

The definitions given by Mr. Ivens of volumetric efficiencies both observed or apparent and real or true are excellent, but the definition of compression efficiency as illustrated in Fig. 1 is open to criticism in that *df* does not represent the amount of air compressed.

We have used two separate efficiencies in considering the matter of compression, viz., cylinder efficiency and efficiency of compression. Cylinder efficiency is defined as the ratio of the work done in a complete cycle to compress isothermally a volume of air at atmospheric pressure equal to the intake piston displacement, divided by the actual work done in the air cylinder. This is the same as Mr. Ivens' definition of compression efficiency.

Efficiency of compression is defined as the product of the cylinder efficiency and the true volumetric efficiency, or it is the work done in a complete cycle in compressing isothermally without clearance a given volume of free air, divided by the work actually expended in compressing the same volume of free air.

The definition of overall efficiency given by Mr. Ivens is probably not stated as intended by the author for the use of "boiler horsepower" is clearly not justifiable in this ratio. The term boiler horsepower means the evaporation of 34.5 lb. of water from and at 212 deg. F. per hr. and cannot be used in such an efficiency expression.

We use the term "net efficiency" to include all losses of air compression and define it for a steam-driven air compressor, as the ratio of the available energy in the compressed air to the available energy in the steam used. This means the internal energy available in the compressed air at room temperature or the energy available by its adiabatic expansion to atmospheric pressure divided by the energy available in the steam supplied if expanded adiabatically in a Rankine cycle from the inlet to exhaust pressure.

The reason for this term may be clearer from a statement of the changes of available energy from steam to compressed air. The thermal efficiency of a steam cylinder represents the ratio of the heat energy turned to work to the heat energy supplied in the steam. The standard steam-engine cycle used for comparison is the Rankine cycle, in which adiabatic expansion is assumed from inlet to exhaust pressures. The ratio of the actual thermal efficiency of a steam engine to the efficiency of a Rankine cycle between the same limits gives the so called efficiency ratio of the steam end.

If the heat energy available in the steam by adiabatic expansion from inlet to exhaust pressure be multiplied by the efficiency ratio the product will represent the energy of the moving steam piston.

If the energy of the moving steam piston be multiplied by the mechanical efficiency the product will represent the energy of the moving air piston, and if this energy be multiplied by the efficiency of compression the product will represent the energy required to compress isothermally the air actually compressed.

If the compressed air could be expanded isothermally without further addition of energy, the items mentioned would include all the losses of energy. As isothermal expansion without addition of heat is impossible there is a further loss to be considered due to the fact that the energy stored up in the compressed air is measured by the work done by adiabatic expansion to atmospheric pressure.

It will be seen from this that the term "net efficiency" includes losses in the steam cylinder due to failure to realize maximum possible efficiency in utilizing the steam, losses due to friction in the compressor, losses due to faulty valve action, poor design of ports, inefficient cooling, and the inevitable loss of energy due to the fact that the expansion of compressed air follows a different path from the ideal compression line. These losses are all summed up in the term "net efficiency" or the ratio of the available energy in the compressed air to the available energy in the steam used.

H. J. THORKELSON.

Madison, Wis.

The Society for Electrical Development has been incorporated to establish cooperative relations among the different electrical interests in the United States, Canada and Mexico with a view of increasing the use by the public of electrical current and to promote the welfare of individuals identified with all branches of the industry. The principal office will be in New York City.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Relative Resistances of Wires

If the resistance of a piece of round copper wire 0.001 in. in diameter and 1 ft. long is 10.8 ohms, what would be the resistance of 1500 ft. of 1-in. square copper wire of the same kind of copper and at the same temperature?

H. M. H.

The cross-sectional area of 0.001 in. diameter wire is

$$0.001 \times 0.001 \times 0.7854 = 0.0000007854 \text{ } sq.in.$$

and the cross-sectional area of 1 in. square wire would be 1 sq.in., and as the resistances of the wires would be inversely as their cross-sectional areas and directly as their lengths, the resistance of the 1-in. square wire 1500 ft. long would be

$$\frac{0.0000007854}{1} \times 10.8 \times 1500 = 0.0127 \text{ } ohm.$$

Diameter of Steam Pipe

What should be the diameter of a steam pipe for furnishing 200 lb. of steam per min. at a pressure of 105 lb. abs., allowing a pipe velocity of 5500 ft. per min.?

E. R. R.

The specific volume, i.e., the number of cubic feet per pound, of dry saturated steam at the pressure of 105 lb. abs. is 4.230 cu.ft. and the volume of 200 lb. would be

$$4.230 \times 200 = 846 \text{ } cu.ft.$$

To convey this quantity per minute would therefore require the steam pipe to have a cross-sectional area of $846 \div 5500 = 0.1538$ sq.ft. or $0.1538 \times 144 = 22.1472$ sq.in., and the diameter required for the steam pipe would be

$$\sqrt{\frac{22.1472}{0.7854}} = 5.3 \text{ } in. \text{ } diameter.$$

Equivalent Evaporation

A boiler supplied with feed water at 80 deg. F. evaporated 1900 lb. of water per hour into dry saturated steam at 130 lb. per sq.in. abs. pressure; what is the equivalent evaporation from and at 212 deg. F.?

R. E.

Each pound of water fed at 80 deg. F. contains nominally $80 - 32 = 48$ heat units above 32 deg. F., and according to Marks & Davis' steam tables, a pound of water raised from 32 deg. F. to dry saturated steam at 130 lb. absolute pressure would require 1191 heat units. Each pound of water raised from feed water at 80 deg. F. and evaporated under the conditions stated therefore received

$$1191 - 48 = 1143 \text{ } B.t.u.$$

The same steam tables give the latent heat of evaporation of water at 212 deg F. and atmospheric pressure as 970.4 B.t.u. Hence for the conditions stated the factor of evaporation would be

$$\frac{1143}{970.4} = 1.1778$$

and the equivalent evaporation is

$$1900 \times 1.1778 = 2237.82 \text{ } lb.$$

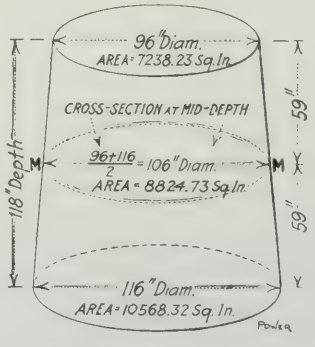
of water evaporated from and at 212 deg. F.

Computing Capacity of Tank

What is the method of computing the capacity, in gallons, of a round wooden tank having inside dimensions of 1196 in. diameter at the top, 116 in. diameter at the bottom, and depth of 118 in.?

W. B.

Referring to the figure the content in cubic inches is found by multiplying by



DIMENSIONS OF TANK

the mean area of cross-section in square inches by the height in inches. The form of the tank being a frustrum of a cone, the mean cross-sectional area is found by adding together the area at the top, the area of the bottom and four times the area of the cross-section at mid depth *MM*, and dividing their sum by six. Employing the values shown in the figure, the mean cross-sectional area would be

$$\frac{7238.23 + 10,568.32 + (4 \times 8824.73)}{6} = 8850.91$$

and the volume would be

$$8850.91 \times 118 = 1,044,407.38 \text{ } cu.in.$$

One gallon being 231 cu.in., the capacity in gallons would be

$$1,044,407.38 \div 231 = 4521.24 \text{ } gal.$$

Stop Valve for Boiler Inspection

What is the purpose of an inspection stop valve and bleeder on a boiler and how are they placed and used?

A. B. S.

When two or more boilers are connected into the same header, to guard against leakage of steam into a boiler through a single main stop valve during internal examination, the boiler connection to the header is provided with an extra stop valve, and a good-sized bleeder and stop valve are placed between the two main stop valves. For purposes of inspection both of the main stop valves are closed and the bleeder valve is left wide open so as to discharge any steam that may leak through the main stop valve which is next to the header.

Thickness of Boiler Shell

What should be the thickness of the shell of a boiler 60 in. in diameter to carry 70 lb. pressure, the efficiency of the longitudinal seams being 68 per cent. and working strength of the shell plates 9000 lb. per sq.in. of section?

E. R. R.

The shell of a boiler 60 in. in diameter carrying 70 lb. pressure per sq.in. would have to resist a strain of

$$\frac{60 \times 70}{2} = 2100 \text{ } lb.$$

on each inch of length of the shell. The working strength of the solid plate being 9000 lb. per sq.in., the thickness of solid plate required to resist 2100 lb. per in. of length would be

$$\frac{2100}{9000} = 0.2333 \text{ } in.$$

But as the efficiency of the joint is to be 68 per cent., the actual thickness of plate for sufficient strength at the joint would have to be

$$0.2333 \div 0.68 = 0.343 \text{ } in.$$

or

$$0.343 \times 32 = \frac{10.976}{32} \text{ } in.$$

i.e., practically $\frac{11}{32}$ in. thick.

Engineers' Study Course—V

Denominate Numbers

In this lesson we deal with numbers that denominate particular quantities. Thus 6 *feet* is a denominate number because it names, designates or denominates feet. But 7 standing alone might mean seven of anything, as bolts, cylinders, inches, feet, tons, yards, pounds, etc. If standing alone, the number is an *abstract number* because it does not designate any unit or denomination.

Denominate numbers as given in books on arithmetic deal with units of measure, which are quantities established by law as standards to follow in buying, selling and computing quantities of things.

A simple denominate number designates units of only one denomination, as 2 ft.; 6 lb.; 4 tons, etc. A compound denominate number designates units of two or more denominations, as 8 tons, 20 lb., 4 oz., or 2 weeks, 4 days, 3 hours, 4 min. In working examples in denominate numbers all we wish to do is to reduce all the different quantities, as the 8 tons, 20 lb., 4 oz. to a common denomination of tons, pounds or ounces.

There are six kinds of measures:

Weight; Extension; Capacity; Time; Angles; Money.

Money will not be considered as every reader is familiar with American money values. The chief thing to observe in the reduction of denominate numbers is this:

Multiply to reduce to lower denominations and divide to reduce to higher denominations.

Example—Reduce 8 tons, 20 lb., 4 oz. to ounces. One ton = 2000 lb. One pound = 16 oz.

2000 lb.	16020 lb.	256320
8	16 oz.	4
16000 lb.	96120	256324 oz. ans.
20	16020	
16020 lb.	256320 oz.	

Example—Reduce 284 in. to higher denominations: 12 in. = 1 ft.; 3 ft. = 1 yd.

12 in.) 284 in.
3 ft.) 23 ft. + 8 in.
7 yd. − 2 ft. + 8 in. ans.

In the last example 12 in. went into 284 in. 23 times and left a remainder of 8; the divisor 12 was inches and so the remainder must be inches. Eight inches do not equal 1 ft., so it must be added as inches. The 3 ft., or 1 yd., went into 23 ft. 7 times and left a remainder of 2 ft., which is not a whole yard, so the final answer must be 7 yd., 2 ft., 8 in.

To reduce denominate numbers to higher or lower terms we must know the relative values of all the different units; that is, how many ounces there are in a pound, how many pounds in a ton, the number of inches in a foot, how many feet there are in a yard, and also the number of square inches in a square foot or square yard and how many gills in a pint, pints in quarts, quarts in gallons, gallons in a barrel, etc.

We will not go into the addition, subtraction, multiplication and division of compound denominate numbers directly as such, for the average man, will invariably reduce the different denominations to a simple denominate quantity before performing such examples, even though he knows how to do it the other way.

For instance, instead of adding direct 2 cwt. + 150 lb. + 8 oz., most men will say, "2 cwt. = 200; 200 + 150 = 350 lb.; 8 oz. = ½ lb. So the answer is 350½ lb."

The accompanying tables have been compiled chiefly from Hering's book of "Conversion Tables," much of which has been accepted by the National Bureau of Standards and checked by L. A. Fischer, Asst. Phys., National Bureau of Standards. Only those units of measures most used by engineers are given. Their equivalents in units of the metric system are given to further enable the student to convert units of the English system into their equivalents of the metric system. Should the student care to go further into the study of the metric sys-

LENGTH OR EXTENSION

1 mil = 0.001 in.
1 millimeter = 39.370 mils
1 millimeter = 0.039370 in.
1 millimeter = 0.001 in.
1 centimeter [cm.] = 0.3937 in.
1 centimeter = 0.0328 ft.
1 centimeter = 0.01 m.
1 inch [in.] = 1000 mils
1 inch = 25.400 mm.
1 inch = 2.540 cm.
1 inch = 0.0833 ft. or $\frac{1}{12}$
1 inch = 0.2777 yd. or $\frac{1}{36}$
1 inch = 0.0254 m. aprx. $\frac{1}{40}$
1 foot [ft.] (U.S.) = 304.801 mm.
1 foot = 30.480 cm.
1 foot = 12 in.
1 foot = 0.333 yd. or $\frac{1}{3}$
1 foot = 0.3048 m. aprx. $\frac{3}{10}$
1 foot = 0.000304 km.
1 foot = 0.000189 mile
1 yard [yd.](U.S.) = 91.440 cm.
1 yard = 36 in.
1 yard = 3 ft.
1 yard = 0.914 m.
1 yard = 0.000914 km.
1 yard = 0.000568 mile
5.5 yd. = 1 rod.
40 rods [rd.] = 1 furlong [fur.]
8 furlongs = 1 mile
1 mile [ml.] = 5280 ft.
1 mile = 1760 yd.
1 mile = 1609.35 m.
1 mile = 1.609 km.
1 mile = 0.868 nautical mile.
1 knot (U.S.) = 6080.20 ft.
1 knot = 1853.25 m.
1 knot = 1.853 km.
1 knot = 1.151 m.

SURFACES

1 circular mil. [cm.] = 0.785 sq.mil.
1 circular mil. = 0.000001 circ. in.
1 square mil. [M²] = 1.273 circ. mils.
1 square mil. = 0.000001 sq.in.
1 circular inch = 1,000,000 circ. mils.
1 circular inch = 785,398 sq.mils.
1 circular inch = 0.785 sq.in.
1 circular inch = 0.00694 circ.ft.
1 circular inch = 0.00545 sq.ft.
1 sq.inch = 1,273,240 circ. mils.
1 sq.inch = 1,000,000 sq.mils.
1 sq.inch = 1.273 circ. in.
1 sq.inch = 0.008 circ.ft.
1 sq.inch = 0.0069 sq.ft.
1 sq.foot = 183.346 circ. in.
1 sq.foot = 144 in.
1 sq.foot = 1.273 circ.ft.
1 sq.foot = 0.111 sq.yd.
1 sq.yd. = 1,296 sq.in.
1 sq.yd. = 9 ft.
1 sq.yd. = 0.836 sq.m.
1 sq.yd. = 0.000206 acre.
1 sq. meter = 10.763 sq.ft.
1 sq. kilometer [km.²] = 10,763,867 sq.ft.
1 sq.mile = 27,878,400 sq.ft.
1 sq. mile = 640 acres

VOLUME OR CAPACITY

1 cu. centimeter = 0.061 cu.in.
1 cu.centimeter = 0.001 liter.
1 cu.inch = 16.387 cu.cm.
1 cu.inch = 0.034 pt. (U.S.) liquid.
1 cu.inch = 0.017 qt. (U.S.) liquid.
1 cu.inch = 0.004 gal. (U.S.) liquid.
1 cu.inch = 0.000578 cu.ft.
1 pint = 473.179 cu.cm.
1 pint = 28.875 cu.in.
1 pint = 16 fluid ounces.
1 pint = 4 gills (U.S.)
1 pint = 0.5 qt. (U.S.) liquid.
1 pint = 0.125 gal. (U.S.) liquid.
1 pint = 0.016 cu.ft.
1 quart [qt.] (liquid; U.S.)
1 quart = 946.35 cu.cm.
1 quart = 57.75 cu.in.
1 quart = 8 gills
1 quart = 0.25 gal.
1 quart = 0.033 cu.ft.
1 quart = 0.946 liters.
1 gallon = 3785.43 cu.cm.
1 gallon = 231 cu.in.
1 gallon = 32 gills.
1 gallon = 4 qt.
1 gallon = 3.785 liters
1 gallon = 0.133 cu.ft.
1 cu.foot = 28,317 cu.cm.
1 cu.foot = 1728 cu.in.
1 cu.foot = 59.844 pt.
1 cu.foot = 29.922 qt.
1 cu.foot = 28.317 liters.
1 cu.foot = 7.480 gal.
1 cu.meter = 61,023.4 cu.in.
1 cu.meter = 1000 liter
1 cu.meter = 35.314 cu.ft.

Note: The volume of a barrel is not given as it varies with different manufacturers marketing products by the barrel. The barrel is not a fixed unit and no value has ever been adopted by Congress.

WEIGHT

1 milligram [mg.] = 0.1 cg.
1 milligram = 0.001 gram
1 centigram [cg.] = 10 mg.
1 centigram = 0.01 gram
1 grain [gr.] = 0.064 gr.
1 grain = 0.002 ounces (oz.)
1 gram [g.] = 15.432 gr
1 gram = 0.035 oz. (av.)
1 gram = 0.032 oz. (troy)
1 gram = 0.001 kilogram (kg.)
1 ounce [oz.] = 437.5 gr.
1 ounce = 28.349 g.
1 ounce = 16 drams
1 ounce = 0.062 lb. (av.)
1 ounce = 0.028 kg.
1 pound [lb.] = 7000 gr.
1 pound = 453.59 g.
1 pound = 256. drams (av.)
1 pound = 16. oz.
1 pound = 0.453 kg.
100 pounds = 1 hundredweight (cwt.)
1 kilogram or kilo [kg.] = 15,432.356 gr.
1 kilogram or kilo = 1000 g.
1 kilogram or kilo = 35.274 oz. (av.)
1 kilogram or kilo = 2.204 lb.
1 ton [tn.] = 2000 lb. (short ton)
1 ton = 20 cwt.
1 ton = 907.185 kg.
1 ton = 0.907 metric ton.
1 long ton = 2240 lb.
1 long ton = 1016.05 kg.
1 long ton = 1.12 short tons
1 long ton = 1.16 metric tons

tem and the conversion of units he should consult some book on the subject.

These tables are good for reference for ordinary work and we suggest that the student file them for this purpose.

MEASURE OF ANGLES OR ARCS

60 seconds ["] = 1 minute [']
60 minutes = 1 degree [°]
90 degrees = 1 right angle or quadrant [L]
360 degrees = 1 circle [cir.]
1 circle = 360° = 21,600′ = 1,296,000″.

THERMOMETER SCALES

1 centigrade degree = $\frac{9}{5}$ or 1.8 deg. Fahrenheit
1 centigrade degree = $\frac{4}{5}$ or 0.8 deg. Reaumur
1 Fahrenheit degree = $\frac{5}{9}$ deg. Centigrade
1 Fahrenheit degree = $\frac{4}{9}$ deg. Reaumur
1 Reaumur degree = $\frac{5}{4}$ or 1.25 deg. Centigrade
1 Reaumur degree = $\frac{9}{4}$ or 2.25 deg. Fahrenheit.

Paper Measure

1 quire = 24 or 25 sheets
1 ream = 20 quires = 480 sheets
1 ream generally = 500 sheets

Miscellaneous Measure

1 dozen (doz.) = 12
1 gross = 12 doz. = 144
1 great gross = 12 gross
1 score = 20

POWER

1 horsepower [hp.]	= 33,000 ft.-lb. per min.
1 horsepower	= 550 ft.-lb. per sec.
1 horsepower	= 746 watts.
1 horsepower	= 0.746 kw.
1 horsepower	= 42.41 B.t.u. per min.
1 horsepower	= 1.014 metric hp.
1 horsepower	= 375 mile-pounds per hr.
1 horsepower	= 10.7 large calories per min.
1 kilowatt [kw.]	= 44,256.7 ft.-lb. per min.
1 kilowatt	= 737.7 ft.-lb. per sec.
1 kilowatt	= 1000 watts
1 kilowatt	= 1.34 hp.
1 kilowatt	= 56.9 B.t.u. per min.
1 kilowatt	= 1.36 metric hp.
1 kilowatt	= 14.34 large calories

PRESSURES

1 foot of water column	= 304.8 kg. per sq.m.
1 foot of water column	= 62.4 lb. per sq.ft.
1 foot of water column	= 0.433 lb. per sq.in.
1 foot of water column	= 0.029 atmosphere [atm.]
1 inch of mercury column [in.hg.]	= 345.4 kg. per sq.m.
1 inch of mercury column	= 70.7 lb. per sq.ft.
1 inch of mercury column	= 0.491 lb. per sq.in.
1 inch of mercury column	= 1.132 ft. of water
1 inch of mercury column	= 0.033 atm.
1 inch of mercury column	= 0.345 meters of water

To convert a reading in *Centigrade Degrees* into corresponding Fahrenheit degrees, multiply by $\frac{9}{5}$ and add 32 (because the zero on the Fahrenheit scale is 32 deg. F. below zero on the Centigrade scale). Multiply Centigrade degrees by $\frac{4}{5}$ to convert the reading into Réaumur degrees. To convert *Fahrenheit Degrees* into Centigrade degrees, subtract 32 and multiply by $\frac{5}{9}$. To convert Fahrenheit into Réaumur degrees, subtract 32 and multiply by $\frac{4}{9}$. To convert *Réaumur Degrees* into Fahrenheit degrees, multiply by $\frac{9}{4}$ and add 32.

EXAMPLES FOR PRACTICE

(1) Reduce 20 tons, 105 lb., 6 oz. to oz.

(2) Reduce 1 mile, 227 ft., 10 in. to inches.

(3) How many inches in 20,000 mils?

(4) How many centimeters in $\frac{1}{12}$ ft.?

(5) How many centimeters in 25 ft.?

(6) A cable is 10,560 yd. long. (a) How long is it in miles? (b) How long is it in kilometers?

(7) If a vessel travels 20 knots, how far does it travel in land miles?

(8) A square foot contains 144 sq.in. How many circular inches does it contain?

(9) The floor area of an engine room is 1240 sq.ft. How large is it in square meters?

(10) (a) A tank contains 8000 gal. of water. How many liters does it contain? (b) What is the volume of the tank in cubic inches?

(11) One cubic foot equals how many cubic inches?

(12) 372,155.5 kg. of coal are received. What is the equivalent in short tons?

(13) Neglecting losses, how many British thermal units must be supplied to a 500-hp. engine to run it one hour?

(14) The water pressure due to the height of a factory tank is 80 lb. (a) How high is the tank? (b) How many feet high would a mercury column need to be to balance this pressure?

ANSWERS TO LAST WEEK'S QUESTIONS

(1) 0.1; 0.09; 0.095; 3.1915; 123.09

(2) 1010.24 + 2.222 + 19.005 + 0.0026 + 51.2 + 3.1416 = 1085.8112

(3) 1029.56 — 350.05 lb. = 679.51 lb.

(4) 4.59 × 42 = 192.78 gal. per min. are pumped; 192.78 gal. × 219.5 min. = 42,315.21 gal. total gallons pumped.

(5) 15,125 lb. ÷ 27.5 lb. = 550 = horsepower of engine; 15,125 lb. ÷ 9.5 lb. = 1592.3 lb. of coal used; 1592.3 ÷ 550 = 2.89 + lb. of coal per hp. per hr.

Economy in a Small Plant*

At the plant of the Buffalo Coated Paper Co. some interesting data were obtained showing what can be accomplished in a small plant by a good engineer employing modern methods. The plant has two 125-hp. return-tubular boilers which have seen over ten years of service. Less than a year ago a new 250-hp. engine was installed to replace an engine which the load had outgrown.

For several years the average daily fuel consumption for the boilers was from 6½ to 7 tons of coal, the yearly contract specifying the requirements at about 2300 tons. The plant was operated 60 hr. per week, 11 hr. daily through the week and 5 hr. on Saturday. A day and night fireman were employed, and the former, with the chief engineer, were on duty 11 hr., namely, from 7 a.m. to 6 p.m., the night fireman serving 13 hr. Besides furnishing steam for power, it was necessary to maintain a constant temperature throughout the night in the drying room where the paper was coated, and besides steam was employed to heat the air passing through the blowers to the drying room during the day. The engineer's report stated, however, that the plant was in actual operation 78 hr. each week. This, of course, included the time of getting up steam pressure in the morning and very often operating the main engine to run certain parts of the machinery before 7 a.m. or after 6 p.m.

About the first of January last the engineer started the new year by removing 1000 lb. of scale from the two boilers. This in itself showed a marked reduction in fuel consumption, but with this the engineer was not satisfied, and in April began to analyze the gas. From this time on a daily log was kept, giving the quantity of water metered, the daily average CO_2 for each boiler, etc.

From Sept. 3 to 18 inclusive the plant was in operation 142 hr. The coal consumed during that time was 36¾ tons. During the same period the night fireman was on duty 194 hr. and consumed 50 lb. less than 9 tons of coal. The average hourly evaporation for each day was 3326 to 4278 lb., the daily feed-water temperature varying from 179 to 200 deg. F. and the average daily steam pressure from 76 to 82 lb. gage. The average evaporation from and at 212 deg. was 10.36 lb. per lb. of coal burned, and the average coal consumption per horsepower-hour was 3.34 lb. The lowest average daily CO_2 reading in either boiler was 10 per cent., and the highest 12.4 per cent.

The first daily average of CO_2 readings which were taken some two weeks after the instruments were installed showed about 6½ per cent. The readings steadily improved, however, and on July 15 a daily average of 10 per cent. was obtained, and at the present writing this figure is considered a poor showing.

Scale has been removed from the tubes of No. 1 boiler twice and from No. 2 boiler once since January, and the engineer is positive that the plant can be operated with a daily coal consumption of not much in excess of 2½ tons, if the steam requirements are not increased. The concrete results of this work were shown when the coal salesman recently came to obtain the contract for next year's supply for the requirements were reduced to 1000 tons per annum. During the year, numerous trials were made with different grades of coal, on the thickness of the fuel bed, the draft and other data tending to produce the best results.

From this work it is very evident that the installation of CO_2 recorders and other accessories are of no particular value in themselves without the coöperation of an intelligent and progressive engineer and fireman. With men capable of this coöperation the results show what can be accomplished even in an ordinary plant if uptodate methods are followed. The annual saving of this company in power-plant operation is now nearly $3000, which is equal to the wages paid to the engineer and the two firemen.

*From data supplied by the William B. Pierce Co., of Buffalo, N. Y.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

An anonymous well wisher sends us a three-page pome for this column and ventures to hope that we may be able to publish it "By arangeing the Punctiation Capitals etc." Our literary repair shop is already crowded, and, anyway, we hold fast to our rule never to do a job of any kind for unknowns.

Our valued contributor, W. J. Au, is hereby installed in our hall of fame. There are still several choice niches, with all the modern improvements, unoccupied. Who's next?

A French scientist has discovered that a hypodermic injection of formic acid will cure the most obstinate case of laziness. Don't believe it! Recently, we saw a power-plant employee who couldn't move fast enough to get in a moving picture. He could defy a heavy charge of dynamite and get away with it.

The general manager glanced casually at the graduate. "I have no position to offer you," he said. "You misunderstand," responded the graduate. "What I want is a job." Two minutes later he was at work.

"Illuminating engineers are making a fight against glare." He must be a sassy old thing who would glare at a poor engineer, although we have a couple of strong glares tucked away for the "engineer" who hung, about a mile over our desk, a faint ray which looks like that 16-cp. lamp in the Goddess of Liberty's fist.

"Koo'-chook!" Sounds like a cold in the head or the name of a captured Turkish town. "Caoutchouc," of course, means india rubber; trying to pronounce it without the aid of a diagram means much profanity. We bounce it. "Rubber" suits us.

You can jolly well believe no operating engineer would short-circuit our classic language as did a coal-mine machine runner in the following bulletin:

no. 4 Masheen awl Shot tu hel. Chain brok. fix the dam thing. it brok twict Last nite. amatoor smoks like heltoo. power bad. lots of short circles. this is the masheen with self compeller truck. weer outa oil tu.

One more jest, a jab or josh—
We've *got* to fill this space, b'gosh!
Sometimes, we can write a volyum—
Ha! This last line fills the colyum!

Members and Guests of the Ohio Society of Mechanical, Electrical and Steam Engineers at Akron

Ohio Society of Mechanical, Electrical and Steam Engineers Meet

The fall meeting of the Ohio Society of Mechanical, Electrical and Steam Engineers was held at Akron, Ohio, on Nov. 21, 22 and 23, with headquarters at the Hotel Portage.

Following an informal reception to the guests and visitors at the Portage on Thursday afternoon the first regular session was held in the evening at the auditorium of the South High School and was opened by an address of welcome by Mayor Rockwell, of Akron; E. M. Adams, president of the society, responded.

The paper of the evening was on "A System of Gas Engine Governing," by George S. Cooper, who discussed in detail the limitations of the hit-and-miss, the throttling, and the variable mixture methods of governing, and pointed out the advantages of combining the two latter methods.

Professor Magruder also made a few remarks in which he pleaded for the establishment of a state engineering-experiment station at the Ohio State University. He pointed out that Illinois, Kansas, Missouri and Iowa now have such experimental stations and Ohio frequently is compelled to submit engineering problems to the University of Illinois. He further emphasized the importance of such a station from an economic point of view.

On Friday evening two very interesting papers were read: "The Lentz System Applied to Steam Engines," by Siegfried Rosenzweig and "The Manufacture of Tin Plates," by E. W. Patten. The former referred to the great steam economies being attained in Germany by the application of poppet valves and high superheat to steam engines. (An abstract of this paper will appear in a later issue.)

Akron, because of its growing importance as a manufacturing center proved an admirable convention city, affording opportunities for many interesting inspection trips. These included visits to the plants of the B. F. Goodrich Rubber Co., the largest of its kind in the world, the Babcock & Wilcox Co., at Barberton, Ohio, makers of the Stirling boiler, the Quaker Oats Co. and a trip through the new power house of the Northern Ohio Traction Co.

The social program for the ladies included a theater party and automobile trips to several points of interest, including a trip to O. C. Barber's farm at Barberton.

The members and guests were tendered a complimentary dinner at the Hotel Portage on Friday evening.

Springfield, Ohio, was selected as the place for the next meeting.

The supply men were well represented

and a number of interesting exhibits were displayed. Among the exhibitors were V. D. Anderson Co., of Cleveland, Bird-Archer Co., of Pittsburgh, the Carey Co., of Akron, C. E. Squires Co., the Crane Co., of Chicago, the Crown Manufacturing Co., of Cleveland, Clement Restein Co., of Philadelphia, the Dearborn Drug & Chemical Works, the Dodge Manufacturing Co., the Flanner Boiler Co., of Toledo, Hoovens, Owens, Rentschler Co., the Homestead Valve Co., Jenkins Bros., the Lagonda Manufacturing Co., the Lunkenheimer Co., W. M. Pattison Supply Co., the William Powell Co. the Schaeffer & Budenberg Manufacturing Co., Tomlinson Steam Specialty Co. and the Union Rubber Co.

NEW PUBLICATION

HOW TO READ A DRAWING. By Vincent C. Getty. J. B. Lippincott Co., Philadelphia. Cloth; 63 pages, 6x9 in.; 61 illustrations; 10 chapters. Price, $1.

Both the drawings and text of this book bear the marks of fearless, wholesome and practical treatment of the subject of mechanical drawing without introducing any unnecessary technicalities or burdening the reader with any fads or fancies of the subject that are of no moment to the practical engineer or mechanic.

Drawing is a language by which we are frequently enabled to convey thoughts and ideas that would be impossible for us to express in words and such wonderful progress has been made in the art and science of representation that many savants who are posted in the world's work of today are inclined to predict that we are rapidly returning to a language of signs which, while more elaborate, is to be a repetition of the hieroglyphics of the ancients. The author may have had this idea before him in selecting the title "How to Read a Drawing," for he not only teaches the reader by easy stages how to read a drawing, but also makes it evident how to make a good mechanical drawing.

Good practical examples of different kinds of drawings are given which are both interesting and instructive, and by perusing the pages of this work any operating engineer should become enabled not only to read drawings but to well represent his own ideas in mechanical drawings, albeit they may lack in the technique of the finished draftsman.

N. A. S. E. Lecture on Safety Devices

Talking before Newark (N. J.) No. 3 Association, N. A. S. E., on safety devices in factories, Monday evening, Nov. 25, Col. Lewis Bryant, commissioner of labor for New Jersey, thanked the engineers for the assistance they have given his department. The engineer, the commissioner said, was the connecting link between the department of labor and the factory owner and manager.

Col. Bryant was pleased when the management of a factory selected its engineer as factory fire chief as the engineer was usually a good adviser to a management reluctant to install the protective apparatus recommended by the department.

A national museum at Washington was much needed for the purpose of exhibiting and testing safety devices and collecting more detailed statistics relative to the number and kind of accidents of which workers are the victims. The commissioner also pointed out how valuable were the recommendations of the National Board of Fire Underwriters. If he wished to know how good a certain apparatus was for fire prevention or protection he could quickly obtain full particulars from the Underwriters. On the other hand, if information was needed on the value of a safety device for a machine, he had to "dig up the information for himself" and then was not sure he had the best device.

In the discussion on the use of the nonreturn stop valve on boilers, the commissioner said his department had not recommended their use. The valve was then described and he promised to investigate its merits.

SOCIETY NOTES

The *Quarterly* of the American Institute of Architects will shortly be changed to a monthly, to be known as the *Journal of the American Institute of Architects*. J. Horace McFarland, Harrisburg, Penn., will be its publisher, D. Knickerbacker Boyd its editor and R. M. Hooker, New York City, its business manager.

The nominating committee of the American Society of Heating and Ventilating Engineers has selected the following names for officers for the next year. They will be elected by a mail ballot, the result to be announced at the annual meeting, which will probably take place Jan. 21, 22 and 23, 1913: President, John F. Hale, Camden, N. J.; vice-presidents, Edmund F. Capron, Chicago; Albert B. Franklin, Boston; treasurer, James A. Donnelly, New York; managers, Frank T. Chapman, New York; Ralph Collamore, Detroit; D. D. Kimball, New York; W. W. Macon, New York; J. M. Stannard, Chicago; Theodore Weinshank, Indianapolis.

A number of Chicago members of the American Society of Mechanical Engineers held a meeting at the Engineers' Club in that city on Nov. 21 to discuss the formation of a local section. P. M. Chamberlain acted as chairman. Representatives of the Western Society of Engineers, the American Institute of Electrical Engineers and the American Institute of Mining Engineers were present, and told of the benefits derived from local sections in other cities.

Dean Goss, of the University of Illinois, made an address on the benefits to be derived from a local section. The sentiment of the meeting was unanimous in favor of the formation of a local section and the holding of regular meetings, and the matter of arranging the details was left to a committee which will draw up a tentative plan to be submitted at a later meeting.

PERSONAL

Thomas A. Edison has been elected president of the Edison Manufacturing companies, succeeding Frank L. Dyer, resigned.

William B. Dickson, first vice-president of the United States Steel Corporation, has been elected president of the International Steam Pump Co. The presidency has been vacant since the death of Benjamin Guggenheim.

Harry Dix, for two years in charge of the heating and ventilating department of the Douglas Milligan Co., Ltd., has joined the engineering staff of the Canadian Domestic Engineering Co., Ltd. Mr. Dix is a Boston man who settled permanently in Canada several years ago.

Harry W. Jarrow has accepted the position of general sales manager for the Diamond Power Specialty Co., Detroit, Mich. Besides a broad selling experience, Mr. Jarrow brings to the technical side of his new work a mechanical engineer training gained both at the University of Pennsylvania and at the Armour Institute of Technology, of Chicago.

The following appointments were made recently by the Magnolia Metal Co.: J. W. Wright, formerly sales manager of the Griscom-Spencer Co., is special representative on the Pacific Coast, with headquarters at San Francisco. William H. K. Gamble, formerly Southern representative of Charles A. Schieren & Co., special representative in the Southwestern territory, with headquarters at Dallas, Tex.

Capt. Robert W. Hunt, of Chicago, has been awarded the John Fritz medal for this year. Capt. Hunt was one of "John Fritz's boys." When about 20 years old he began work in the iron industry at Pottsville, Penn. Later he studied chemistry and became the chemist of the Cambria Co. He was at various times the superintendent of steel works in Wyandotte, Mich., Johnstown, Penn., and Troy, N. Y. In 1888 he founded the firm of R. W. Hunt & Co., consulting engineers, inspectors, etc. He served in the Civil War from 1861 to 1865, rising from private to captain.

Moments with the Ad. Editor

In a certain street in Brooklyn lives a man whom the children call "Crazy Jimmie."

He is a perfectly harmless old fellow, always good natured, always ready to stop and chat with man or woman and to play with a child or pet a dog.

He earns his living by gathering stray dimes, quarters and half-dollars from housewives by doing odd jobs for them, washing down stoop fronts, carrying up coal, splitting wood, washing windows and similar tasks.

Nothing crazy about that, you see. What has gained him the name, however, is the way he goes around the street singing and laughing so that people sitting in their easy chairs smile at each other and say, "There goes poor Crazy Jimmy."

The other day several of us stopped him in the street and talked for a few minutes. He spoke very intelligently in a simple, everyday sort of way.

"Well, Jimmie," said one of the party, "You're all right except for one thing. Why do you shout and sing in the streets so?"

"That's how I get my living," answered Jimmy.

"But I thought you did odd jobs around houses and work like that."

"I do. But if I didn't sing and shout in the street the ladies would forget all about me. Now when they hear me they come out and say, 'Hey, Jimmie, I've got a job for you.' "

So after all poor Jimmie's craziness is nothing more than knowing enough to advertise his services.

* * * * * *

The great force and power that is modern advertising must have originally started in some humble and obscure way similar to that in which Jimmie offers his services.

Perhaps the oldest form of advertising of which we know is the town-crier who, very like Jimmie, used to wander through the streets and lanes of old-time towns crying out the great happenings of the day and also of the wares and goods of enterprising local merchants.

In fact, there is one town-crier still plying his quaint trade in the ancient village of Provincetown, up on the end of Cape Cod, in Massachusetts.

How inadequate that method seems in these days of wide reading, in universal interest in advertised goods—

When one popular weekly carries its message of reliable clothes, household utensils, food and other things to more than two million people—

When "Power," this paper of yours, every week gives over 30,000 of you the opportunity of reading about reliable machines and devices that will help you in your work, help to make your plant more efficient, help you in many cases to benefit yourself financially and in your surroundings.

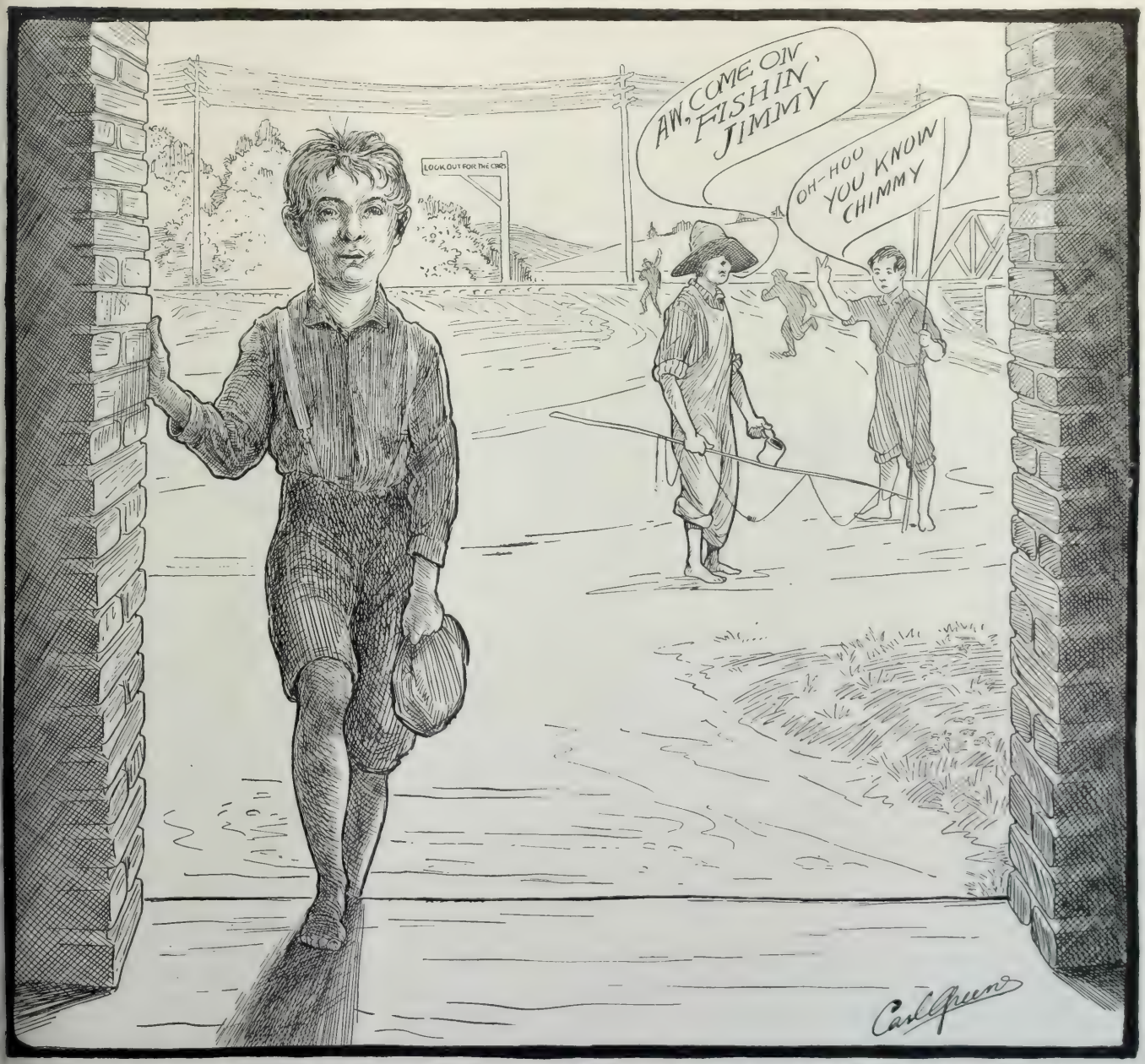
In the old days of the town-crier there would have been a very limited chance for you to hear of and examine and, in time, use these products.

In these latter days of the printing press you should certainly be glad of and use the opportunity—*continuously*. Get the habit of reading the advertising pages of "Power."

POWER

Vol. 36 NEW YORK, DECEMBER 10, 1912 No. 24

THE BEST RAW MATERIAL



"SAY, MISTER, CAN I COME IN?"

Tests of Wood, Paper and Steel Pulleys

By H. A. Woodworth

Account of tests made by E. D. Biggs and the author at Purdue University in 1909 to determine the breaking strength of pulleys of different kinds with a view to disclosing their principal points of weakness.

The object of the following tests was to determine the relative bursting speeds of different types of pulleys and, if possible, whether the weak points in each form of construction correspond with previous experiments.

The condition of stress existing in the rim of a flywheel was first discussed by J. B. Stanwood (*Trans.*, A. S. M. E., 1893). He showed theoretically that the bending due to centrifugal force might very materially reduce the bursting speed. This was later proved experimentally by Prof. C. H. Benjamin (*Trans.*, A. S. M. E., 1899-1912), who tested model flywheels to destruction by revolving them in a bomb-proof case at varying speeds until they gave way. Benjamin's tests proved that joints between arms consisting of a pair of lugs connected by bolts were a great source of weakness. Further to prove these theories, and test the most improved designs of pulleys a series of tests of 24-in. wheels of cast iron, steel, wood and paper were arranged for with E. D. Biggs and the writer in charge. The tests were conducted at Purdue University in the spring of 1909.

Fig. 1 shows the testing plant which consisted of a cylindrical pit in which was placed a 3-in. vertical shaft driven by a vertical motor. The shaft was supported in the center of the pit by cast-iron bearings bolted to two horizontal channel irons, the ends of which were imbedded in the concrete that formed the walls of the pit.

The pit was 7 ft. deep and 8½ ft. in diameter. The shaft was belt-driven from a 10-hp. variable-speed motor which could be operated from 800 to 2400 r.p.m. Sand was piled around the inside of this pit to prevent flying pieces from coming in contact with the concrete wall when failure occurred. By this means the original fracture was obtained.

The starting box, or controller for varying the speed of the motor, was placed in a safe position some distance from the pit. The tachometer was placed on a heavy wooden block, about 4 ft. from the pit. This block was imbedded in the ground, with its top on a level with the top of the pit.

To determine the speed ratio of the main shaft to that of the tachometer a series of simultaneous readings were taken of the main shaft and tachometer respectively, and from these data a curve was plotted with the tachometer readings as ordinates and the main shaft revolutions per minute as abscissas. This calibration curve was determined for every three or four pulleys tested, and proved to be a straight line through the origin upon which the results of all subsequent tests fell.

After balancing the pulley, photographing, weighing and running the calibration curve, preparations were made to test the first wheel.

This was placed on the shaft in the testing pit, and the pit was covered with planks to prevent any fragments from escaping. The wheel, which will be referred to hereafter as test wheel No. 1, was a two-arm solid-rim built-up type wooden flywheel, shown in Fig. 2 and weighed 29.37 lb. It was strengthened by six ½-in. bolts, four of which were placed at the hub and furnished means of clamping the flywheel to the shaft, the other two being placed one at each end through the continuous arm.

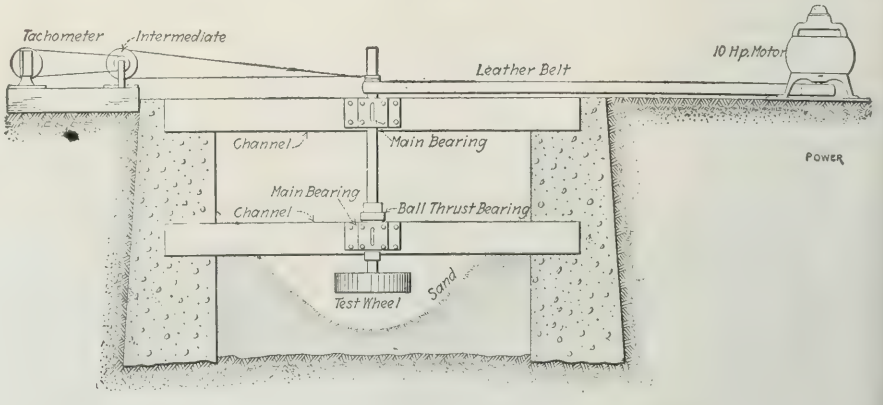


FIG. 1. PIT FOR TESTING PULLEYS TO DESTRUCTION

The test started at a very low speed and as each new speed was obtained the operator stepped the motor to the next speed, using great caution to gradually accelerate the pulley. When the pointer indicated a speed of 2720 r.p.m., there was a noise like the report of a muffled powder explosion as the pulley let go. A fine débris of splinters, bolts and nails, together with large pieces of the rim and flywheel arms was found lying all over the bottom of the pit showing the original disintegration fractures due to the excessive speed.

After the explosion the débris was gathered together and assembled as nearly as could be, and a photograph reproduced in Fig. 2 taken of it. The duration of this test was about 12 min. The peripheral speed at the time of fracture was 285 ft. per sec. The failure came very suddenly. The fracture was probably caused by one bolt in each end of the continuous arm giving out due to the centrifugal force. Tabulated data and results of the tests are shown in the accompanying table.

No. 2 was a duplicate of No. 1 and burst at a speed of 2550 r.p.m. or a peripheral speed of 267 ft. per sec. The whole rim was shattered, but the arms remained intact.

No. 3 was a split-rim built-up wooden pulley, the split being made at the ends of the arms, as shown in Fig. 3. This wheel was a duplicate of the first two wheels tested, except that the former two had solid rims. It burst at a speed of 2210 r.p.m. or a peripheral speed of 231.8 ft. per sec. The whole rim was shattered but the arms remained as shown in Fig. 3. A close inspection reveals the uniformity in the strength of the split-rim wheel where the joint is at the end of the arms.

Wheels Nos. 4 to 10 inclusive were duplicates of No. 3. In all these wheels the fracture was due to weakened rims caused by the holes made in them at the factory, in which were placed balancing weights. These weights together with the weakened section caused the failure. In some cases the bolts holding the arms to the two sections of the rim were considerably bent. In general the split-rim flywheel of this type has only 80 per cent. the strength of the same type with a solid rim. The results of the tests are shown in the table.

No. 11 was a solid castiron wheel, shown in Fig. 4. It had six arms, each having a cross-sectional area of 0.99 sq.in. It had a 6-in. face and weighed 70.44 lb. This wheel was held on the shaft by two case-hardened setscrews. Two attempts were made to break it, but each time the fuses burnt out at the switch. The maximum speed attained was 3720 r.p.m., or a peripheral speed of 389.4 ft. per sec.

No. 12 was a duplicate of No. 11. To avoid air resistance, sheet-iron disks were placed on each side of the wheel, and wired together through the spokes or arms. This did not avail anything and the maximum speed obtained was 3380 r.p.m., or a peripheral speed of 353.8 ft. per sec. Further tests of these wheels were abandoned, owing to want of more power, as 15 hp. proved insufficient.

No. 13 was a paper wheel having a solid rim and web, shown before and after being disintegrated in Fig. 5. This wheel was balanced in the web. The rim was held to the web by ⅝-in. wooden pins parallel to the axis of the shaft and approximately 4 in. from center to center. The hub consisted of two heavy cast-iron plates bolted to opposite sides of the center of the web forming the hub.

This wheel burst at a speed of 2820 r.p.m., a peripheral speed of 295.2 ft. per sec. The fracture occurred as anticipated. A part of the rim 1 ft. long flew off, separating itself into one large piece and several small fragments. The wooden pins connecting the rim to the web were broken off squarely. It is thought the fracture was due to the heaviness of the rim and the weakness of the pins connecting it to the web.

No. 14 was a duplicate of No. 13, and broke in the same manner. The fracture was presumably due to the same cause. These two tests demonstrate the lack of strength in the rim of this type of wheel, which suggests a field for future improvement.

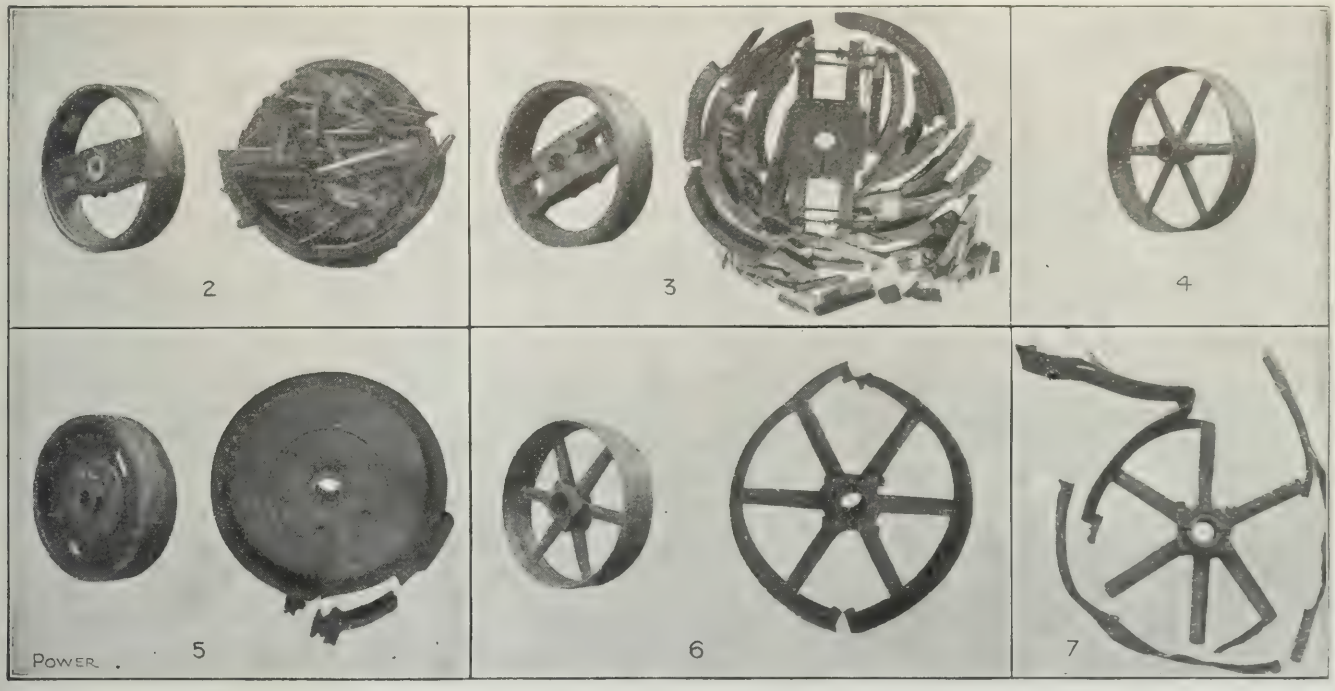
No. 15 was a pressed-steel split wheel; that is, it consisted of two sections or halves.

Fig. 6 shows this wheel before and after fracture had occurred. In this type the rim was riveted to the arms, of which there were six. The sections of the rim were fastened together by a small flange which was riveted at each joint to the rim. A ¼-in. bolt connecting these flanges held the two sections of the rim together. Each arm consisted of two flat pieces of iron 1¾ x ⅛ in. in cross-section, riveted one on each side of a seam on the inside of the rim. This type, having a very light rim, should have stood a very high speed, but it was seen that the rim was very weak at the joints.

The ends of each half of the rim where they were joined together were bent outward, thus rupturing the small flanges and bending the bolts that connected the two small flanges. The instant that a rupture was suspected the power was shut off, and when the motor stopped the rim was found bent out of shape, but not torn from the arms. The speed obtained was 2240 r.p.m., or a peripheral speed of 234.5 ft. per sec.

No. 16 was a duplicate of No. 15. The first fracture occurred in the same manner as with No. 15. Instead of shutting down the motor when the first distortion was suspected, it was continued until the greater part of the rim was torn away from the arms which resulted in a loud crash. The rim was torn entirely away from four of the arms, but a part still remained attached to the other two arms. The metal was sheared out, where the flanges were riveted to the rim, but the rim was sheared out where the arms were riveted to it. The speed was the same as for No. 15, being 2240 r.p.m., or a peripheral speed of 234.5 ft. per sec. In this case the bolt failed, as shown in Fig. 7. This illustration is interesting, as it shows clearly the bending action of the centrifugal force on the bolts, using the inner edge of the flange as a fulcrum.

The test of No. 15 was stopped at the critical moment. To have continued it would have resulted as shown in Fig. 6. This indicates how the joints failed. The joint opened like a clam shell, the bolts and flanges sheared out from the rim, and the arms remained uninjured. The joints in Fig. 6 display the shearing action, or cantilever effect, which in experiment 16 indicates that, as the bolts were strained and the joint loosened, one side of the joint was more affected by the centrifugal force than the other, causing a shearing action in the joint. These wheels burst at a much lower average speed than the paper, solid-wood or even split-rim wooden wheels. The test proves the source of weakness in the rim joint midway between the arms.



FIGS. 2 TO 7. VARIOUS PULLEYS TESTED TO DESTRUCTION

The stresses given in column 7 of the table were computed by the formula

$$\textit{Stress per sq.in. of section, lb.} = \frac{V^2}{C}$$

where C is 100 for wood and paper and 10 for iron. The velocity V is the linear velocity of the center of gravity of the section. The values worked out in the table show to what extent the strength is affected by bending. The above formula has been shown by Mr. Stanwood to represent approximately the tensile strength per square inch of rim section due to centrifugal force.

In every case the rim demonstrates the faulty design followed in modern practice. The surprisingly low bursting speed of the steel wheels proved conclusively

that rim joints midway between the arms should be avoided and these joints placed at the ends of the arms. The strength of the joint at the end of the arm is shown in the tests of the solid-rim wooden wheels, Nos. 1 and 2 as compared with the split-rim wheels Nos. 3 to 10 inclusive. These were duplicates, except that one had a split rim and the other a solid rim.

These tests are the first ever made of paper, wood or pressed-steel wheels. In fact, this is the first time the original fracture of a wheel tested to disintegration has been determined. In all previous tests the casing was so designed that flying particles struck it with such force, that they were again broken, making it impossible to reassemble the pieces. It was due to the remote control that we were able to obtain the bursting speed so accurately.

Balancing wheels by adding weight in the rim between the arms produces a cantilever effect. In the iron wheels a piece of iron the size of a goose egg is sometimes riveted to the rim between the arms to balance it. At very high speeds the bending, due to the centrifugal force of this weight, causes disintegration. The wooden wheels are balanced by inserting iron lugs in the wooden cross-section of the rim which decreases the cross-sectional effective area, weakening the rim, and failure occurs at this point. Wheels should be balanced by weights at the ends of the arms.

Conclusions

(1) Balancing the wheels in the rim causes fracture at low speeds and thereby lowers the factor of safety.

(2) Rim joints midway between arms are serious defects and materially reduce the bursting speed.

(3) The solid-web-and-rim paper flywheel of the same type as tested (Fig. 7-8) will safely withstand a rim speed of 106 ft. per sec. By properly strengthening the rim the speed may be materially increased and a sufficient factor of safety retained.

(4) Wood flywheels with solid rim (Fig. 2) have an ample factor of safety at a rim speed of 90 ft. per sec. if the wood is of good quality and the same design is followed out. The speed of this wheel may be increased by using lighter bolts at the ends of the arms, as the weight of the bolts caused the original fracture in 90 per cent. of the wheels tested.

(5) Wood split wheels of the same design and material as shown in Fig. 3 will have a sufficient factor of safety at 72 ft. per sec. rim speed. The design may be improved by using lighter bolts and discontinuing the practice of balancing in the rim.

(6) The iron flywheels, Fig. 4, which were not tested to destruction, if of good iron and free from serious cooling strains will be safe at a rim speed of 120 ft. per sec. These wheels stood the best test of any of which there is record. The bending in the rim, due to centrifugal force, is so slight that it may be neglected in this type of wheel.

(7) Steel wheels of the split-rim type, Fig. 6, are unsafe at speeds above 80 ft. per sec. It has been noted that in various cases wheels of this type gave away through a solid rim and without failure of the bolts, although the strength of the bolts was less than one-third that of the rim section, which shows that the strength of this joint as usually calculated has very little to do with the operative strength of the flywheel.

It is not necessary to go farther into the design and the safe speeds of various types of flywheels, as this subject has been splendidly treated by F. A. Halsey in the Mar. 28 issue of the *American Machinist* in an article entitled "Design of Small Flywheels."

The Design of Ignition Arches

By Joseph Harrington

Principles underlying the design of ignition arches, with reference both to the arch that absorbs heat from combustion of the volatile at the front of the furnace and to that which receives radiant heat from the incandescent fuel at the rear. Flat arches compared with sprung arches.

An ignition arch is a refractory tile roof placed over part of a furnace for the purpose of producing or stimulating the ignition of fresh fuel and is used largely in connection with stokers in boiler furnaces. The form and extent of this arch have undergone radical modifications in recent years and the principles governing its influence on the fuel bed were formerly considered quite different from those which now appear to control its effects.

A few years ago ignition arches were built as short as 2 ft. and were seldom over 4 ft. in length. They were placed at slight inclines to the horizontal with the rear tilted upward, and their height was measured from the grate surface without reference to the distance from the surface of the fuel to be ignited. The actual distance was slight, the under surface of the arch being placed but 2 or 3 in. above the fuel.

The volatiles, liberated by the heat of the furnace, ignited at the surface of the coal and burned in contact with the under side of the ignition arch. They imparted some of their heat to the firebrick which was raised to a state of incandescence and radiated heat to the incoming fuel with sufficient rapidity to perpetuate the ignition and render the operation continuous.

No account whatever was taken of the temperature produced by the combustion of the fixed carbon at the rear of the furnace, and that is just where the most desirable source of heat was neglected. One thing that developed in the writer's experience with radiating arches of uniformly heated surface was their observance of the following laws:

1. The maximum intensity of radiation was perpendicular to and at the center of the radiating surface.

2. The temperature of any point within the affected zone was inversely as a power of the distance from the incandescent surface, disregarding the influence of the fuel bed or burning volatile.

A recognition of these two facts helped in many places where ignition was poor or uneven. The first led to a horizontal arch set to throw the center of radiation farther forward, and thus increase the effectiveness of the radiation, and the second is the reason for the greater uniformity of igniting effect with flat arches as compared to sprung arches. Fig. 1 shows the relative effect of a variation in the inclination of the arch and Fig. 2 the variation in the temperature under a sprung ignition arch. The sketch A, Fig. 1, illustrates the heat distribution with a horizontal arch, the semicircular shaded portion showing approximately the variation in intensity of radiation. It will be noted from this that the area subjected to radiation includes practically the entire extent of fuel surface directly beneath the arch. Sketch B illustrates the variation in the extent and intensity of the radiation when the arch is tilted, and sketch C the still greater reduction in ignition effect when the arch is tilted still more.

Fig. 2 is a sectional view with lines to indicate the variation in temperature at different points in the furnace, the lines *a*, *b*, *c* and *d*, defining zones of equal temperature and consequently of equal ignition effects, due to the heat radiated from the under surface of the arch and the adjacent side walls. It will be observed from this that those portions of the fuel bed nearest the sides are subjected to a greater intensity of ignition than is the center of the fuel bed. The result of this is the more rapid ignition and consequently earlier completion of combus-

tion along the sides than at the center. The rapidity of ignition is, of course, in direct proportion to the temperature, so that with such an arch, the combustion proceeds more rapidly at the sides than in the middle. This undesirable effect is offset somewhat by making the arch as flat as possible, but the influence of the curvature remains, and on wide furnaces is serious. A flat arch provides a uniformity of radiation and the ignition effect is of necessity the same for the entire width of the furnace.

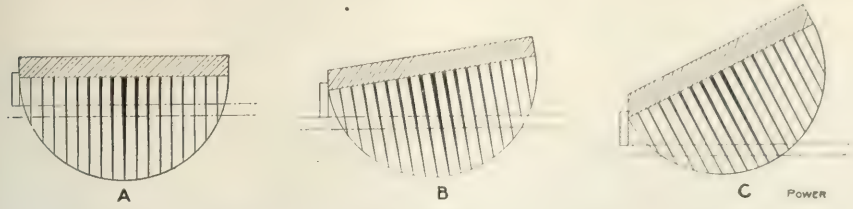


FIG. 1. HEAT DISTRIBUTION WITH HORIZONTAL AND INCLINED ARCHES

In accordance with the second law of radiant arches, the nearer the incandescent surface of the arch approaches the fuel to be ignited, the greater will be the intensity of the ignition. It is obvious that when the distance is but a few inches, that a slight variation will decidedly affect the influence of the arch. It has heretofore been the common practice to build these arches at a definite distance from the grate surface, no account being taken of the location of the surface of the fuel bed. It is quite obvious, however, that this is or should be the deciding factor and the arch can only be intelligently set when the thickness of the fuel bed is predetermined.

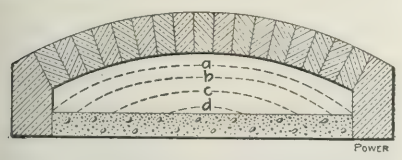


FIG. 2. VARIATION IN TEMPERATURE AT DIFFERENT PARTS OF FURNACE

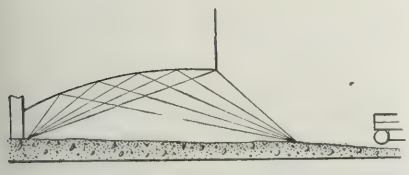


FIG. 3. REFLECTING ARCH FOR TRANSFERING HEAT FROM BACK TO FRONT OF FURNACE

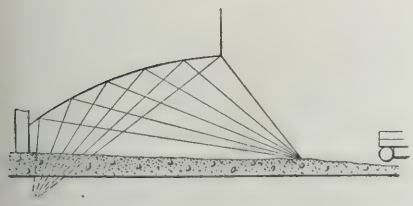


FIG. 4. ARCH SET OUT OF FOCUS

The writer early gave this matter consideration and has followed the practice of setting igniting arches purely with reference to the surface of the fuel bed and not with reference to the grate surface. On the other hand, there is a certain limitation to the proximity to the fuel bed with which an arch can be set. There must be sufficient space to permit of combustion of the volatile matter and to permit the free flow of gases away from the fuel bed. Otherwise, the combustion is smothered and the value of the arch is entirely lost.

All of the foregoing discussion has related exclusively to that type of arch designated as radiant or those arches which after absorbing heat from combustion of the volatile constituents of the coal have become incandescent and are then in a condition to radiate heat to the incoming fuel.

As suggested, no attention had heretofore been paid to the influence of the rear end of the fuel bed. The development of new types of stoking apparatus and the necessity of igniting various grades of coals resulted in experimentation with arch slopes and lengths in connection with various kinds of fuel and various types of stokers. During the course of this work, it became apparent that there was another source of heat than the combustion of the volatile at the front of the furnace and that this source of heat was far more reliable and abundant than the other. Attempts were then made definitely to utilize this heat by means of the ignition arch, which was set in various positions and the degree of igniting effect carefully noted. It was early apparent that the old ideas concerning these arches were only half correct and a new theory as well as a new practice has been the outgrowth of such experience.

An analysis of the setting of an arch successfully utilizing the heat from the rear of the furnace showed that it was set just as if a ray of light was to be reflected from the rear to the front; see Fig. 3. The rays of heat follow exactly the well known law of the reflection of light, namely, that the angle of incidence equals the angle of reflection.

Space will not permit setting forth all the considerations which govern the correct design of a reflection arch. The essentials can be worked out mathematically and with a given length of fuel bed and thickness of fire, an arch can be designed that will exactly reflect heat from the rear or incandescent portion of the fuel bed to the point of entrance of the green fuel. Knowledge of the location of the center of supply with various grades of fuel, the effect of varying percentages of moisture and ash on the temperature of the glowing coke and the rate of absorption of heat by the fresh fuel must be had. There are many combinations of these variables which, taken with the influence of the surrounding brickwork, make the exact locating of a reflecting arch a complex problem. Therefore, the following treatment of this subject will be confined to the simple problem of a clean, free-burning coal on a chain grate.

Fig. 3 shows a reflecting arch which is designed to transfer the heat from a single point in the incandescent fire to a single point in the front of the furnace, and is made to illustrate the principle upon which the entire subject is based. The arch surface assumes a concave form and is indeed nothing more than a condensing mirror with the focus close to the junction of the fuel surface and the stoker gate. Such an arch is undesirable, not only from a construction standpoint but because it concentrates the reflected heat too much.

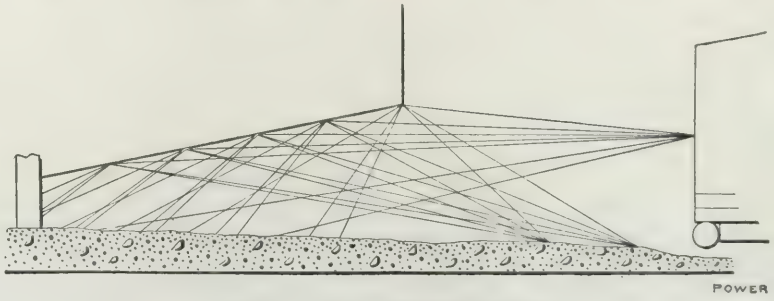


FIG. 5. FLAT INCLINED ARCH

FIG. 6. LOW HORIZONTAL ARCH

The igniting influence must be of some duration in order to thoroughly inflame a reluctant fuel so that this arch would

be actually of greater effect if it were so set as to be "out of focus" as illustrated by Fig. 4.

The objections to the curved arch can be overcome without any departure from the governing principles by the use of a flat inclined surface; see Fig. 5. In this the heat rays emanating from several points of the incandescent fuel are shown reflected from the arch, striking the fresh fuel near the point of entrance, yet sufficiently distributed to give the required duration.

The ignition arch defines the extent of the zone of ignition of the furnace of which it is a part and just as long as the fuel remains under the arch, it must be considered as still subjected to the igniting effect. An ignition arch may, therefore, be too long. Every inch of arch beyond the amount required for complete and proper ignition is a detriment to the furnace. This, then, becomes one of the matters to which all the conditions of the fuel and furnace contribute and it is one which can only be successfully solved in the light of a varied experience.

The law governing the radiation of heat from an incandescent arch applies equally to an incandescent fuel bed so that the height of the arch above the fuel exerts a material influence on its ability to collect the heat rays from a point at the rear. This is well shown in Fig. 6, wherein the low horizontal arch is well designed to prevent most if not all of the heat at the rear of the furnace from coming forward either directly or by reflection.

Far removed as it may appear, the brickwork of the bridge-wall should exert a considerable influence on the ignition, but the low arch effectually cuts off this desirable heat. This type of arch, therefore, is in no sense reflecting and in spite of its continued use is restricted in its successful application to a few limited conditions of coal and draft. It is herein shown as illustrating a design that has persisted throughout the period of improvement in furnace design. Today, every element contributing to the progressive combustion of coal has been carefully studied. The requirements of each stage are comparatively well known and the heat distribution can be scientifically controlled. The most potent element of such control is unquestionably the reflecting arch.

Safe Piping for Boiler Plants

By David Moffatt Myers

A typical layout in which safety and economy go hand in hand. A feature is two valves between the boiler and header and two valves on the blowoff.

Provision for safety in their boiler plants is shamefully if not criminally neglected by many owners. Every article that is published which is written with a view to improve conditions for safety of workmen and operatives deserves to be read and to be studied in proportion to its authority. The design of steam and blowoff piping in boiler plants, especially factory boiler plants, offers a large field for improvement in reducing danger to the men and at the same time simplifying and bettering the plant from an engineering standpoint. Reference is made to the article bearing on this subject on page 488 of the Oct. 1 issue of POWER.

In the same connection is submitted herewith a blueprint of the steam-piping layout for a plant recently designed by the writer, which consists of four 6½x 20-ft. horizontal tubular boilers to operate at 125 lb. working pressure. In order to increase the safety of the piping to operatives, and to provide for proper drainage and expansion, this plan is submitted for the attention of owners who will not, or think they cannot, afford the services of a competent engineer for designing their piping system. The several points that demand special attention are as follows:

Two Valves between Boiler and Header

Each lead has two valves between the boiler and the header, a provision that should always be made. The expense of the extra valve is slight as compared to the value of a man's life. If the usual single valve gave way during the internal cleaning or inspection of a boiler, he would be trapped therein and killed by the inflow of steam from the boilers connected to the header. With two valves the man's chances are doubled.

Some insurance inspectors insist on opening a flange union between the stop valve and the boiler, and slipping in a blind flange for their additional protection when in the boiler. Most of them, however, are careless. No doubt familiarity breeds contempt, but it does not lessen the risk.

In the plan given, the valve closest to the boiler is of the nonreturn type and acts much like an ordinary check valve. In case there should be a failure in any boiler in the battery such as the blowing out of a tube, the steam from the other boilers on the line would be prevented from entering the injured boiler. This precaution greatly reduces the extent of possible damage. Furthermore, this nonreturn valve would automatically prevent some careless operative from turning steam into an empty boiler which is off for cleaning or repairs and perhaps has a man working inside.

The construction of the leads from the boiler to the header admit of perfect expansion in every direction, thus avoiding bad strains in the piping. Owing to the location of the steam valves in the horizontal run of the lead, it is impossible for dangerous water to collect in the piping when a boiler is shut down. Any condensation flows either back to the boiler or to the main header.

In connection with the nonreturn valve, the question may be asked why a combination automatic nonreturn and stop valve is not used. The answer is that in seventy-five cases out of a hundred that were recorded by one of the large boiler-insurance companies, only twenty-five out of the hundred operated when they should have done so. In other words, actual experience with these double-function valves has indicated that in general they are not sufficiently reliable to be counted upon in all emergencies. If not used frequently, and carefully watched, they are liable to stick and refuse to work when most needed. Consequently the valve with the simpler mechanism, although of the single function (that of preventing steam from flowing into the boiler) has been selected.

Main Stop Valve

Means for preventing the sudden and dangerous emptying of the boilers by the bursting of an engine steam pipe or the knocking out of a cylinder head are provided at the main stop valve *M*, located between the battery of boilers and the steam main leading to engines and heating lines. This main stop valve is provided with a chain wheel, the chain of which passes around another wheel within easy reach of the firing floor.

In case of a blowout, such as mentioned, the engine-room crew will be free to escape and save themselves. For by spinning around the crank on the lower chain wheel on the safe side of the engine-room wall, the main stop valve can be quickly closed and with safety to all concerned. Fig. 2 shows the means for operating this main stop valve. A reduction gear is not used since quick closing is desired.

Expansion and Drainage

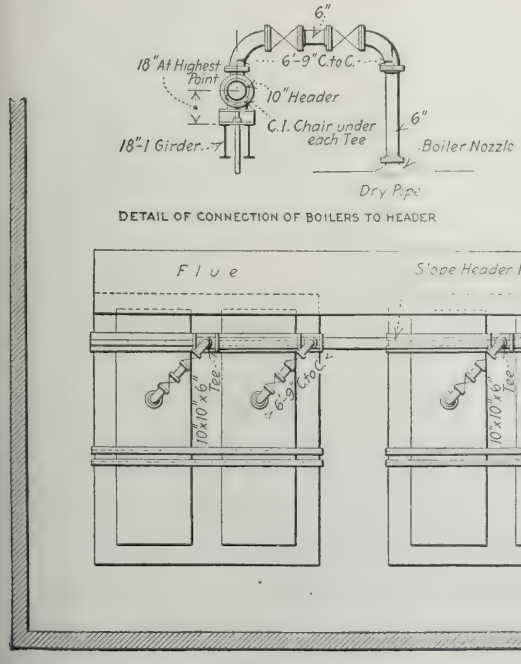
As for the safety of the leads and headers, these are all of extra-heavy

piping. The valves, flanges, tees and other fittings are extra heavy. All elbows have long radius to reduce friction losses and improve expansion. The leads may be equally well run at right angles to the boiler header instead of at an acute angle which was done in this case on account of limited space. The 10-in. boiler header is sloped downward in the direction of the natural flow of the steam so that any condensation will drain with the current. A drip pipe will be taken out of the bottom of the wall header or main which is the lowest part of the boiler-room piping, and the condensation will be caught and discharged by a trap.

The wall header or main which lies at a right angle to the boiler header is anchored to the boiler-room wall. The piping to the left is free to expand without strain, owing to the flexible construction shown, the piping to the right has sufficient expansion by virtue of bends to the engines and other lines. Steam should always enter a header or main from the top and be taken out from the top to engines and mill lines, the headers or mains to be drained as described.

Two Valves on Blowoff

In addition to the above precautions each boiler should have two valves on its bottom blowoff pipe between the boiler and the blowoff header. If there is a surface blowoff pipe connected to the blowoff header this pipe also should have two valves separating it from the header.



These extra valves are recommended first for the protection of a man who may be inside of a dead boiler to prevent his injury or death in case of the possible giving way of a single valve for either of these purposes. Secondly, these extra valves are economical from an engineering standpoint as they halve the possibility of trouble and loss by leakage which is most common to blowoff valves.

Safety and Efficiency

There is one thing more that requires emphasis. When a man is inside of a boiler, it should be made an impossibility for any thoughtless or vicious

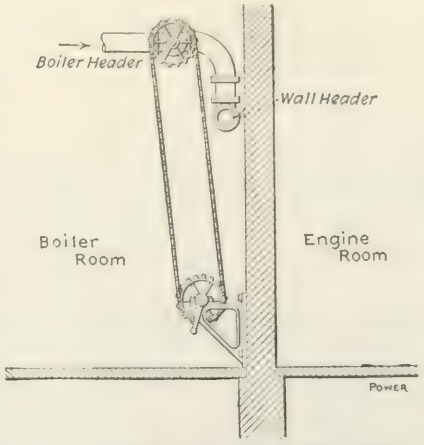


Fig. 2. Device for Operating Main Stop Valve

person to turn hot water or steam into that boiler. This can be accomplished by padlocking the steam and blowoff valves, and then by giving the keys of the padlocks into the keeping of the man who is to enter the boiler.

The insurance inspector has required the insertion of a connection for the fire pump at *A* in Fig. 1, so there would always be available a supply of steam for this purpose even when the main stop valve might be closed.

Perhaps it will be noticed by some readers how strangely coincidental seem to be the requirements for safety with those of good engineering in steam-piping practice. Stepping up to a broader view, which includes a study of the whole plant, this same coincidence of engineering economy and welfare of operatives is found by the investigator. The more comprehensive the study, the more strikingly will it be found that the highest economic efficiency obtains in those plants or industries where brotherly consideration of the welfare of the workers is the ruling spirit in the mind of owner and manager.

It has long been in the minds of many that there may be an essential underlying and close connection between scientific and moral laws. If moral laws do not sufficiently appeal to some owners and managers to protect their boiler piping and workmen from accident, they will be wise to heed the laws of science and engineering in order to insure their personal profit and to improve the efficiency of their business.

New Cheat River Hydroelectric Plant

The West Penn Traction & Water Power Co., Pittsburgh, has completed arrangements for the equipment of its new 40,000-kw. hydro-electric plant, the first of a series of developments on the Cheat River, at the Pennsylvania-West Virginia line. The electrical apparatus includes four 10,000-kw. Allis-Chalmers vertical waterwheel type alternators, two 300-kw. motor-generator exciter sets and accessories of the same company's design.

The territory to be served by this system completely encircles the great Pittsburgh industrial district and current will be furnished for electrical operation of many of the steel mills, coal mines, etc.

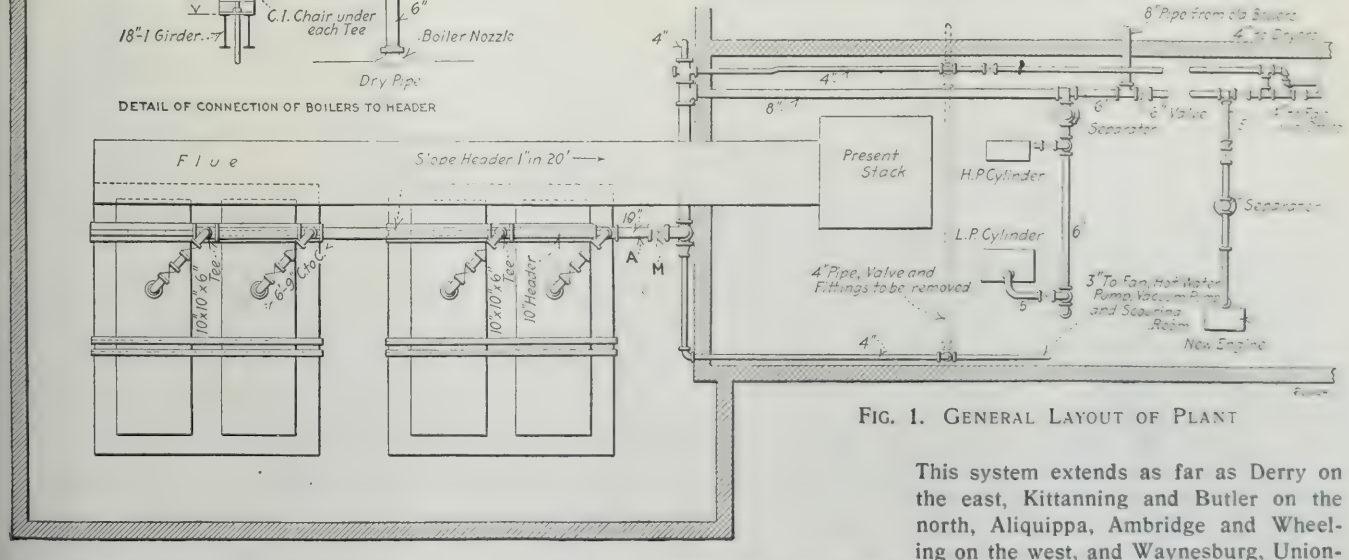


Fig. 1. General Layout of Plant

This system extends as far as Derry on the east, Kittanning and Butler on the north, Aliquippa, Ambridge and Wheeling on the west, and Waynesburg, Uniontown and Connellsville on the south. It involves 259 miles of 110,000-volt lines.

The Port Arthur Light & Power Co., Port Arthur, Tex., recently organized with a capital stock of $600,000, has taken over the property of the Port Arthur Traction Co., says *The Tradesman*.

Small Westinghouse Steam Turbine

The steam turbine has won its greatest success in large capacities but its phenomenal advancement and the concurrent development of rotary pumping and blowing apparatus, and of electric generators of high rotative speed, has extended and emphasized the demand for units of smaller capacity.

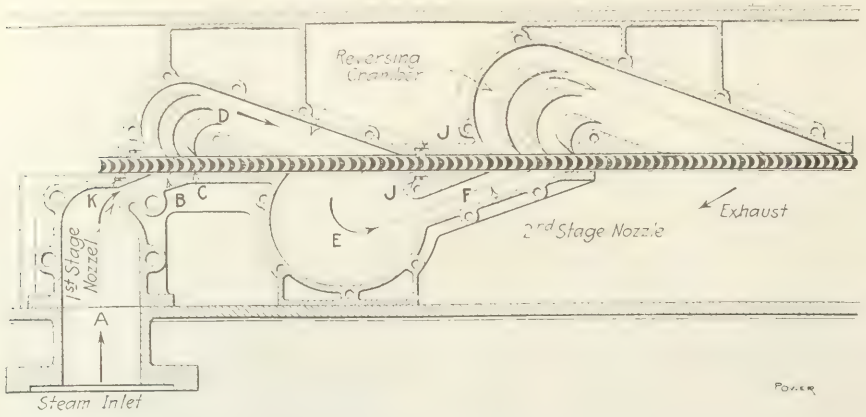


FIG. 1. DIAGRAM OF THE COURSE OF THE STEAM IN THE SMALL WESTINGHOUSE TURBINE

should run at speeds allowing them to be directly connected to such apparatus, and which without multiplication of runners, should be able to abstract a large proportion of the energy due to the considerable pressure- or temperature-fall available.

Since the total amount of steam required by a turbine of small capacity is not greater than can be discharged by a few nozzles of moderate size occupying only a small portion of the periphery of the runner, and since the steam must be made to act a number of times upon a wheel revolving at any practicable rim velocity, in order that the energy induced by expansion between the limits usually available may be absorbed, the natural impulse is to make the steam act again upon that portion of the wheel not occupied by the initial nozzles. This method is described in the patent of Wilson (1849) and that taken out by Ferranti prior to 1870, and successfully used in the Elektra turbine (see POWER, Apr. 6, 1909). It is shown diagrammatically as used in the small Westinghouse turbine in Fig. 1.

Steam enters the chamber *A*, is expanded in the nozzle *B*, passes through the blades *C* of the revolving wheel, is reversed in the chamber *D*, redirected upon the wheel, as shown by the dotted arrow line, and when it arrives at the chamber *E* has been brought nearly to rest. It is then expanded again in the second-stage nozzle *F* and its velocity abstracted by two more passages through the wheel as before, coming out finally into the casing at *H*, with its velocity again largely abstracted, and ready to

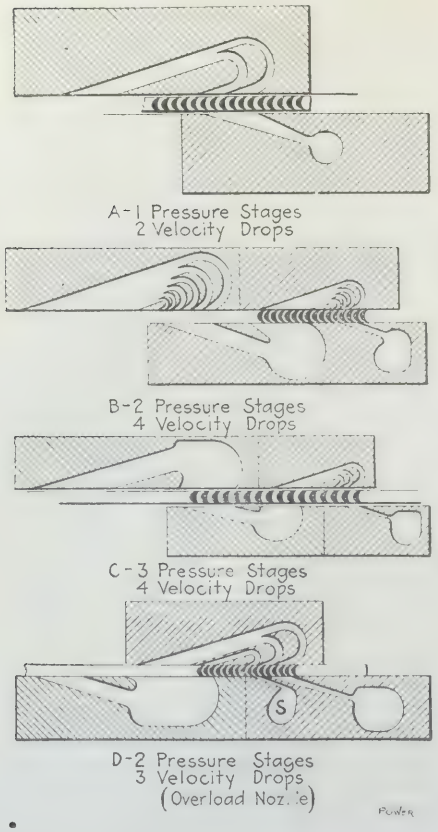


FIG. 2. VARIOUS COMBINATIONS OF PRESSURE AND VELOCITY STAGES

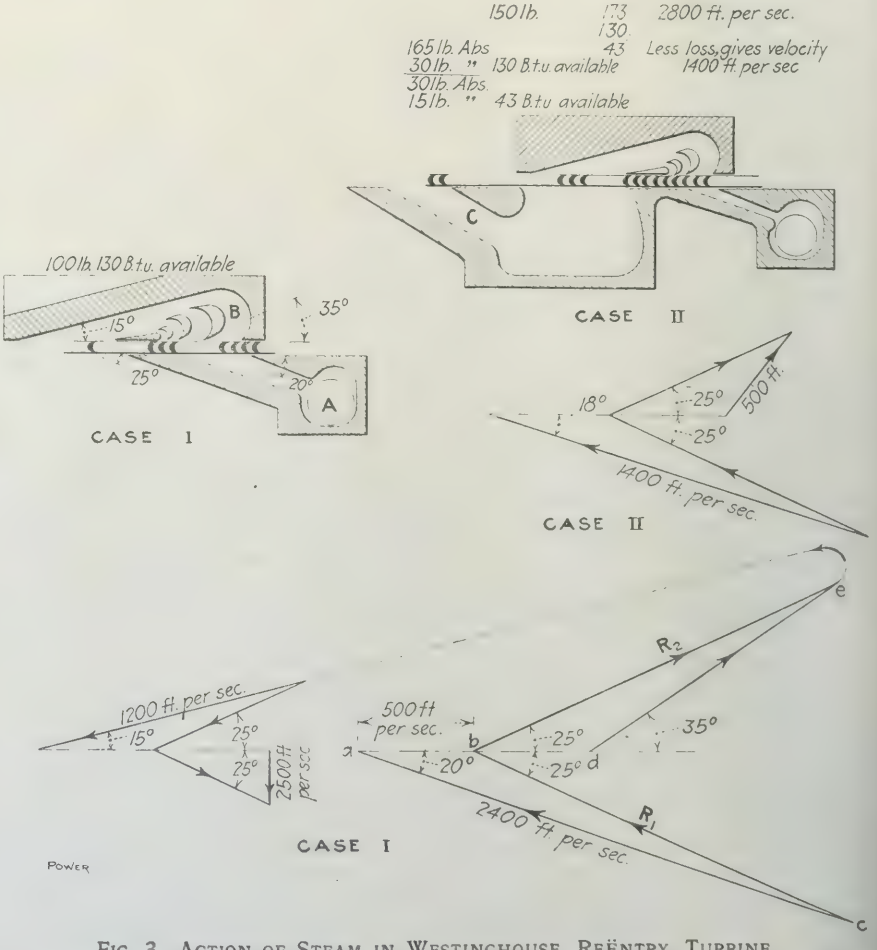


FIG. 3. ACTION OF STEAM IN WESTINGHOUSE REËNTRY TURBINE

find its way to the exhaust outlet. Guide frames are used in the reversing chambers to redirect the steam without shock or eddy. A packing is provided at *J* to prevent leakage between stages and at *K* to prevent leakage from the high-pressure stage into the casing. The wheel runs in steam of the exhaust pressure and low density with less windage than as though it were in contact with the denser steam of the initial pressure, as is the case with the disk in the initial stage of a multicellular machine.

It is apparent that this arrangement permits the designer considerable latitude in the choice of stages both pressure and velocity and that to a given rotor in a given cylinder or casing any number and variety of nozzles may be fitted, up to the limit of the blade circle. For units under 50 hp. the descent from boiler to exhaust pressure is made in one expansion stage and the velocity thus generated is taken out by two passes through the wheel. Such a turbine would consist of a single nozzle and reversing chamber, as shown in the upper sketch in Fig. 2, the small amount of steam involved usually not warranting the complication even of the second-stage nozzle and reversing chamber.

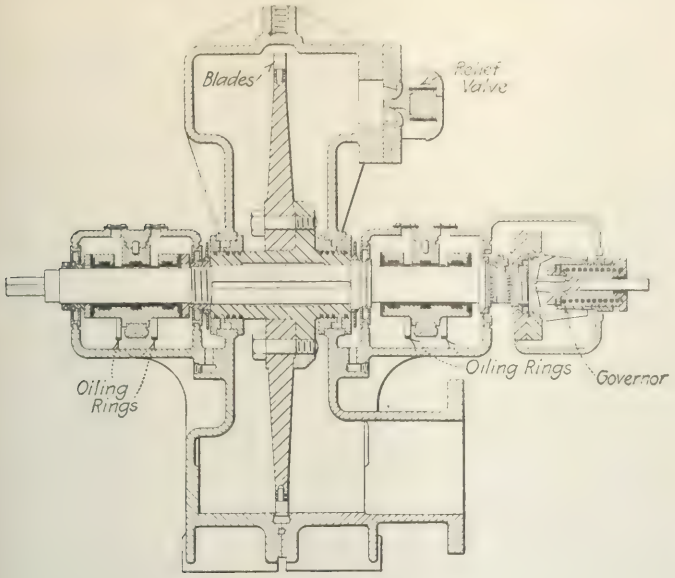


FIG. 4. THE POWER TYPE

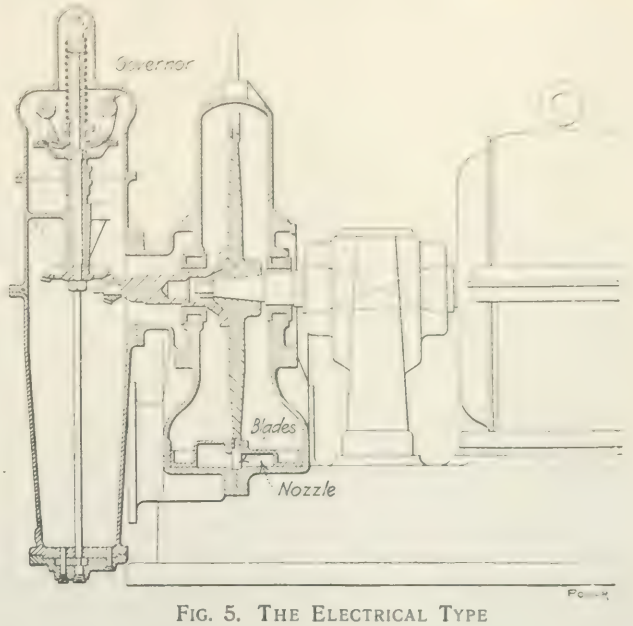


FIG. 5. THE ELECTRICAL TYPE

The nature of the problem will be apparent from the consideration of a couple of concrete cases.

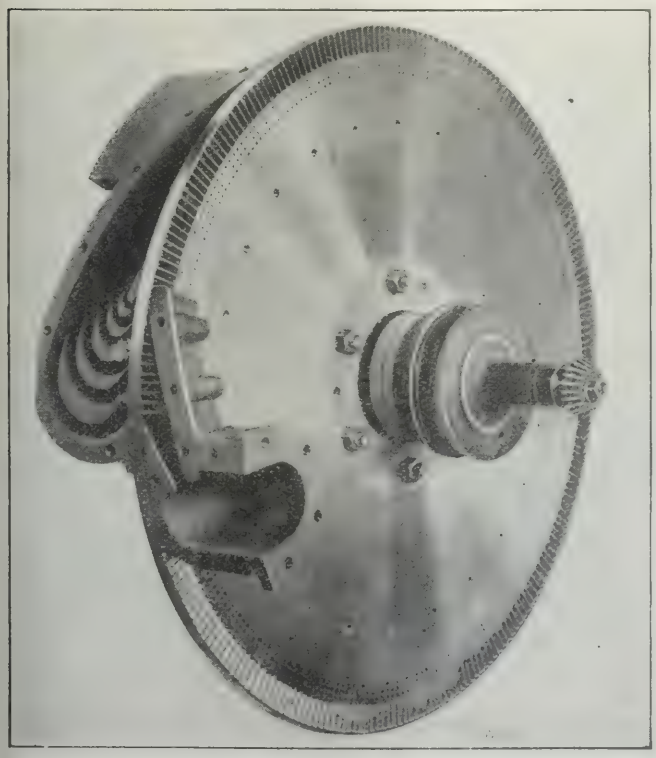


FIG. 6. WHEEL, NOZZLE AND REVERSING CHAMBER OF A NON-CONDENSING TURBINE

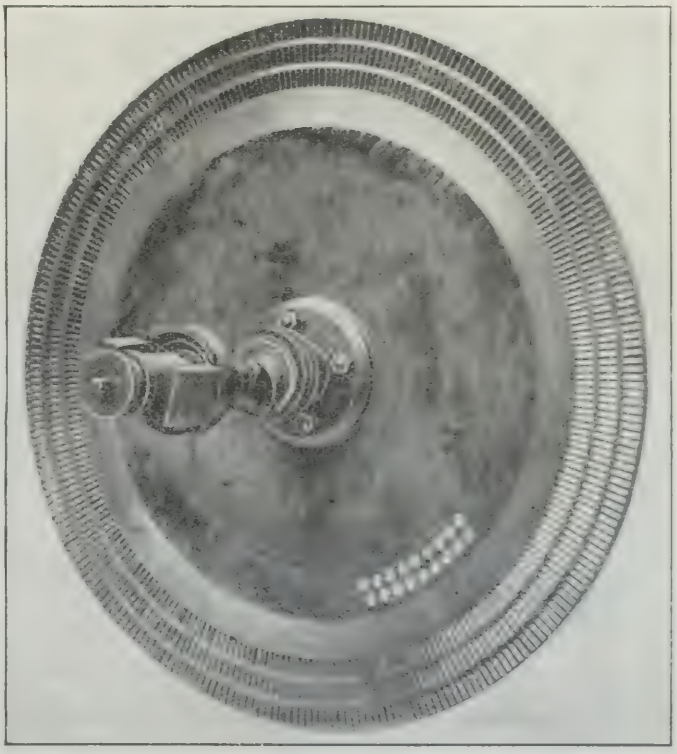


FIG. 7. THE RUNNER OF A 450-HP. CONDENSING TURBINE, 1000 R.P.M.

CASE I: Assume 100 lb. initial and 15 lb. final pressure both absolute, steam initially dry saturated.

This drop makes available about 130 B.t.u. per lb. which corresponds with a

velocity of some 2550 ft. per sec. With a proper allowance for nozzle friction, the velocity of the jet acting upon the wheel becomes about 2400 ft. per sec.

Taking a blade speed of 500 ft. per sec. and a nozzle angle of 20 deg., the direction in which the steam actually approaches the moving bucket and its velocity in that direction would be indicated by the direction and length of the line *bc* in Fig. 3, completing the triangle *abc*, of which the line *ac* (representing by its length and position the 2400 ft. velocity and the direction of the issuing jet) and the line *ab* (representing by its length and position the velocity and direction of movement of the bucket) are the other sides. The bucket changes the direction indicated by the arrow-head R_1 in Fig. 3, to that indicated by R_2. Reducing the length of this line in proportion to the loss of velocity by friction in the bucket and setting off *bd* equal in length to the bucket speed and drawn in the line of the bucket motion, the line *de*, indicating by its length and direction the velocity and taken out, the length of the line *de* shows that there is still enough velocity to make it worth while to redirect the jet upon the wheel.

This is done through the reversing chamber *B*, friction decreasing the absolute velocity to 1200 ft. per sec. and thus establishing an outlet angle of 15 deg. for the blade angle of 25 deg. Under these conditions the steam jet issues at right angles to the wheel with a small residual velocity of 250 ft. per sec., as shown in the lower left-hand diagram of Fig. 3, thus reducing the loss from this source to a minimum.

CASE II: Assuming now the pressure range from 165 lb. to 15 lb., both absolute. The steam is expanded from 165 to about 30 lb. in the first stage. The available heat would still be 138 B.t.u. and therefore the velocity developed remains as above. Then the high-pressure

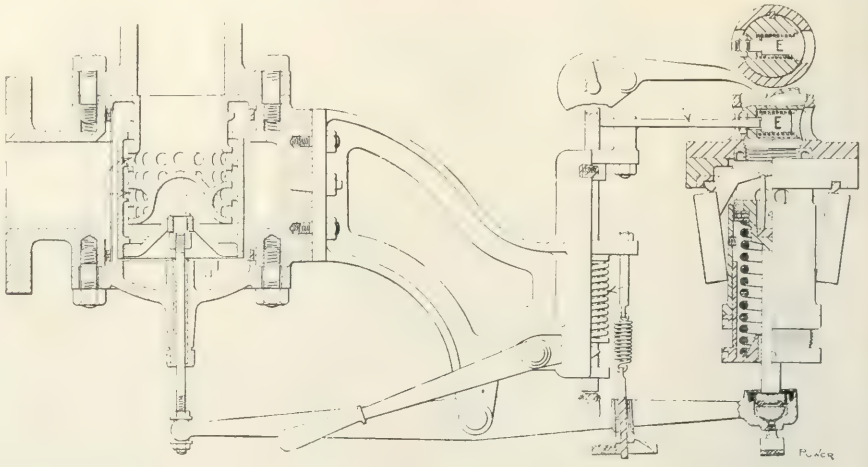


FIG. 8. THE ADMISSION VALVE

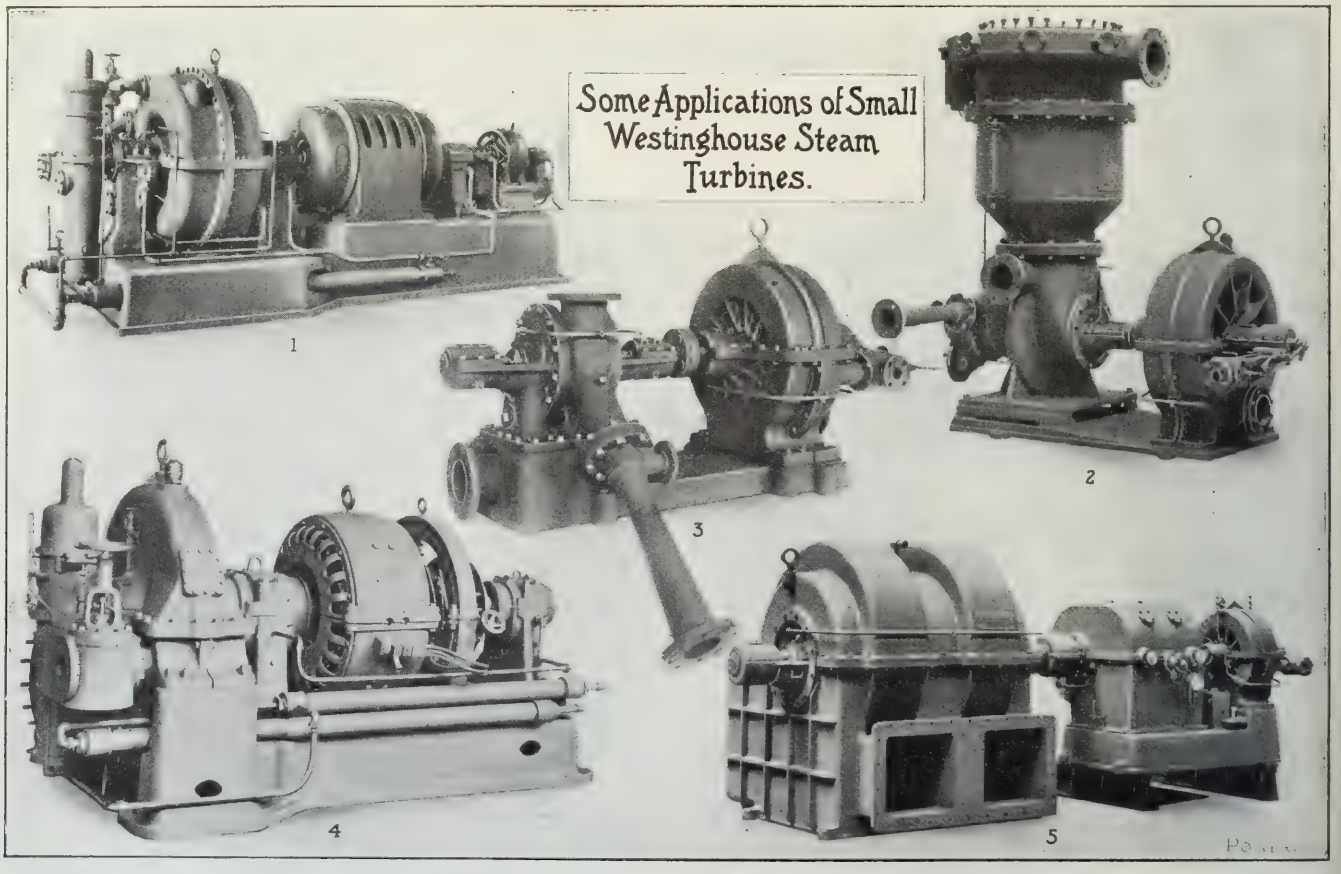


FIG. 9. 1. 100-KW. CONDENSING TURBINE DRIVING 60-CYCLE GENERATOR WITH DIRECT-CONNECTED EXCITER AT 3600 R.P.M.—2. LE BLANC CONDENSER, BOTH PUMPS DRIVEN BY TURBINE—3. AIR PUMP OF LE BLANC CONDENSER DIRECTLY CONNECTED TO TURBINE—4. A 100-KW. 125-VOLT DIRECT-CURRENT NON-CONDENSING SET—5. *A* 25,000-GAL.-PER-MIN. PUMP, TURBINE DRIVEN WITH REDUCTION GEAR

direction of the steam as it issues from the blade, is found. Although the energy remaining in the jet is only as the square root of its velocity, so that if the velocity is halved three-quarters of the energy is

nozzles and reversing chambers *A* and *B* in both cases will be similar except that in Case II they will be of less width, due to the greater steam density. However, with the pressure of about 30 lb. at the end of the first stage, a velocity of some 1400 ft. per sec. may be developed by further expanding to 15 lb. absolute which is accomplished by the addition of the nozzle *C*, shown in the sketch illustrating Case II in Fig. 3. Had the initial pressure been higher, the velocity at *C* might have been sufficient to warrant the addition of another reversing chamber in the second stage. Depending upon the pressures and speeds used a third stage may be profitably employed. Numerous combinations are shown in Fig. 2.

Small turbines are naturally divided into two classes, those used for driving direct- or alternating-current generators, and power turbines such as are applied to blower or pump drive. A subdivision may be made, depending upon whether the service is condensing or noncondensing, although in small sizes the latter is the more common case.

Figs. 4 and 5 show sections of the noncondensing power and electric machines. While exactly similar in operating principle, there is some difference in detail. For power work, the shaft governor gives amply close regulation, but in the electric machines where the speed must be controlled within narrow limits over a wide range of load, the governor employed is the same as that used on the large Westinghouse turbines. But two bearings are used in the noncondensing electric units, one on each side of the generator, the turbine disk being overhung, as shown in Fig. 4. These bearings are lubricated by a closed system served by a pump on the governor shaft. The power-turbine bearings are ring-oiled. The construction of the glands for preventing the leakage of steam from the casing, is also different, the power turbines having free packing rings fitted in grooves turned in the hubs of the disks, whereas the electric machines have a hydraulic sealing device.

Fig. 6 shows the wheel of a noncondensing power or electric machine with one nozzle and reversing chamber clamped in the proper position relative to the wheel. As the photograph indicates, these chambers are bolted to the inside of the turbine casing which forms one wall of the steam passage.

The casing is split horizontally in both types, and the upper half may be removed without disturbing any connections. The lower half, which carries the nozzles and reversing chambers, is also split vertically in the plane of the wheel. Since the steam is expanded to approximately atmospheric pressure in the nozzles, the casing is designed for a moderate final pressure, any excess of which is prevented by a relief valve which is indicated in Fig. 4 and may be seen in the various photographs. The runners of the turbines, both electric and power, are of boiler-plate steel bolted to the hubs. Blades are formed with roots which set in a groove in the periphery of this disk, and are through-riveted. The material used for the blades is nickel-iron which is of high tensile strength, offers great resistance to corrosion and erosion, and has practically the same coefficient of expansion as the disk, thus avoiding strains due to unequal expansion when the casing is filled with steam.

For condensing service, where steam volumes become great, a modified construction is necessary. The power turbines which run at slow speeds are more affected than the electric machines. Fig. 7 shows the runner of a 450-hp. condensing turbine to operate at 1000 r.p.m. The first-stage nozzle and reversing chamber of such a machine directs the steam upon the outer row of blades only. The second-stage nozzle makes use of the two outer rows, and the third and subsequent passes, if the latter are necessary, use all three rows of blades. In the electric machines the blade speed is such that except in the high-pressure stage no reversing chamber is necessary. During the rest of the expansion, there is a set of nozzles for each passage of the steam through the blades.

Fig. 8 shows the admission valve used on all the impulse turbines made by the Westinghouse Machine Co. The construction combines poppet and slide-valve characteristics. When closed, it is double-seated. After lifting from the seats approximately ⅛ in., the sleeve uncovers ports in the valve body which admit steam to the nozzles. This construction prevents scoring of the valve seat and consequent leakage when the turbine is operating at light loads. The connection of the valve stem with the governor linkage, and the operation of the safety overspeed stop are also evident from the engraving.

The first small turbines built by the Westinghouse Machine Co. were used to drive the discharge and air pumps of the Westinghouse-Leblanc jet condenser. There has been developed since then a complete line of turbine-driven power-house auxiliaries, including boiler-feed pumps, blowers, hotwell pumps and circulating pumps for surface condensers. Since the head against which circulating pumps for surface condensers usually discharge is relatively low, and the volume of water to be handled is large, the direct connection of a turbine to the pump is hardly possible on account of the slow speed involved. In several instances of large condenser installations, this difficulty has been overcome by the use of a reduction gear between the turbine and pump. In the group of photographs reproduced herewith is shown a 25,000-gal.-per-min. pump so driven, discharging against 13½ ft. total head. The pump runs at 300 r.p.m. and the turbine at 3000. The gear efficiency, due to the flexibility supported pinion shaft used, is 98 per cent. The group shows various other applications of the small turbine.

Ulco Lead Rope

This rope is made of extremely thin, long fibers of pure lead, and is principally used for jointing gas and water pipes where liquid metal could not be poured into the joint, owing to restrictions.

ULCO LEAD ROPE

When wrapped in cheesecloth or sheet rubber, lead rope makes an efficient and satisfactory metal packing, and can also be used on piston rods, valve stems for water, gas and steam.

When folded in cheesecloth or rubber and placed between flanges of high-pressure lines, lead ropes makes a strong gasket. There is no limit to size or thickness, and it can be used in any form or shape. The illustration shows a spool of Ulco rope, and the manner in which it can be separated. It is manufactured by the United Lead Co., 111 Broadway, New York City.

How Heat Flows into the Boiler Water

Henry Kreisinger and Walter T. Ray, of the Bureau of Mines, in their recent work entitled "The Transmission of Heat into Steam Boilers," published by the Bureau as its Bulletin 18, describe the passage of the heat from the furnace to the boiler as follows:

The accompanying sketch shows diagrammatically the modes of heat travel into boiler water.

The diagram shows the metal plate covered with a layer of soot on the gas side and with a layer of scale on the water side. Next to the soot layer and entangled in its recesses is a layer or film of motionless gas. This film of gas under ordinary conditions adheres so tightly to the soot or the metal that it may almost be considered a part of the solid plate. It is reasonable, therefore, to assume that the dry surface of the plate is situated somewhere within this film of gas, as indicated in the sketch. The wet surface of the heating plate is perhaps located on the inside of the boiler in a film of water and steam adhering to the layer of scale, or, if the boiler is clean, to the metal plate.

In any steam-generating apparatus the heat is produced in the furnace by the burning of fuel. As the process of combustion goes on the heat evolved is immediately absorbed, partly by heating of the fresh fuel but mainly by the gaseous products of combustion. The absorption of heat by these substances causes a rise in their temperature, so that by observing the temperature one can readily determine how much heat they contain.

The heat evolved and contained in the gaseous products of combustion is transferred through the gas-filled space and then transmitted through the heating plates into the boiler water. The process of transmission takes place in the three distinct modes shown in the sketch, each of which is governed by a definite law not applicable to the others. Before the heat reaches the body of the boiler water it changes its mode of travel at least twice. It is first imparted to the dry surface of the heating plate in two ways: (*a*) By radiation from the hot fuel bed, the furnace walls, and the luminous flames, and (*b*) by convection from the hot, moving gaseous products of combustion. When the heat reaches the dry surface it changes its mode of travel and passes through the soot, metal and scale to the wet surface purely by conduction. From the wet surface of the plate the heat is carried into the boiler water mainly by convection.

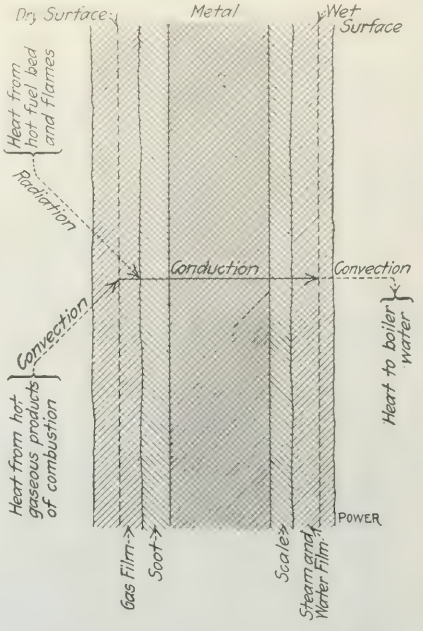
It may be well to define and explain briefly these modes of heat travel and the dry and the wet surfaces of the heating plate.

Radiation

Radiation is that mode of heat propagation by which heat passes through space from one body to another without any material agency; the two bodies do not come into contact directly nor does a third body or other bodies transmit the heat. In the case of a furnace and the heating plate of a boiler, the heat from the hot fuel bed and furnace walls flows by radiation through the space filled with gases, without heating the gases appreciably, directly into the boiler plate. The quantity of heat transmitted by radiation would not be lessened if the gases did not fill the space; in fact, it would be slightly greater if there were a vacuum between the fuel bed and the plate.

Convection

Convection of heat from one place to another always implies the motion of a



How Heat Flows into a Boiler

fluid receiving or giving up the heat. Thus, in the case of the hot gases and the boiler, the heat is transferred to the heating plate (from the gas) by small particles of the gas moving from the body of the gas to the plate and imparting their heat to the latter by their impact upon it. In other words, convection is a process of continuous interchange of position between cooled particles (molecules) of gas next to the heat-absorbing surface and hotter particles within the body of the gas. The quantity of heat imparted to a unit area of dry surface of the plate depends on the rate at which these particles of gas exchange positions.

Conduction

Conduction of heat is the process by which heat flows from a hotter body to a colder one, when the two bodies are in contact, or from a hotter to a colder part of the same body. Conduction implies no visible nor mechanical motion of different bodies or different parts of the same body, but implies a direct contact. In a boiler the heat passes by conduction from the particles of hot gas that touch the soot coating, into the soot, and then through the metal and the scale (if there happens to be any of the latter) into the particles of water next to the scale. These particles of water absorb heat until they change into small bubbles of steam which are carried away by the circulation of the water. Since the heat in these steam bubbles is taken from the heating plate through the water into the steam space by mechanical, or visible motion, it is said to travel by convection.

The particles of gas in contact with the soot coating of the tubes may form a gas film of appreciable thickness; then the heat near the outside of the film has to travel by conduction through part of the thickness of the film a longer distance than does the heat near to the soot coating. Thus the boundary along which convection ceases and conduction starts is not a well defined surface, but rather a gas layer of measurable thickness. In addition, gases are permeable to radiation, so that the heat traveling by radiation penetrates the gas film and enters the soot coating almost directly. In consequence, the dry surface of the heating plate may be defined as the thin layer of gas near, or at, the outside surface of the soot coating, where the heat ceases to travel by convection and starts to travel by conduction. For similar reasons the wet surface of the heating plate may be defined as the thin layer of steam and water, near or at the surface of the layer of scale away from the metal plate, where the heat ceases to travel by conduction and starts to travel by convection facilitated by the circulation of the water in the boiler.

The smallest central station in the United States is said to be in Sacramento, Neb., which has a grand total population of 12. However, Sacramento is progressive, as evidenced by the fact that it already has a central station. E. G. Anderson, proprietor of this embryo lighting company, operates a 3-hp. Fairbanks-Morse coal-oil engine having a total connected load of 30 tantalum lamps.—*Electrical World.*

The United States Navy's coaling station at Bradford, R. I., will shortly become one of the Navy's largest fuel-oil supply stations when a new tank, foundations for which are now being laid, is completed. It will have a capacity of 750,000 gal., which will give the station a total storage capacity of more than 1,500,000 gal. The oil will be mostly for the use of the torpedo-boat destroyers.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

Counter Electromotive Force

As explained in a lesson on the dynamo, the passage of a wire across a magnetic field causes an electromotive force to be generated in the wire. This is true of the wires on a motor armature as well as those of a dynamo. The electromotive force induced in the wires of a motor armature is in the opposite direction to the flow of the current which is supplied to the motor to drive it; the induced pressure is therefore called "counter electromotive force."

The counter electromotive force of a motor armature tends to prevent the passage of current through the windings, but it can never rise high enough to do so because that would deprive the armature of the power which drives it and thereby causes the generation of the counter electromotive force. The latter, however, is almost as great as the impressed electromotive force which forces the current through the armature; the difference between the two is the "drop" in the armature and its connections due to their electrical resistance.

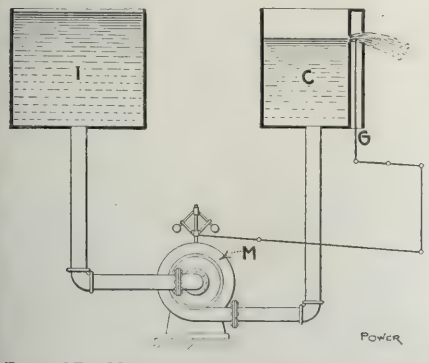


Fig. 107. Mechanical Analogue of Relations between Armature Electromotive Forces and Speed

For example, if the complete armature resistance of a 110-volt motor were ⅓ ohm and the armature current were 15 amperes, the drop due to resistance would be

$$\frac{1}{3} \times 15 = 5 \text{ volts}$$

The difference between this and the supply voltage would be the counter electromotive force, which, in the case assumed, would be 105 volts.

Fig. 107 illustrates crudely the principle of impressed and counter electromotive forces and motor speed. A water motor M is supplied from a tank I which is kept filled from an outside source. The discharge from the motor is delivered to another tank C and the level in this tank is supposed to be controlled by a gate G, the position of which is controlled by a flyball governor driven by the motor. The gate and its mechanism are supposed to be frictionless. When the motor is stationary, there is no water in the counter-pressure tank C, the gate being at its lowest position, opening the whole side of the tank. Therefore, when the water is first turned on, the motor is subjected to the full head of the tank I.

Suppose the motor to be running at normal speed and the main water level to be 110 in. above the outlet; if the resistance of the water passages be such as to require a head of 5 in. to force water through the system at such a rate as to keep the motor speed normal, then the governor would adjust the gate G to a level 105 in. above the motor outlet, giving the required difference of 5 in. between the impressed head in the tank I and the counter head in the tank C. Should the load on the motor be increased, the motor would slow down, thereby lowering the gate G and reducing the counter head until the difference between the two heads was sufficient to force the required water through. This is almost a complete mechanical counterpart of the behavior of an electric motor under the opposing influences of its counter electromotive force and the impressed electromotive force.

The counter electromotive force of a motor armature depends on the quantity of magnetic flux that passes between the field magnet and the armature core, the number of wires around the armature core, the way the wires are connected together and the speed at which the armature revolves. The way the wires are connected together is taken care of by stating the number of paths through the winding. Expressed as a formula:

$$\frac{Flux \times wires \times r.p.m.}{60 \times paths \times 100{,}000{,}000} = counter\ e.m.f.$$

In this formula, "flux" means the total magnetic lines of force passing between the pole faces and the armature core.

Speed

The counter electromotive force of a motor armature varies exactly with the speed, other things being fixed, but this is putting the cart before the horse in considering motors. The practical viewpoint is that the speed depends on how much counter electromotive force the armature is allowed to generate. Transposing the preceding formula will show that relation, thus:

$$\frac{6{,}000{,}000{,}000 \times paths \times counter\ e.m.f.}{Flux \times wires} = r.p.m.$$

The counter electromotive force being equal to the difference between the impressed electromotive force and the resistance drop in the armature, it can be easily calculated if the armature current and the resistance of the armature circuit be known: Thus:

Impressed electromotive force — (resistance × current) = Counter electromotive force

From this last formula it will be evident that the counter electromotive force of a motor cannot be constant unless the impressed electromotive force and the armature current are constant or else are varied oppositely (it being assumed that the armature resistance is constant, which is almost always approximately true). However, the "drop" in the armature circuit is usually small as compared with the impressed electromotive force; consequently, the speed of a "constant-potential" motor does not vary much if the impressed electromotive force is maintained steady.

For example, suppose that a 110-volt motor had an armature resistance of ⅓ ohm, its full load armature current were 15 amperes and the current at no load 1½ amperes. At full load the drop would be 5 volts, leaving 105 volts for the counter electromotive force to balance, and at no load it would be ½ volt, leaving 109½ volts to be balanced by counter electromotive force. If the speed at full load were 1500 r.p.m., the speed at no load would be 1564 r.p.m., assuming no change in the magnetic flux maintained by the field magnet.

Reference to the speed formula will show that changes in counter electromotive force can be compensated by making proportionate opposite changes in the field flux. For example, in the case just assumed, if the field flux at full load were 2,100,000 maxwells (lines of force), increasing this to 2,190,000 maxwells at no load would exactly compensate the change in counter electromotive force and maintain the speed constant. Perhaps working through the whole problem will make this clearer. Assume, therefore, the following values:

	No load	Full load
Armature wires	400	400
Armature paths	2	2
Armature resistance (ohms)	$\frac{1}{3}$	$\frac{1}{3}$
Armature current (amp.)	$1\frac{1}{2}$	15
Armature drop (res. × current)	$\frac{1}{2}$	5
Impressed e.m.f.	110	110
Counter e.m.f.	$109\frac{1}{2}$	105
Normal magnetic flux (maxwells)	2,100,000	2,100,000

Substituting these values in the formula for speed gives the following comparison:

$$R.p.m.\ at\ full\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 105}{2{,}100{,}000 \times 400} = 1500$$

$$R.p.m.\ at\ no\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 109\frac{1}{2}}{2{,}100{,}000 \times 400} = 1564$$

Increasing the flux at no load to 2,190,000 maxwells:

$$R.p.m.\ at\ no\ load = \frac{6{,}000{,}000{,}000 \times 2 \times 109\frac{1}{2}}{2{,}190{,}000 \times 400} = 1500$$

In special cases, the field flux is varied automatically with variations in the armature current of a constant-potential motor in order to keep the speed constant. The large majority of such machines, however, are operated with a steady magnetic field, because the speed variation is not great enough to make it worth while to complicate the construction of the machine in order to avoid it.

Power

The mechanical power of an electric motor is commonly expressed in the familiar terms of "horsepower"; one horsepower being the equivalent of 33,000 foot-pounds expended per minute, as in all other machines. One horsepower is also equivalent to 746 watts* as explained in an early lesson on units. The mechanical power delivered by a motor is equal to the power which it receives from the supply circuit, minus the power absorbed by the machine itself. Expressed differently, the delivered power is equal to the applied power multiplied by the efficiency of the machine. Thus:

$$Applied\ power \times efficiency = Delivered\ power$$

The power must, of course, be expressed in the same units both times, and the efficiency must be written as a fraction or a decimal. For example, if the applied power be 1865 watts and the efficiency 80 per cent., the delivered power will be

$$1865 \times 0.80 = 1492\ watts$$

and this is equivalent to

$$\frac{1492}{746} = 2\ hp.$$

There is also a relation between torque and speed and horsepower. Multiplying the torque by the number of revolutions per minute and by 2π (6.2832) gives the foot-pounds of work done per minute; dividing this by 33,000 gives the horsepower. Thus:

$$\frac{6.2832 \times r.p.m. \times torque}{33{,}000} = hp.$$

This can be proved by going over the data of an actual machine and checking through the electrical and mechanical figures. These figures are as follows:

Armature diameter (inches)	4
Full load r.p.m.	1,500
Full load armature current (amp.)	15
Number of armature paths	2
Current in each wire (amp.)	$7\frac{1}{2}$
Number of armature wires	400
Total magnetic flux, maxwells	2,100,000
Delivered horsepower	2
Power absorbed in driving armature (hp.)	0.11
Total full-load hp. developed	2.11

According to the formula previously given, the torque was

$$\frac{2{,}100{,}000 \times 400 \times 7\frac{1}{2}}{852{,}230{,}000} = 7.39\ lb.\text{-}ft.$$

Transposing the last formula:

$$\frac{33{,}000 \times hp.}{6.2832 \times r.p.m.} = torque$$

and filling in the known values for horsepower and speed:

$$\frac{33{,}000 \times 2.11}{6.2832 \times 1500} = 7.39\ lb.\text{-}ft.$$

The interested student will find it worth while to check back even farther, verifying the figures for horsepower. The input of the armature was

110 *volts* × 15 *amperes* = 1650 *watts*

The resistance of the armature circuit was ⅓ ohm and the drop, therefore, was ⅓ × 15 = 5 volts. This makes the power lost in overcoming the electrical resistance 5 × 15 = 75 watts, leaving 1650 — 75 = 1575 watts for driving the armature and doing useful work at the pulley. The power required to turn the armature at 1500 r.p.m. was found by test to be 83 watts. Subtracting this from the 1575 watts available leaves 1492 watts for work outside the motor; 1492 watts equals 2 hp.

The mechanical power developed by a motor armature, when expressed in watts, is equal to the counter electromotive force multiplied by the armature current. This includes the power wasted by friction and core losses as well as the useful power delivered at the pulley. In the case just considered, for example, the counter electromotive force was 105 volts and the armature current 15 amperes; the product of these (105 × 15) is 1575 watts, which has just been shown to cover 83 watts lost by friction and armature core effects and 1492 watts delivered as useful work.

Efficiency

There are several kinds of efficiency, but unless some particular kind is named, the efficiency of a motor is commonly intended to mean the ratio of output to input, thus:

$$\frac{Output}{Input} = efficiency$$

The output, as previously stated, is equal to the input minus the losses in the machine. Since the input and losses are almost always measured or computed in watts or kilowatts, it is advisable, when discussing efficiency, to express the output in watts, to avoid confusion. This is done by multiplying the output in horsepower by 746.

There are four sources of loss in a motor: the resistance of the field winding, the resistance of the complete armature circuit, the friction of the moving surfaces (the shaft in its bearings, the commutator under its brushes and the armature surfaces sweeping through the air) and hysteresis and eddy currents in the armature core. These were all fully explained in a previous lesson on the dynamo. Expressing the losses in watts, they are simply added together to get the total losses for subtraction from the input.

Taking as an example the motor previously discussed, the following values are obtained:

Full load conditions	Amperes	Volts	Watts
Armature winding	15	5	75
Field winding	1.95	110	215
Armature core and friction			83
Total losses			373

The input from the line was 16.95 amperes at 110 volts, making 1865 watts. Subtracting 373 from this leaves 1492

TABLE 6. AMPERES PER HORSEPOWER

Efficiency %	110 V.	115 V.	120 V.	220 V.	230 V.	240 V.	250 V.	500 V.	550 V.	575 V.
75	9.04	8.65	8.29	4.52	4.32	4.15	3.98	1.99	1.81	1.73
76	8.92	8.54	8.18	4.46	4.27	4.09	3.92	1.96	1.78	1.71
77	8.81	8.42	8.08	4.40	4.21	4.04	3.88	1.94	1.76	1.68
78	8.70	8.32	7.97	4.35	4.16	3.99	3.82	1.91	1.74	1.66
79	8.58	8.21	7.87	4.29	4.11	3.93	3.78	1.89	1.72	1.64
80	8.48	8.11	7.77	4.24	4.05	3.89	3.74	1.87	1.70	1.62
81	8.37	8.01	7.67	4.19	4.00	3.84	3.68	1.84	1.67	1.60
82	8.27	7.91	7.58	4.14	3.96	3.79	3.64	1.82	1.65	1.58
83	8.17	7.82	7.49	4.08	3.91	3.74	3.60	1.80	1.63	1.56
84	8.08	7.72	7.40	4.04	3.86	3.70	3.56	1.78	1.61	1.54
85	7.98	7.63	7.31	3.99	3.82	3.66	3.52	1.76	1.60	1.53
86	7.89	7.54	7.23	3.94	3.77	3.61	3.48	1.74	1.58	1.51
87	7.79	7.46	7.15	3.90	3.73	3.57	3.44	1.72	1.56	1.49
88	7.71	7.37	7.06	3.85	3.69	3.53	3.40	1.70	1.54	1.47
89	7.62	7.29	6.99	3.81	3.64	3.49	3.36	1.68	1.52	1.46
90	7.54	7.21	6.91	3.77	3.60	3.45	3.32	1.66	1.51	1.44
91	7.45	7.13	6.83	3.73	3.56	3.42	3.28	1.64	1.49	1.43
92	7.37	7.05	6.76	3.69	3.53	3.38	3.24	1.62	1.47	1.41
93	7.29	6.97	6.68	3.65	3.49	3.34	3.20	1.60	1.46	1.39
94	7.22	6.90	6.61	3.61	3.45	3.31	3.18	1.59	1.44	1.38
95	7.14	6.83	6.54	3.57	3.41	3.27	3.14	1.57	1.43	1.37

*The exact equivalent is 745.65 watts, but 746 is commonly used.

watts for useful work. The efficiency, therefore, was

$$\frac{1492}{1865} = 0.80$$

or 80 per cent.

As motors are operated on circuits of standard voltages, it is usual to take the input in amperes instead of watts, in considering a motor's performance. Table 6 gives the number of amperes required at each of the standard voltages to enable a motor to deliver one horsepower, at different efficiencies ranging from 75 to 95 per cent. The table will be found convenient for ascertaining how much current a given motor will require, the efficiency being known or estimated, and also for ascertaining the efficiency of a motor of a given rating in amperes and horsepower.

Example I. What is the full-load current of a 20-hp. motor of 88 per cent. efficiency, built to run on 240 volts?

Starting at 88 in the efficiency column of the table and following that line over to the 240-volt column, the number found there is 3.53 (amperes per horsepower). Multiplying this by 20 gives 70.6 amperes as the full-load current.

Example II. A 220-volt motor has a nameplate stamped "15 hp." and "60 amperes." What is its efficiency?

If the current at 15 hp. is 60 amperes, the current per horsepower will be

$$\frac{60}{15} = 4 \text{ amperes}$$

In the 220-volt column, the number nearest to 4 is 3.99 and in the line with this, in the efficiency column, is 85; the efficiency at rated load, therefore, is 85 per cent.

Running 60-Cycle Alternator at 50 Cycles

In answer to an inquiry by one of our readers as to how much is lost in the efficiency of a 2300-volt alternator if designed for 60 cycles but run at 50 cycles, we offer the following comment:

The efficiency of a generator is the ratio of the power delivered at the terminals to this amount plus the internal losses. Thus, if the output in watts is designated by P, the armature copper loss by L_A, the field copper loss by L_F the hysteresis loss by L_H, the eddy current loss by L_E, and the loss due to friction and air resistance by L_W, the efficiency can be expressed as follows:

$$\text{Efficiency} = \frac{P}{P + L_A + L_F + L_H + L_E + L_W}$$

When an alternator designed for a frequency of 60 cycles, is operated at 50 cycles, several quantities in the foregoing formula are increased. A fundamental principle in generator design is that the voltage depends upon the density of field flux, the speed of the moving element and the number of conductors in series on the armature. It therefore follows that in lowering the speed such that the frequency is reduced from 60 to 50 cycles, the voltage, if maintained constant, must be upheld by strengthening the field flux. This necessitates a heavier current in the field winding, with a consequent increase in the field copper loss. The stronger field also increases the iron losses in the armature and field frame. The variation in the friction and air resistance will be so small as to be practically negligible. If the increased values of these losses are substituted in the formula, it will be seen that the efficiency will be materially reduced.

In order to accurately determine the efficiency mathematically, for a 60-cycle alternator when operated at a frequency of 50 cycles, detailed information relative to the design of the machine must first be secured. The diameter of the air space must be known, together with the number and size of the field poles, the dimensions and construction of the armature core, the qualities of the iron used, the type of armature winding, the excitation voltage and many other features, all of which are involved in the calculations of the various copper, iron and mechanical losses. None of these quantities are stated in the question; hence owing to the wide variety in alternator sizes, designs and characteristics, and the consequent practical impossibility of citing any efficiency values at various frequencies which would accurately apply to alternators in general, no reliable definite value for the desired loss can be given.

CORRESPONDENCE

What Reversed the Motor?

In reply to the question, appearing in POWER for Oct. 29, "What Reversed the Motor?" I wish to submit the following:

The motor reversed its direction of rotation because its field magnetism was reversed by inserting the lamp in place of the resistance coil in the field circuit.

Assuming the ¼-hp. motor to have an efficiency of 60 per cent., the normal watts consumed per hour would be

$$\frac{746}{4} \times \frac{100}{60} = 311 \text{ watts}$$

With the motor operating as a series machine under full load the current in the field coils would be

$$\frac{311}{110} = 2.83 \text{ amp.}$$

Thus 2.83 amp. is the normal magnetizing current. When the 50-watt lamp was inserted in place of the resistance coil the current in the field winding was reduced to $\frac{50}{110} = 0.45$ amp. or less, as the voltage across the lamp was reduced because of the resistance drop in the field coils. In this way the lamp reduced the field current to less than ⅛ its normal value. Because the magnetism in the field poles increases very slowly until a certain current value is reached, the magnetic field produced was very weak.

The armature winding of the motor may be divided into two parts, one consisting of coils which produce a cross-magnetizing field only and the other of coils which produce a demagnetizing field only. The more the brushes are shifted from the axis of symmetry the greater the proportion of armature coils that tend to produce a demagnetizing field. This demagnetizing effect of the armature coils on the field destroyed the magnetism of the fields and magnetized them in the opposite direction.

Thus when either the field magnetism or the armature current are reversed in a motor its direction of rotation is reversed.

AMOS L. UTZ.

St. Joseph, Mo.

Mr. Boone's trouble is very similar to trouble I experienced with a 15-hp., 112-volt, shunt-wound motor, driving a circular saw. The motor was about 2000 ft. from the generator and received its current over a No. 6 B. & S. wire. On light load it would operate satisfactorily; but with a heavy load the motor would stop, and start up in the reverse direction running a little above normal speed, with no sparking at the brushes.

A voltmeter showed 20 volts across the motor terminals when the armature was stalled, and would increase a few volts when the armature was running backward. By shifting the brushes forward slightly the machine would start in the proper direction, after stalling. I came to the conclusion that the polarity of the field was completely reversed by the effect of the armature current, which would account for the reversed rotation; also that the motor could not reverse its field again, since any increase of voltage at the field terminals would weaken the field caused by the armature current, which would increase the armature current, and drop in the line, lowering the voltage at the motor terminals again. I think Mr. Boone can trace his trouble to a weak field. By placing a lamp in series, he weakened the field much below normal, causing the armature current to be above normal, with a corresponding increase in the armature reaction, and with the large current in starting the already weak field was reversed.

ORAL CHANDLER.

Ft. Mansfield, R. I.

The first coal mine worked in the Southern States was opened in the Richmond basin, Virginia, in 1822. Soon after, mines were opened in Maryland, Kentucky, Alabama, Tennessee and Georgia.—*The Tradesman.*

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

A System of Gas Engine Governing *

By George S. Cooper

Among the problems that have arisen in connection with the widespread adoption of the internal-combustion engine, few have led to a wider variety of design than the governing problem.

As compared with steam-engine governing, the number of possible methods is much greater. The steam engine operates on a medium of practically unvarying characteristics as to pressure, temperature, and chemical composition, prepared in a separate piece of apparatus. Thus there is left to the engine's governing mechanism only the determining of the quantity of the medium to be admitted to the cylinder at each stroke. Since nearly all steam engines making any pretense to economy operate on the automatic cutoff principle, the duty of the steam-engine designer is to provide a mechanism which will cut off the admission at such a time that the mean effective pressure will be proportional to the power demand, thus holding the speed of the engine constant within 1 or 2 per cent.

In contrast to this simple requirement a much more complex function devolves upon the internal-combustion engine designer. The engine, if it be of the four-stroke-cycle type, must receive the fuel in a liquid or gaseous form, mix it thoroughly with the proper quantity of air to form a combustible mixture, admit this mixture to the engine cylinder during the suction stroke, compress it into the clearance space during the next stroke, ignite the charge and expand the products of combustion during a working stroke, and, finally, during the exhaust stroke, expel the spent gases from the cylinder, the whole cycle occupying two revolutions. In the two-stroke-cycle engine these events take place during one revolution consisting of a compression and a working stroke, the fresh charge being forced in and the exhaust gas expelled either by a separate pump, in double-acting engines, or by utilizing the crank end of the main cylinder or the crank case as a pump in single-acting engines.

As in the steam engine, the mean effective pressure must be graduated to the load driven if the speed is to be maintained constant. This may be done in a number of ways.

Where the so called "hit-and-miss" method is employed, a constant volume of mixture having a constant fuel and air ratio is admitted at each suction stroke until the speed rises above normal, when the governor acts to prevent the admission of fuel during one or more cycles until the speed drops slightly below normal; then the admission of fuel is once more resumed. This method has proved quite satisfactory for small engines, especially those of the single-cylinder, single-acting type, but for medium and large multi-cylinder machines more complete methods have been sought.

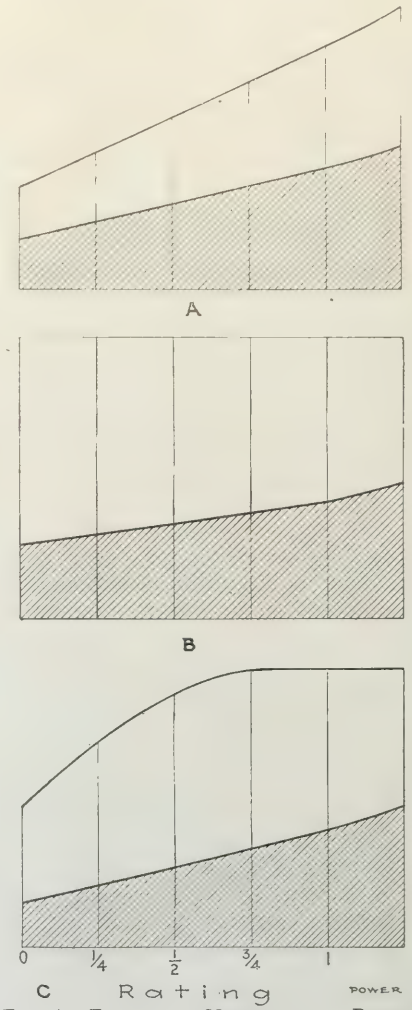


Fig. 1. Explosive Mixtures at Different Loads for Different Governing Systems

One of the most widely adopted governing methods is the "throttling" system. In this the engine receives a charge of constant fuel and air ratio at every suction stroke but the quantity of this charge is varied to suit the load on the engine by passing it through a throttling valve, the opening of which is controlled by the governor. Diagram A, Fig. 1, shows graphically how the quantity of the mixture is varied throughout the load range. The shaded portion of the diagram represents the volume of gas and the unshaded portion the volume of air used. Ordinates therefore represent the total volume of the explosive charge. Since the compression pressure is proportional to the volume of the charge admitted, it follows that the ordinates also represent the compression pressure, and their varying heights from no load to maximum show how the compression varies with the load.

The simplicity of the throttling method commends it but it is open to the objection that it involves a greatly reduced compression on light loads together with a loss of work due to wiredrawing the mixture on the suction stroke.

A third method is illustrated in diagram B. Here all loads from the lightest to the heaviest are carried by merely changing the ratio of gas to air. The total mixture volume does not vary and therefore the compression is constant throughout the load range. This method satisfies the theoretical requirements of full compression on light loads but involves the use, on light loads, of very lean mixtures which, due to the great excess of air present, burn with poor efficiency and frequently fail to ignite; consequently this method is open to much objection from the standpoint of fuel economy.

Diagram C illustrates a combination of the two latter methods which makes it possible to maintain economical compression on light loads without weakening the mixture to such an inefficient point. In this system all loads from maximum down to about 60 per cent. of the rated load are handled by simply varying the ratio of gas to air, the compression remaining constant. Loads less than about 60 per cent. of the rated capacity are handled by varying both the quality and the volume of the charge. By means of this combined action, the lightest load may be carried without impoverishing the charge to the extent involved in the plain quality method, and without reducing the compression to so low a value as in the throttling method.

The means by which this is accom-

*From a paper read before the Ohio Society of Mechanical, Electrical and Steam Engineers at Akron, Ohio, Nov. 21, 1912.

plished are shown in Fig. 2, which is a cross-section through the valves and combustion chamber of the Buckeye four-stroke-cycle gas engine. Exhaust valve *E* and the inlet valve directly above are operated by lifters and wiper cams driven from a single eccentric on the layshaft.

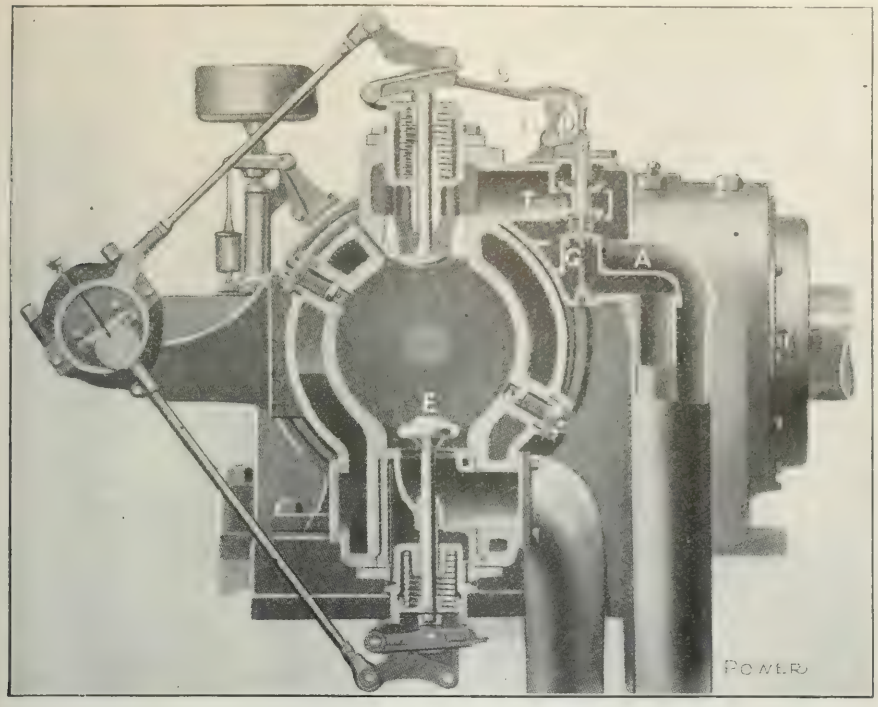


FIG. 2. CROSS-SECTION THROUGH COMBUSTION CHAMBER OF BUCKEYE FOUR-STROKE-CYCLE GAS ENGINE

Each combustion chamber of this engine is supplied with fuel by a separate mixing chamber. The gas enters the lower compartment of each mixing chamber through a small pipe inclosed within the air pipe. A gas valve *G* prevents the gas and air from mixing at all times except during the suction stroke. Valve *G* derives its motion from the main inlet-valve lifter through the segment lever *B* which fulcrums on a roller carried by lever *C*. This lever is under control of the governor with the result that, while the gas valve lifts at every suction stroke, the extent of its lift is variable, being greater at heavy loads and less on light loads. Surrounding the gas-valve stem is a tubular stem attached to the balanced throttling valve *T*. This valve is also governor-controlled through lever *D*. Note, however, that while the gas valve opens and closes at every suction stroke, the lift varying with the load, the throttling valve is held stationary as long as the load is constant and moves to a new position only when the load varies. At all loads above 60 per cent. of the rating this valve is held so high above its seat that the mixture is not throttled appreciably, the governing being done entirely by the variable lift of the gas valve. As the load falls off, the increased speed causes the governor weights to expand and move the fulcrum roller under the segment lever to a position where the gas valve will have a decreased lift, and at the same time the action of the governor reduces the opening of the throttling valve. All loads from maximum down to about 60 per cent. are handled purely by varying the mixture, the total volume of mixture and, consequently, the compression remaining unchanged during this range. At about two-thirds load the throttling valve begins to reduce the volume of mixture admitted to the cylinder so that all lighter loads are carried by weakening the charge and using less of it.

However, since the fulcrum roller has a relatively small movement between 60 per cent. and friction load, it is evident that the throttling effect strongly predominates during the lower part of the load range, that is, the quality of the charge varies but little on the lighter loads, governing being accomplished mainly by changing the volume of the charge. Since lean fuel mixtures burn more slowly than rich ones it has been found advisable to vary the ignition timing as the load on the engine varies. The governor does this automatically by advancing the ignition timer on light loads and retarding it on heavy ones.

This governing mechanism may be adjusted to plain quality governing, plain throttling governing, or any combination of the two according to the kind of gas used. With a view to determining the relative fuel economy of these three systems the writer ran a series of tests on a 150-hp. single-acting tandem engine equipped as described. Fig. 3 shows the results of these tests. The economical results obtained on maximum load are the same in all three systems, as is to be expected when it is recalled that, regardless of the system of governing employed, ultimate capacity is obtained by admitting at each suction stroke a cylinderful of mixture having that proportion of gas and air which will give the greatest mean effective pressure. The greatest differences appear on the light loads. The plain throttling system gives much better results than the quality system, while the combination method shows better economy than either of the others. Fig. 3 does not necessarily represent the best results obtainable under either system. It simply shows comparative results with the various governing methods, all other conditions being equal.

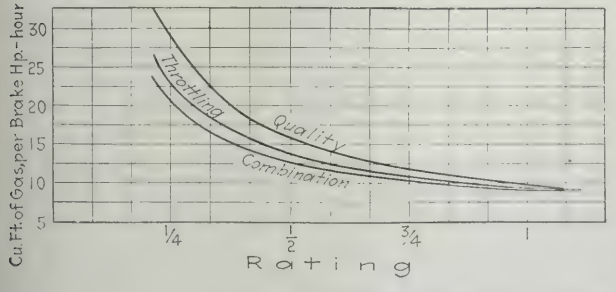


FIG. 3. RELATIVE FUEL FOR DIFFERENT GOVERNING SYSTEMS

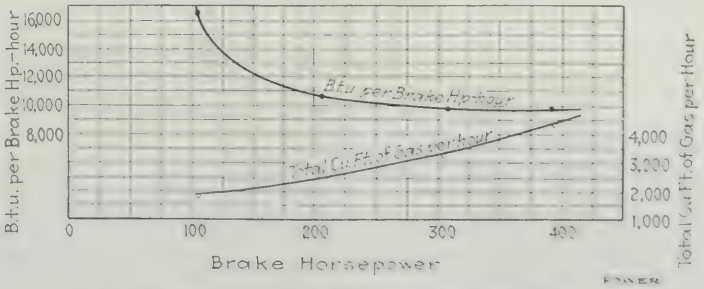


FIG. 4. ECONOMY TEST ON 400-HP. ENGINE, COMBINATION GOVERNED, USING NATURAL GAS

The combination method possesses certain operating advantages worth mentioning. First, owing to the fact that the governor controls the gas-valve lift, the quality of the explosive mixture is to an extent self-regulating; for example, should the pressure in the gas line drop, the action of the governor would

increase the lift of the valve, thus restoring the quality of the explosive mixture to normal. A further advantage is that maximum richness of mixture is employed only when the engine is working at its ultimate capacity. Over-rich fuel mixtures are wasteful because a portion of the gas is rejected unburned owing to the lack of oxygen.

The combination system, owing to its flexibility of adjustment, is readily adapted to any power gas in general use.

Natural gas and carbureted water gas have a range of explosiveness relatively small as compared with producer and blast-furnace gas. That is, the ratio of gas to air when using the first named gases must be kept nearer a constant figure than when using the last named. Therefore, engines using natural or illuminating gas are adjusted with the throttling effect predominating. When using producer and blast-furnace gas the quality may vary more widely; hence governing is accomplished by varying the gas and air ratio through the greater part of the load range and throttling is resorted to on the lightest loads only. Fig. 4 shows the results of an economy test of a 400-hp. engine, combination governed and operating on natural gas. Attention is called to the flatness of the heat-consumption curve between rated load and half load.

It is frequently stated that internal-combustion engines show their best fuel economy when operating at or near their maximum capacity. This is correct when the heat consumption is expressed in terms of net or brake horsepower; but the popular conception seems to be that this characteristic results alone from a decided falling off in combustion efficiency at all load conditions less than maximum, the conclusion being that the permissible range of compression pressures and mixture ratios is very narrow. The principal arguments favoring the "hit-and-miss" governing system are based on this hypothesis. When, however, the heat consumption is reduced to terms of indicated horsepower instead of brake horsepower, it is seen that the fuel consumption per indicated horsepower is almost constant from maximum down to about one-third load, and that the apparently high heat consumption on moderate loads is due not so much to a falling off in combustion efficiency as to the decreased mechanical efficiency resulting from the fact that engine friction does not decrease in proportion to the decreased load but remains nearly constant at all loads. For example, when the engine is working at less than about 20 per cent. of its rating, the power necessary to overcome friction is actually greater than the net delivered horsepower and, therefore, the heat consumption per brake horsepower is more than twice the consumption per indicated horsepower.

A Reversible Marine Oil Engine

By Edwin Lundgren

Recent years have brought a remarkable development of the oil engine, particularly for marine purposes, and for this class of service, especially in large sizes, reversibility is absolutely necessary. Propellers with adjustable blades, reversing gears and couplings, hydraulic and electric transmission can be used, but are really only makeshift; the large marine oil engine should be directly reversible just as the marine steam engine.

Viewing the achievements of the notable manufacturers of oil engines in Europe, it will be found that most of them have adopted the two-stroke-cycle engine for ship propulsion. Among these are Sulzer Bros. in Switzerland, Maschinenfabrik Augsburg-Nürnberg in Germany, Fiat in Italy, Atkiebolaget Diesels Motoren in Sweden and many others.

This is largely because it is easier to reverse a two-stroke-cycle engine than a four-stroke-cycle machine; the only thing that has to be different for running in the opposite direction being the time of fuel injection. The problem of reversing then is to cut off the fuel supply, stop the engine, and start it in the opposite direction by means of compressed air; then have the fuel injected at a different time. This is comparatively simple in a two-stroke-cycle engine. Still this cycle has some decided disadvantages. Admitting the mechanical simplicity attending it due to the absence of valves and valve-gear, and the lighter weight, it must be said that the fuel economy of the four-stroke cycle is not reached by the two-stroke-cycle engine. The latter will at least use 5 to 10 per cent. more fuel and more cooling water.

Of still greater importance, for a Diesel engine in particular, is the fact that it is much more difficult to keep the fuel-injection nozzle of the two-stroke-cycle engine free from carbon. This is due to the greater heat prevailing in the cylinder where there is one explosion during every revolution, whereas in the four-stroke-cycle engine during every second revolution fresh air is drawn into the cylinder and has time to cool the walls effectively. The order of events in the four-stroke-cycle is suction, compression, explosion, exhaust, suction, etc. It is evident that this cycle cannot simply be reversed, but that it is necessary to change the drive of the valve-gear or the valve-gear itself in order to obtain the proper succession of working events.

One of the best solutions of the problem which meets all requirements of reliability is a construction based on Weyland's patents and built by the Nordiska Motorverkstad in Finspong, Sweden. A general view of this engine, which is not a Diesel but of the low-pressure type, is shown in Fig. 1. The valves are actuated by a camshaft which can be shifted in axial direction to bring into play one of the two sets of cams—one for forward motion, the other for reverse direction.

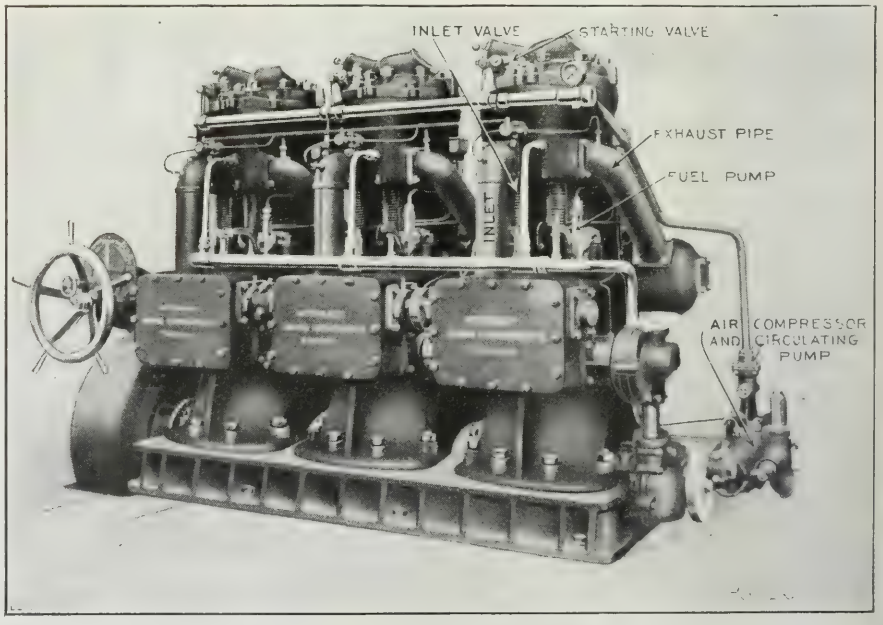


Fig. 1. General View of Engine

Besides this regular camshaft, driven by two pairs of spiral gears, there is another shaft, the so called operating shaft, which carries a number of cams, one each for the inlet, exhaust, fuel-injection and starting valves. The valve stem of each valve itself is lifted by means of a lever, the stationary fulcrum of which is formed by the edge pressing against the corresponding cam on the operating shaft by a spring; see Fig. 2.

It is apparent that by turning the op-

erating shaft with the cams, this edge is lowered or raised, thereby bringing the roller on the valve lever more or less within reach of the cam on the camshaft.

Referring to Fig. 2, *A* represents the fuel cam, *B* the starting-valve cam and *C* the exhaust cam, the lines of the inlet cam coinciding partly with those of the exhaust cam and partly with those of the fuel cam. For starting the engine the operating shaft is turned in such a position that cam *B* points downward. When the cam has been placed in this position the fuel valve is idle, while the compressed-air valve is actuated. As soon as the engine has gained sufficient momentum, the operating shaft is turned so that the lever for the fuel injection is pressed down against the camshaft bringing the fuel pump into action, while at the same time the lever for the air valve is allowed to recede toward the center of the operating shaft and thus is put out of action.

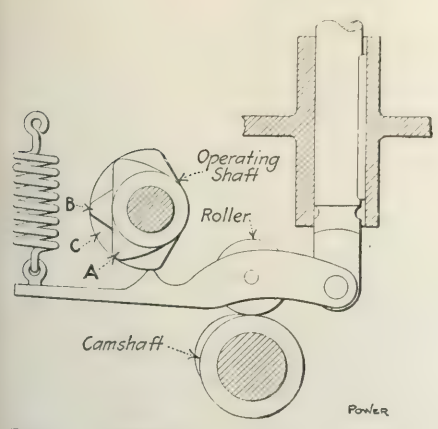


FIG. 2. OPERATING CAMS AND CAMSHAFT

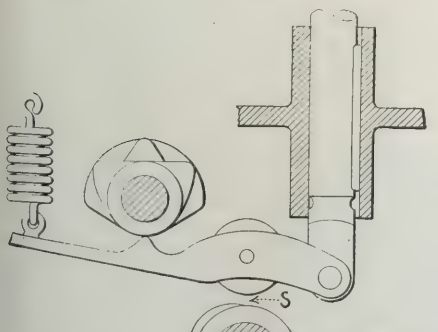


FIG. 3. NEUTRAL POSITION OF OPERATING SHAFT

The engine is then running in the direction wanted, and also at the speed wanted, as the lift of the fuel pump can be varied as desired. To reverse the engine the operating shaft has first to be turned into the position indicated in Fig. 3, where all the valve levers are clear of the cams on the camshaft and consequently all the valves remain closed. The camshaft is now shifted axially so that the cams for reverse direction are placed under the valve levers.

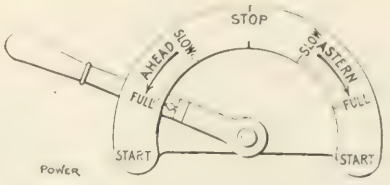


FIG. 4. CONTROL LEVER

The shifting motion is also derived from the turning of the operating shaft by means of an arm connecting the two shafts with bushed bearings on either shaft and provided with pins, one guided in a circular groove on the camshaft, the other in a cam-shaped groove on the operating shaft. The operating shaft is then turned further so that the engine is given a start by compressed air in the reverse direction. After a few revolutions the starting valves are cut out and the fuel pumps are put into action in precisely the same manner as described for running in the other direction.

All these operations are performed by throwing a single lever which actuates the operating shaft through bevel gears. Also the speed is under absolute control of this one lever. Fig. 4 represents the dial for different positions of the lever. The reversing can be done in a few seconds, and very little compressed air at 60 lb. pressure is required.

There are a number of ships equipped with these engines, particularly in Russian waters which give entire satisfaction. This reversing principle has also been applied to engines of the strictly Diesel type.

The Gas Engine as an Economical Power Producer*

Gas engines have been very greatly improved in design and construction during the last few years and undoubtedly offer the most economical means of producing power. The heat consumption of large gas engines in practical working is about 10,000 B.t.u. at normal full load and about 9500 B.t.u. at the maximum overload. The net heat value of blast-furnace gas varies with the duty of the furnace and the character of the fuel used, but in general varies between 90 and 110 B.t.u., or, say, 100 B.t.u. per cu.ft. as an average. Therefore, the consumption of blast-furnace gas per brake horsepower per hour would be approximately 100 cu.ft.

With coke-oven gas the heat consumption is the same. The heat value of coke-oven gas varies considerably, but is generally in the neighborhood of 450 to 500 B.t.u., occasionally less. Taking 450 B.t.u. as the usual figure, the consumption per brake horsepower per hour at normal full load is about 22½ cu.ft.

The exhaust gases from gas engines can also be used for generating steam, and when an engine is developing something like its full load, from 2 to 2¼ lb. of steam per brake horsepower at 60 lb. pressure is regularly generated, and this steam can be utilized for driving auxiliaries, heating and other purposes.

As regards the power available from blast-furnace and coke-oven plants, the following figures are interesting:

The calorific value and volume of gases evolved by a blast furnace depend upon the character of the furnace burden, and to some extent upon the method of driving, but, as an average figure (Northeast Coast practice) the gas evolved per ton of pig iron produced is about 160,000 cu.ft., measured at atmospheric temperature and pressure. Of this gas, about one-third is used by the ovens, about one-eighth by the blowing engines (if driven by gas engines), and about 10 per cent. is lost or used up in miscellaneous ways; thus about 45 per cent. of gas is available as surplus, or approximately 72,000 cu.ft.

Taking an average heat value of 100 B.t.u. per cu.ft., the horsepower developed by large gas engines would amount to about 30 brake horsepower for every ton smelted in 24 hr. If ordinary steam-blowing engines were already installed, the available surplus would drop to about 25 brake horsepower per ton of pig in 24 hr., and might even fall below this value.

With coke ovens the production of gas naturally varies with the quality of the coal, but an average figure is 10,000 cu.ft. of gas per ton of coal. The surplus gas, when regenerative ovens are employed, amounts to about 5000 cu.ft. per ton of coal. Where nonregenerative ovens are employed, the surplus is very much less, sometimes amounting to 2500 cu.ft. In these cases the high temperature of the escaping gases enables a good deal of steam to be evaporated by suitable boilers heated by the waste gases. Roughly speaking, about 1 to 1¼ lb. of steam will be generated for each pound of coal coked, and this steam may be used for driving steam turbines or other classes of engines. It will thus be seen how much power can be obtained by utilizing what was at one time a mere waste product.

*Extracts from the presidental address by W. C. Mountain delivered at the annual meeting of the British Association of Mining Electrical Engineers, in Shieffield, England, Sept. 27, 1912.

Someone has taken the trouble to count the number of pieces in an electric fan and to measure the length of wire used in the coils. An ordinary 16-in. fan is said to contain 1372 ft. of wire and to possess more than 750 pieces of material.—*Electrical World.*

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Ventilating System for Small Schools

By Charles A. Fuller

To design a mechanical ventilating system for a small school, which will come within the requirements of the state laws, often presents a more perplexing problem than the larger schools and auditoriums. With the small school first cost is usually of vital importance and must be kept down to the limit. The space allowed for apparatus is generally restricted, and operating costs must be studied carefully.

The accompanying plans are of a four-room school in which the ventilating requirements are well within the 30 cu.ft. of air per minute per pupil, prescribed by the state code, and the cost of the installation is only slightly more than of a direct-heating system.

In each class room direct wall radiation compensates for the heat loss through the walls and windows, with no allowance for air change. Each of the four indirect stacks placed at the base of the warm-air flues is composed of 10 sections of 4-in. Vento cast-iron radiation, the sections being spaced on 4-in. centers, and each containing 10.75 sq.ft. of surface, or 107.5 sq.ft. for each stack. With the above spacing there is a free area of 0.35 sq.ft. per section and a total free area of 3.5 sq.ft.

Allowing 30 cu.ft. of air per minute per pupil and 50 pupils per room, gives 1500 cu.ft. per min. for each room. Dividing by the free area through the stack 3.5 sq.ft., gives a velocity of practically 430 ft. per min. The temperature rise at this velocity is 78 deg., that is, the incoming air will be heated from 0 to 78 deg.

Under these conditions the condensation will be 1.23 lb. per sq.ft. per hr. As the normal condensation in a direct radiator is approximately 0.3 lb. per sq.ft. per hr., 1 sq.ft. of indirect radiation in this case will be equivalent to

$$\frac{1.23}{0.3} = 4.1 \text{ sq.ft.}$$

of direct radiation. Each stack is then equivalent to

$$4.1 \times 107.5 = 440.75 \text{ sq.ft.}$$

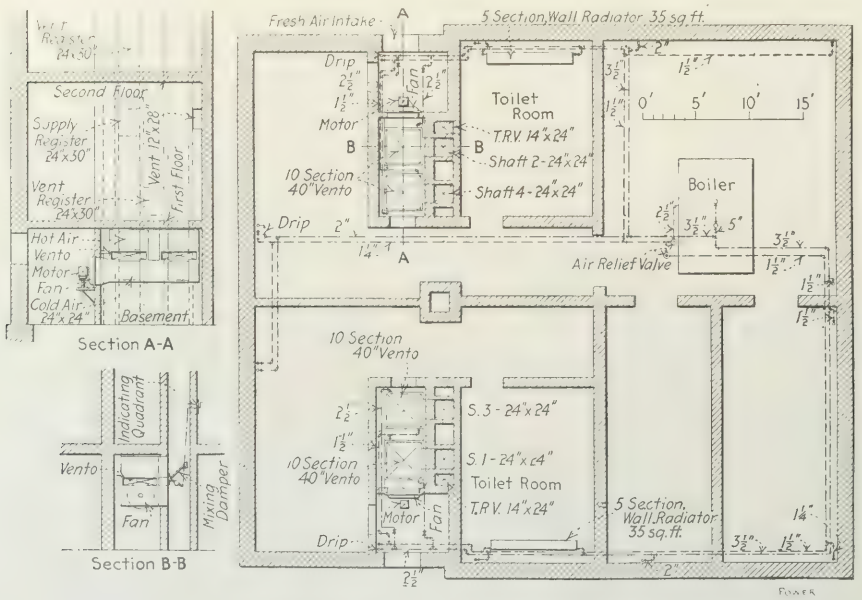


Fig. 1. Plan of Basement, Showing Boiler Connections and Indirect Stacks

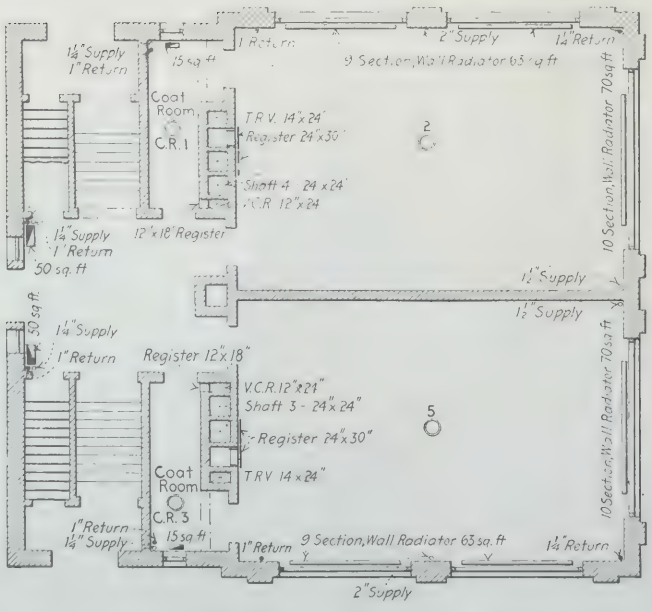


Fig. 2. First Floor Plan

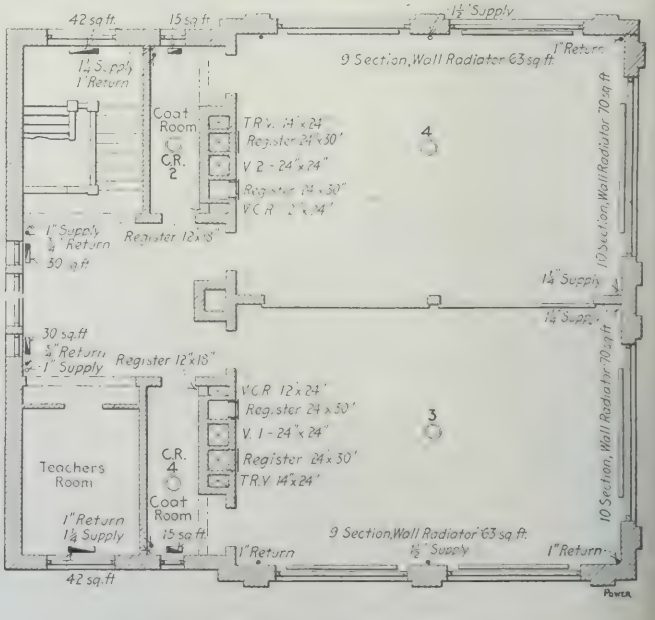


Fig. 3. Plan of Second Floor

of direct radiation and the four stacks 1763 sq.ft.

In the building there is 1158 sq.ft. of direct radiation and adding the 1763 sq.ft. for the stacks makes the equivalent of 2921 sq.ft. of direct radiation for the entire building. The boiler installed is of the cast-iron sectional type with a capacity to serve 4400 sq.ft. of radiation, an increase of 50 per cent. over the actual amount.

Two fans are installed, as shown in the basement plan. Each is of the disk type with a 36-in. wheel directly connected to a 1-hp. motor. When discharging into the atmosphere against no pressure, the fan at 600 r.p.m. is capable of delivering 10,000 cu.ft. of air per min. The resistance of the stacks and flues will cut this down by half, but even then the fan is amply large, as the maximum demand per fan is 3000 cu.ft. per min.

The horizontal ducts connecting the fans with the stacks and bypass openings could be omitted if the chamber into which the fan discharges were reasonably air-tight. This would lessen the friction of air flow and also decrease the first cost of installation slightly. The air would pass in the same manner either up through the stack to the vertical flues or through the bypass opening to the flue according to the position of the mixing damper. This damper is controlled by a chain and lever, with an indicating quadrant in the room into which the flue discharges. This is easily operated in the room to give any desired temperature of incoming air, regardless of outside temperature, if it is between 0 and 70 deg., without materially affecting the quantity of air delivered.

The supply registers are in the usual position, about 8 ft. from the floor, with the exhaust registers at the floor line. Separate exhaust registers and ducts are provided for the coat rooms and the toilet rooms in the basement. The vent flues are joined into two main flues passing up through the roof, and capped with 30-in. "Autoforce" ventilators. When toilet-room vents are carried up with the exhausts from class rooms, as in this case, care should be taken to insure that these tap into the main flue at a point considerably above the exhaust register connections from class rooms.

No air valves were installed on either the radiators or indirect stacks. All the air in the system is removed through two large automatic float air valves installed at a high point in the dry return near the boiler. The return end of each radiator and stack has an automatic impulse check valve.

The Victoria Falls & Transvaal Power Co., of South Africa, has almost completed its fourth electric power station on the Vaal River at Vereeniging. It will be one of the largest in that country.

Heating a School

By Fred L. Wagner

A few figures used in determining the size of the heating and ventilating plant in a school building may be of interest to members of the craft who are employed in such buildings. A building containing 30 classrooms will be considered. The first thing to estimate, is the probable heat losses which occur through the walls, windows and roof

Heat Losses

Assume the building to be 190 ft. long, 80 ft. wide and 55 ft. high, containing 10 classrooms on each of three floors; the playrooms are in the basement. Each classroom will average about 200 sq.ft. of glass, or 6000 sq.ft. in all. The glass in the basement will approximate 1500 sq.ft., and the area in the halls, landings and entrance doors will approximate 1500 sq.ft., making a total of 9000 sq.ft. of glass surface.

According to the dimensions given the exposed wall and roof surface in this building would amount to

$$42{,}800 - 9000 = 33{,}800 \ sq.ft.$$

The heat loss through single glass is generally taken at 1.03 B.t.u. per sq.ft. per deg. difference in temperature between the outside and inside air, and that through ordinary brick walls at 0.2. In this case an outside temperature of zero was taken and an inside temperature of 70 deg. F., making a difference of 70 deg. Through the glass the loss per hour would be

$$9000 \times 1.03 \times 70 = 648{,}900 \ B.t.u.$$

while the loss through the walls and roof would be

$$33{,}800 \times 0.2 \times 70 = 473{,}200 \ B.t.u.$$

The total loss amounts to 1,122,100 B.t.u. per hr., which will have to be supplied by the heating plant to maintain a temperature of 70 deg.

Heat for Ventilation

The foregoing figures do not take into consideration the heat required for ventilation, which in schools is greater. In this school there are 48 pupils in a classroom, making a total of 1440, and 31 adults. An allowance of 30 cu.ft. of air per min. is required for each pupil and 50 cu.ft. per min. for each adult, making a total of 44,750 cu.ft. Increasing this amount one-fourth for hallways and basement gives 55,937, or in round numbers 56,000 cu.ft. per min.

Under ordinary pressures and temperatures 1 B.t.u. will heat 55 cu.ft. of air 1 deg. To find the number of heat units needed to heat the 56,000 cu.ft. of air from 0 to 70 deg., divide by 55 and multiply by 70;

$$\frac{56{,}000}{55} \times 70 = 71{,}272.7 \ B.t.u. \ per \ min.$$

or 4,276,362 B.t.u. per hr., which is greatly in excess of the actual heat loss in the building.

Surface of Coils

The amount of radiation needed in the blast coils will next be considered. Owing to losses of heat through ducts, walls and glass and infiltration of cold air, the air coming from the coils must be at a much higher temperature than 70 deg. In zero weather when all the coils are cut in, the temperature of the air leaving the coils often reaches 120 deg. and above, and in the present case 120 deg. was taken.

The amount of air needed for ventilation is 56,000 cu.ft. per min., and a fair average velocity for the air entering the heater is 1200 ft. per min. At zero temperature and with steam in the coils at 5 lb. pressure, it will take 20 rows of 1-in. pipe to heat the air to 120 deg.

Authorities on heating and ventilating agree that 1 sq.ft. of surface in blast coils with the air passing over it at the above stated velocity will heat 13 cu.ft. of air per min. from 0 to 120 deg. Using the figures at hand, 4307 sq.ft. of coil surface will be required. As school houses are heated only during the day, provision must be made for rapid reheating; therefore one-quarter will be added to the theoretical amount, which will increase the coil surface to 5384, or practically 5400 sq.ft. The number of sections and general dimensions of the heater may be determined from data given by the maker of the particular type chosen.

Size of Blowers

To obtain as nearly as possible an even distribution of air and heat, two fans are installed in a building of this size, the capacity of each fan being one-half of the quantity of air needed. There will be needed two fans, each having a capacity of 28,000 cu.ft. per min. To avoid noise the fans must be run at low speed, and to keep down the area of the ducts an air pressure of ¾ oz. would be used. This installation would require fans with wheels about 7x3½ ft. For driving each fan a 15-hp. engine or motor would be about right.

Boiler Capacity

The boilers installed to furnish the steam for the coils and auxiliaries should be of ample capacity to furnish the maximum amount of steam needed during the severest weather. In this case the boilers were rated on the supposition that they must supply 2 lb. of steam for every square foot of coil surface per hour. The 5400 sq.ft. of coil surface at the above allowance will condense 10,800 lb. of steam per hour. Dividing by 34½ lb., the evaporation per horsepower per hour from and at 212 deg., it will be found that the coils will absorb 314 hp.

per hr. during the coldest weather. The feed pump and vacuum pump will require approximately 10 hp., and the fan engines 30, making a total of 354 hp. in boiler capacity. Three return-tubular boilers of 115 hp. each will be of ample capacity to supply all the steam needed in this particular building. This boiler capacity will not only heat the air, but cover the losses through walls and windows.

[For a building of the dimensions given, the boiler capacity appears to be excessive. Rather than base the horsepower on the condensation in the coil surface it would have been better to rate the boilers on the heat required to supply the loss through walls and windows and to raise the air for ventilation from 0 to 70 deg. F. Adding the heat units required in each case, dividing by 1000 to obtain the pounds of steam and then dividing by 34½, will give the horsepower needed for normal operation.

$$\frac{1{,}122{,}100 + 4{,}276{,}362}{1000 \times 34.5} = 156.48 hp.$$

All exhaust from pumps and fan engines can be used in the heating system, but allowing a little extra capacity for rapid heating, 200 hp. of boilers should be sufficient.—EDITOR.]

Extravagant Steam Pipes

BY W. H. BOOTH

When an engine shows an initial pressure too far below boiler pressure, and its valves are properly set with a reasonable degree of lead, either the steam pipe is too small or the flow of steam is otherwise obstructed. It may be that a stop valve is small or that the engine ports open insufficiently. But in steam work, pipes are often too large or too small.

When a line supplies steam for heating only, and no power, it is a serious fault to make a pipe line so large that the steam pressure does not drop considerably at its extreme end, when perhaps only one or two radiators require supply. In a correct system of steam heating the condensed steam should gravitate back to the boiler room. Certainty of operation and flow of steam is guaranteed in a pumped return. There should be no need for traps. So long as steam can reach the most distant point of its use the pipe from the boiler need be no larger than will just supply this steam on the days of maximum demand.

In a complaint recently investigated where there was alleged excess of steam used, it was found that the steam main from a pair of boilers supplied no steam for power purposes. All boiler makers fit their boilers with steam outlet mounting blocks big enough to fit steam pipes which shall supply steam to an engine, within a reasonable distance, without loss of pressure of serious moment. But this does not demand that the steam main shall be of the size indicated if heating steam alone is wanted.

In this case the steam outlets on the boilers were 6 in. in diameter and the steam pipe was 6 in. for some distance, beyond which it branched off 4 in. each way. A great length of 4-in. main was laid underground in a brick culvert which was inaccessible. This had been full of water at sometime while the pipe had been carrying steam and the condition of the insulation may be guessed, for it could not be seen except at a few manholes, and there it was practically detached. The boiler pressure was 50 lb. and the first thing tried before making suggestions was to close all live outlets from the pipe and measure the pipe drainage all of which returned to one tank in the ground.

This test showed that the pipe condensation was the greater half of the mean consumption of feed water. The pipes were much too large, and it was advised that in place of a 4-in. main there should be a small wrought-iron pipe starting not over 2 in. and finishing at ½ in. It was quite sufficient that steam should reach the extreme end of the line at atmospheric pressure.

There were numerous similar wastes, duplicate steam pipes, a 7-in. main on another set of boilers where a 2-in. would have done the work of heating, if only a special 3-in. pipe to a noncondensing engine had been added. The exhaust from this engine would be then sufficient to have done much of the heating work. The engineer in charge knew that the conditions of the place were bad and that the steam mains were wasting half the coal, but his words were unheeded, and the writer's advice was regarded as too drastic to be followed. To cast out the big 4-in. and 7-in. mains, which had cost so much to put in was regarded as flinging away capital. All over the place water and steam were being run to waste and the management was weeping over the expense. They would spend thousands to show an economy of 5 per cent., but could see no sense in scrapping this pipe condenser half a mile long and saving 50 per cent. This is said to be a fair sample of the public institutions throughout England.

It appears to be usual to put in steam-heating mains as large as if they had a big engine to drive at their extreme end. For years one such institution ran all condensed water from pipes and heaters into the sewers and would have continued to do so, if not stopped by external pressure. Such places are built at an enormous public cost, excessive apparatus and pipes are put in and coal bills ever after form an excessive charge on the taxes. It seems hopeless to have such places put upon a sound basis. They advertise for good engineers, but when they get them, proceed to tie their hands.

Ventilating the Boiler Room

The old plant had outlived its usefulness and it was decided to put in new and larger boilers and uptodate machinery. The old chimney in the corner of the boiler room was not large enough and besides the space it occupied was badly needed for the new boilers. It was decided to build a new stack outside of the building wall. To prevent disturbance on the floors above only the lower part of the stack from the boiler-room floor to the ceiling was torn out. Holes, level with the floor above, were cut through both building walls and 12-in. I-beams resting on iron plates inserted. After putting in some cross-pieces to help support the stack, the lower portion was removed, leaving a splendid ventilating shaft.

As the plant is located in a wood-working factory, the boilers are inclosed by fire walls and naturally it would be a hot place in summer. As it is now, visitors are surprised to find at the front of the boilers the coolest spot in the plant.

There was some talk that the old chimney being used as a ventilator would affect the draft of the new stack, but this does not seem to be the case, and the draft from the new smoke-stack is all that could be expected.

R. CEDERBLOM.

Chicago, Ill.

Friction Loss in Wrought-Iron Pipe

There seems to be something wrong in Mr. Durand's chart on page 649 of the Oct. 29 number of POWER. A 10-in. pipe with a velocity of 1.5 ft. per sec. has a friction head of 0, and with a velocity of 1 ft. per sec., the friction head, according to the chart, would be negative.

WILLIAM KENT.

Montclair, N. J.

[Below the critical velocity in a pipe, the usual formulas for determining the friction head do not apply. The friction is almost negligible and to show it with any degree of accuracy would require a chart on a much larger scale than that prepared by Mr. Durand. According to the average values given in the table from which the curves were plotted, the friction head per 100 ft. of 10-in. pipe, with a velocity of 1 ft. per sec., is 0.043 ft. It is evident that the friction-head values do not hold to the same straight line below the critical velocity. In the chart under discussion this was neglected as only the extremities of the lines would be affected, and mention was made in the text of the inaccuracy of the friction values for a velocity of 1 ft. per sec. To avoid any misunderstanding, however, as in the present case, it would have been better to cut off the chart horizontally at 0.05 ft. friction head.—EDITOR.]

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POWER

Wanted: More State Boards

Only two states, Massachusetts and Ohio, seem to have taken up the subject of state control over boiler construction and inspection in genuine earnest. In both of these states the activity of the state boards has resulted not only in great benefit to public safety, but in awakening the engineering public to a proper sense of responsibility that rests with the man in charge of a steam plant.

Complaint is sometimes to be heard of unnecessary stringency in the examination of applicants for different grades of operating engineers. To test a man's intelligence and capacity it is necessary to make inquiries beyond the commonplaces of his daily routine. How else can one demonstrate his ability to cope with new questions? How else can value be placed on a man's intelligence and enterprise if not by inquiring into subjects that are more than ordinary?

The examining boards are doing a great service not only in protecting life and property, but also in uplifting the standard of deserving engineers.

To the Oiler

Some engineers have stepped directly from the boiler room into the engine room and shouldered the responsibilities of engineers; others have risen directly from oiler to engineer and assumed like responsibilities. But the man who has done the work of fireman and oiler before becoming an engineer is by far the more fortunate.

Every engineer should have had ample experience in firing boilers before he assumes charge of the plant equipment. It is indeed hard to make the average oiler who feels thoroughly competent to perform the duties of engineer realize this. He knows that firing is hard and dirty work; perhaps, too, he has been unfortunate enough to look upon the fireman as one of a lower order than himself, and so he aims to avoid the fires for fear he will lose a little of the dignity that goes with the job of oiler.

We would say to the young oiler who aspires to become a real engineer: Forget your dignity before some wise old flannel-shirted fireman pulls it out of you in a fashion that will make you feel cheap. Instead of holding aloof from the fireman, cultivate his friendship; he can tell you helpful things that are exceedingly hard to get through other channels of information. When you get on the night watch and have the bulk of your work done, do not select a dark corner to have your quiet smoke, but go clean a few fires; do just what you would do if it was up to you to run the place for the rest of the night, but be sure you do what is right. Some time when the load is heavy, if you can spare the time, clean a fire or two and find out what it is to properly clean them without "losing the steam." You cannot learn this from a book or by simply looking on.

When an engine is to be overhauled, a pump plunger packed, or when the cable from the damper regulator to the main damper breaks, leaving the damper shut during a heavy load, when such things—and they are many—happen, help to make the repair, get into the work "up to your neck" as the saying goes, for some day these things will happen in *your* plant. You will be held responsible for a shutdown and it is then that the experience gained at the expense of sweat and blistered hands will "pull you out of a hole" that the mere onlooker would be buried in.

When new apparatus comes into the plant, learn what it is, what it does and how it does it. Be "on the job" when it is first started and find out what its good and bad features are; new machines are usually "cranky" and it is well to get acquainted with their peculiarities—catalogs will not tell you.

To carry out these suggestions you must show a willingness to be of help and an eagerness to learn, and must do both tactfully or you will be offensive; hence you acquire something besides engineering skill.

There is no real short cut to becoming a good engineer, but the foregoing combined with technical study is the surest and quickest way.

Ambition First

At some time or other to every engineer comes the feeling that his ability is confined and he is filled with an impulsive desire to advance to a broader field of labor where his attainments can be more freely exercised and appreciated. This feeling may be induced by a desire for greater wealth; or, the gaining of added knowledge; or, perhaps, the combination of these for the greater power of accomplishment. But, whatever the inspiration, if the opportunity is presented and grasped at this time the realization of the desire depends on the test of the justification of self-confidence and the rapidity of progress. If, however, the opportunity be not recognized, or delay of indecision allows it to pass, the great majority readily fall back into the old ruts, the old habits of mental lethargy again dominate and the glorious vision of possibility is, perhaps forever, obscured by the ascendency of inherent laziness and lack of desire.

The condition of those who look back and realize "what might have been" had they not succumbed to the apathy of "lost opportunities" is pitiful. Constantly do they bemoan the hard luck which they consider has forced them to remain in their narrow environments, and seldom, if ever, do they admit that the advancement of their more favored brother engineer is the result of resolute, untiring effort in maintaining a fixed purpose.

It seems strange that anyone engaged in the stimulating, progressive vocation of steam engineering should ever be content to relinquish the effort of self-advancement, and the fact that so many do simply drift along can only be ascribed to hereditary habits of mental inaction so long indulged in as to easily become a potent force unless strenuously resisted.

The spectacle of the man bewailing the "lack of opportunity" would not be so pitiful were it not possible that it could be changed. The conditions demanded are simple. The methods to be used are easily explaind. There is only one factor that is repugnant, and that is the effort which, of necessity, must be made by the individual himself.

The great secret of advancement is ambition. Literally defined, it means the cultivation of "the desire of power, honor, fame and excellence." To cultivate the desire is simply to learn to concentrate one's energies on a single thing at one time. To some this is comparatively easy, almost natural, and their progress is rapid. To others, the development of the power of concentration requires an effort, and the reward will be directly proportionate to the effort expended.

Ambition does not distinguish between men or the quality of their social status. The uneducated laborer in the boiler room and the college-bred son of the owner are, insofar as the reward of properly concentrated effort is concerned, on an equality; the college education being merely a preparation for the concentration of future effort, while the field for bettering the comparative conditions is an incentive all in favor of the laborer.

Going to Extremes

POWER has published considerable relating to keeping power-plant records, and has tried to impress upon engineers the importance of knowing what the plant is doing. While the desirability of keeping plant records has not diminished, there is danger in overdoing the matter by incorporating items in the record sheet about which the chief engineer knows nothing or by not carrying the record far enough.

For instance, in one plant the kilowatt readings were taken once an hour and a load curve plotted from the readings. The highest peak shown by the load line was 1575 kilowatts, which was between 4:30 and 6:30 p.m. In reality the highest peak load was 2325 kilowatts, as shown on the switchboard instruments between the hourly reading.

It goes without saying that reading the switchboard instruments once an hour is not enough to enable the engineer or general manager to obtain accurate information as to what the generating units are doing. Fifteen-minute intervals should be the limit for reading switchboard instruments and this has been adopted by most large stations.

It is common to incorporate data items in a report sheet which have no bearing on the actual operation of the plant. For instance, one report sheet of a steam power plant contained the item "Average kw. per b.hp.," but there was no means of indicating the brake horsepower and the only way of obtaining the desired result was to use the builder's original test figure as a basis from which to make computations. If the steam end of the machine became inefficient the same brake horsepower would be used, although more steam would be used in developing the normal machine load.

An instance of making unnecessary labor for the engineer is an item reading "Tons (grs.) Coal Used," when all coal was purchased in short tons. Why it was desired to change from short to long tons in making out the record sheet is not apparent.

Items such as clerical labor, although there was no clerk; barometer reading, with no barometer; CO_2 reading, without a gas-analyzing apparatus, were also in evidence. Total pounds of water evaporated and water evaporated per pound of coal were also to be tabulated, but there was no means of ascertaining what the evaporation was other than a water meter that might at some time have been correct.

The aim should be to incorporate in power-plant record sheets only such data as bear on the plant operation. Anything else makes the report more complex and adds to the work of the engineer.

The N. A. S. E. is getting along in years. No. 1, of Boston, one of the earlier of the subordinate associations, celebrated its thirtieth anniversary on Thanksgiving Eve. Much has happened in power-plant engineering since the organization of this association. Who can predict what will happen between now and the completion of the first half century of its existence?

Speaking of a steel for boiler plate that, being of only moderate high tensile strength can be made remarkably ductile, an authority on the subject humorously said: "Why, a boiler made of such steel couldn't explode! It would simply bulge out and push you away from it."

Did you ever stop those leaks in that boiler setting? Better do so now; they are costing you unnecessary coal and labor.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Troubles of Quarry Pumps

Probably I have had my share of pump troubles, and a relation of them will no doubt prove of interest.

The equipment at the Nesbit quarry plant consisted of one compound duplex, 12x8-in. pump, one 6-in. duplex pump, one 8-in. centrifugal pump, submerged type and an 8-in. pulsometer. Except the pulsometer these were worked to their limit day and night and a breakdown meant a flooded quarry if their operation was delayed too long.

One night in January about 12 p.m. the superintendent routed me out of bed and stated that the night man had called him up by phone, saying the compound pump had stopped and that he could not start it and wanted me to hurry down to "get things going." Arriving at the pump house I found the water within a foot of the pump cylinders, and as there was not steam enough to start the pulsometer, I made a hasty examination of the pump and found the trouble to be in one of the water cylinders. Taking off the head showed that the nut and lock nut had backed off and the follower plate had turned sidewise in the cylinder; and as the piston rod came back on the return stroke it bent in a half circle, cramping the piston and stopping the pump.

Well it was up to me to get this rod straightened, so getting some oily waste, I wound it on the rod, and with a gasoline torch to help out, heated it until I could bend it back into place, using a track jack and some wooden blocks to bend it. As soon as the piston came loose and worked freely I repacked it, and after screwing on the nuts riveted the shaft slightly over the locknut to hold it until a better job could be done.

The water in the meantime had risen until it flowed into the cylinder and as it was nearly zero outside one can imagine how comfortable I felt. A day or two afterward I took out this rod and straightened it properly, drilling a hole through the locknut for a cotter pin. This held for several months when a new rod was secured.

Not long after this accident a heavy rain on top of a deep snow caused a bad flood and during the night the compound pump "threw" a valve. After shutting down the pump to replace the valve, the night man opened the high-pressure cylinder drain cocks and before he could get the pump started the water chased him out of the pump house. Another hurry call was sent for me, but by the time I arrived the water was over the pumps and two other pumps had stopped, the steam condensing too fast in the cylinders to give the required working pressure.

This had me "stumped" and I stood around not knowing what to do. Sometime later as I was standing in the engine room I happened to glance at the air compressor and it occurred to me that air would not condense like steam. Why not turn air into those pumps? Acting at once with the help of the superintendent and a laborer, we made a connection between the main air line and the steam main.

When the connections were completed, air was turned on, the two pumps that had stopped cleared themselves of water and started easily. Adding a little steam to the air kept it from freezing and the pumps were soon running at full speed. But the compound pump refused to start because the air went out of the open cylinder drain cocks too fast to allow the necessary pressure to accumulate in the cylinders. Not caring to do any fancy diving in 8 ft. of ice water, I "fished" for two hours with a bent rod before I closed three of the valves and got the pump started. The fourth valve had lost its stem and bonnet which had unscrewed and fallen off. It took three days to get the water down as the compressor could not furnish enough air to keep the pumps running up to their capacity, but as soon as the steam chests were clear of water I changed over to steam and soon had the water down to the working level.

Several times since I have had to use air but I have never delayed so long before making the change.

Another source of trouble was the breaking off of valve studs, level with the seats, in places that could not be reached to drill them out without dismantling the pump. The seats refusing to come loose unless cut out and extra seats were not always on hand. This I overcame by breaking out an arm of the seat so as to insert a double U-bolt that came up on each side of the center passing through two extra holes cut in a valve and projecting high enough to receive a spring and plate under the nuts. This allowed the valve to play as usual with very little loss of water and gave good service until they could be correctly repaired.

One night a rather peculiar breakdown happened to the centrifugal pump. This pump was driven by a 2-in. vertical shaft 28 ft. long with gears at the upper end and held up by a thrust bearing that was water cooled. The speed was 800 r.p.m. and the cooling water came through a ¼-in. pipe tapped from the discharge line. This ¼-in. pipe split near the bearing one cold night and a fine spray of water struck the shaft, forming an ice collar. The night man paid no attention to it, thinking it harmless as it only extended over about a foot of the shaft and seemed light like frost, but shortly before morning the shaft snapped in the center of the ice ring with a clean break that appeared as though the shaft had been cut with a hacksaw. This caused the engine to break its governor belt and by the time the night man got to it, it was doing its best to get out of the engine house.

The engine needed considerable repairing before being used again. Nothing broke, but things got twisted and sprung and the bevel gears lost all their teeth.

The company has since installed larger and better pumps and these troubles are a thing of the past as far as night work is concerned.

C. L. MALLERY.

Myerstown, Penn.

Cause of a Noisy Heater

An open cast-iron heater of standard make had been in use several years. At times lately this heater would emit a loud noise, accompanied by a heavy jar and violent fluctuation of the water level; this disturbance did not occur at regular intervals, but was most frequent when the boilers were using the most water.

The heater was connected to the exhaust of a cross-compound Corliss engine, and was located higher than the connection from the low-pressure cylinder. A valve in the exhaust line between the heater and engine was adjusted so as not to allow any more steam to go to the heater than could be condensed by the maximum amount of water admitted to the heater, the balance of the exhaust was used for drying. It was thought probable that the trap on the exhaust line did not operate, allowing water to accumulate in the line to the heater, then blow through causing the noise, but after examination this did not prove to be the cause.

When watching the exhaust head, located some feet above the heater, it was observed that at each occurrence of the noise the steam that was discharging

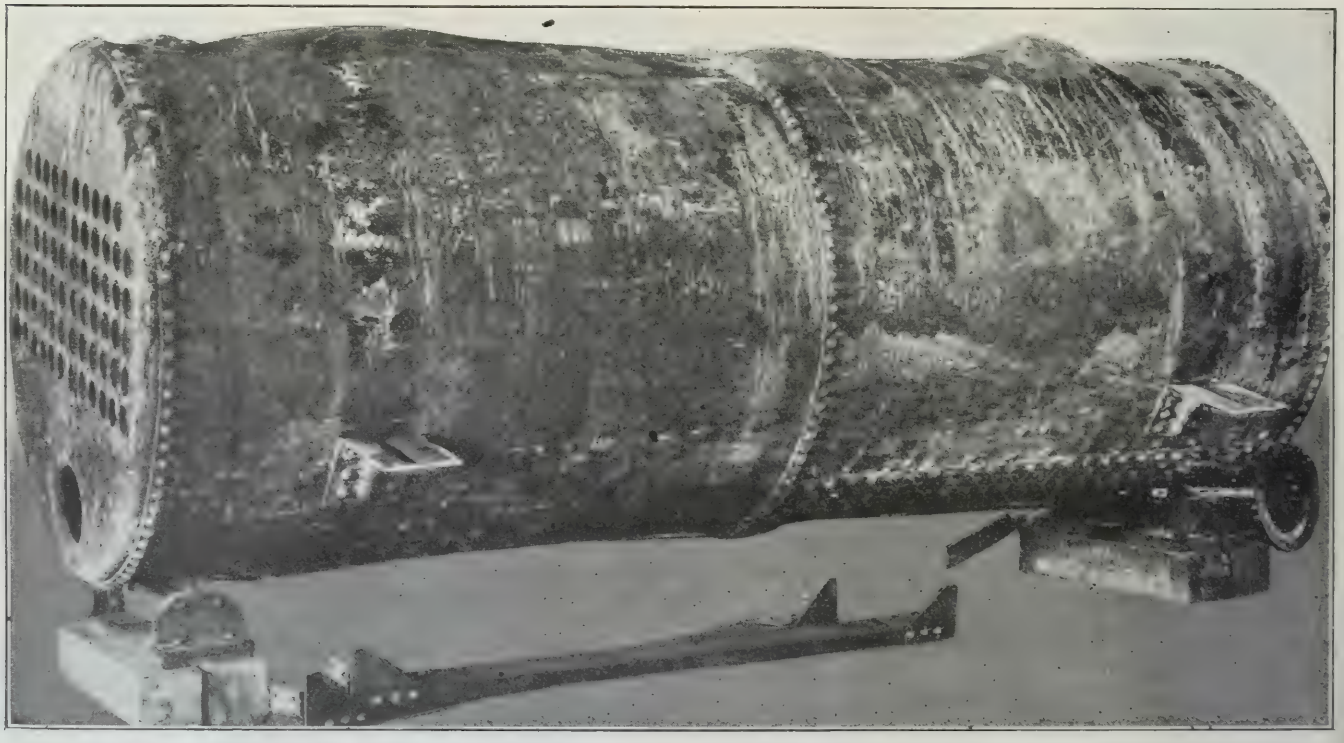
from the head would rush back down the pipe, indicating that there was a vacuum in the heater. This gave a clew that led to the close inspection of the balanced float-controlled valve on the water feed pipe. The stem of this valve was loose and corroded so as to take more than the usual power to make it move. When the valve closed it did not open again until the water level was almost below the float, when the latter was at its lowest position, the float would then open the valve which, being loose on the stem, would open wide, throwing a heavy spray into the top of the heater, thus condensing the steam and forming a partial vacuum that caused the noise and jar. After the valve was put in good order, we were not troubled with the noisy heater.

R. E. Rockett.

New York City.

Sediment Causes Serious Bags in Boiler

The illustration shows a 72-in. by 18-ft. boiler of our make (Houston, Stanwood & Gamble Co.) that was badly damaged in the plant of a large manufacturing establishment a short time ago.



Showing Bags Due to Sediment. Boiler Is Upside Down to Rest on Timbers

This boiler is butt-strapped, triple-riveted, made for 125 lb. working pressure and at the time of the accident had been in use only about a year. It is one of three boilers, set separately, which were installed at the same time. The factory is in a small town, and is probably the largest user of water in that town. Lately, a large amount of new water pipe had been laid, and these pipes had been dipped in coal tar. No adequate provision was made for preventing the débris that got into the pipes before they were laid, along with the coal tar, from being pumped into the boiler. Much water was used for other purposes and the strong currents thus created in the pipes allowed the deposits to be carried in large quantities to the boiler. The sediment in the largest bag was about 3 in. thick. The sediment is gray and powdery, feeling very smooth when rubbed with the thumb and fingers.

C. R. Houston.

Cincinnati, Ohio.

Steam Boiler Legislation Needed

A 72-in. return-tubular boiler was supposed to have been cleaned at the end of the previous season. Whether the boiler was inspected or even looked into before being put into commission this spring, is not known. The fireman, however, could not blow it down with 10 or 15 lb. of steam and the engineer told him that it would be all right when the steam pressure got higher.

A new man took charge about the middle of the season and a stream about the size of a lead pencil was the best he could get through the blowoff. He was alarmed and wanted to investigate, but was not allowed to do so. The blowoff was not protected; and sand and scale about 5 in. deep rendered the shell-heating surface practically valueless. Apparently the only thing that prevented a disastrous explosion was that the boiler could make all the steam necessary with very moderate firing.

Doubtless many other similar conditions exist in localities not protected by law, but a few such exposures should bring about some necessary steam-boiler legislation.

A. J. Carr.

Portland, Maine.

Cotton Ropes for Power Transmission

The first rope drives were in Dundee, as that town is the center of the hemp and jute trade. The ropes employed were hemp and even stiffer than manila hemp for they were more closely spun or twisted.

As a textile district, Lancashire was more important and much busier than Dundee, and rope driving became very popular. But Lancashire, being built on cotton, would have nothing to do with hemp, but must employ cotton ropes. Just prior to this rope-driving enthusiasm the rise of numerous engineering firms had taken place.

Their proprietors were men of little technical knowledge and the wheel gears in the works (i.e., toothed gears) were as rough and as noisy as themselves. These undesirable gears accelerated the introduction of the American driving system with large wide belts on big driving pulleys and some excellent belt drives were installed.

Rope driving superseded belts almost entirely in Lancashire, but they endured

longer in Yorkshire, for that was the home of the woolen industry and the Yorkshiremen preferred to do anything rather than follow the lead of Lancashire. The wide leather belts did not run badly, but had an objectionable flapping, air-clapping trait even when running up hill onto the driven pulley; this is unknown with ropes.

Even within recent years, although ropes have long been almost the sole system of driving, badly designed rope drives have been installed by electrical engineers whose ideas of driving were at variance with the characteristics of electric machines. Without hesitation they would set a dynamo low down on the floor and drive it from the large rope pulley, the slack side of the ropes running down to the driven pulley. Whenever a rope runs down hill in that way it tends to throw itself into a loop instead of hanging down in a catenary curve and enveloping more, instead of less, than half the circumference of the driven pulley. Matter in motion tends to proceed in a straight line, and as the rope comes over the top of the driving pulley it will try to keep a horizontal path, but it is hauled down by gravity and by the preceding part of the rope. If the pulleys are 30 ft. apart and the rope is traveling at 6000 ft. per min. it will cross the 30-ft. gap in 0.3 sec. But in 0.3 sec. a body will fall

$$16 \times 0.3^2 = 1.44 \text{ ft.}$$

The vertical distance between the top of the, say, 30-ft. driving drum and the 6-ft. driven drum is perhaps 12 or 10 ft. and so there is an effort for the rope to make a loop or hump of 8 or 10 ft. above the driven pulley. Is it surprising that ropes will run around a small driven pulley without even touching it? In a traction load these rope humps are especially bad if the generator has no flywheel effect. But with a well designed system the results are very good. One of the best of nine traction stations under my charge, so far as freedom from breakdowns and other troubles is concerned is the one rope-driven station.

In Lancashire practice there may be 40 ropes on the large pulley which are led off to the various rooms of the mill, so many ropes to each driven pulley. The ropes to the highest pulley go up at a very steep angle, those to the lowest may run slightly down hill. If the ropes be looked at when running, only the lowest set will ordinarily have any swaying motion; they run toward the driven pulley with a slight waving motion. But all the other ropes hang steady, and have a slightly reversed curve onto the faces of the driven pulleys.

Opinion is divided as to whether a rope should have three or four strands. These stranded ropes cannot pull out of shape for the third strand has nowhere to go except the place intended for it. Three strands of a four-strand rope can throw off the fourth weakling and put a rope out of shape.

Personally I could never see any good in making a four-strand rope around a straight centered core that must somewhat easily break when the twisted part of the rope stretches. But I have used four-strand small ropes, or rim bands on mules and they acted reasonably well.

It is usual to dress cotton rope slightly with plumbago and black lead. This serves to hold down fluffiness.

In Yorkshire, square ropes of leather have been used. They are made from 1¼- or 1½-in. strips of leather formed into a square and are easily pieced, using French nails to hold the strips together. They do not lie even but cling to both sides of the grooves and present all sides upward at different times for they do not always keep each face square but twist in themselves. They drive well and are very satisfactory.

A few leather-covered wire ropes are used. These have a ⅝-in. steel-rope core to do the work and an external cover of 1½-in. leather washers which may be threaded on before splicing, the remainder being put on by cutting an acute diagonal through them and slipping them on. Such washers may be punched out of any old belting. These ropes do well and run for years. I know of cotton ropes, too, which have run for 25 years without a change.

A speed of 4800 ft. per min. is regarded by some as the maximum best speed. Beyond this velocity it is held that centrifugal force, increasing as the square of the velocity, adds more to the stress in the rope than the speed increases the capacity. At usual working tensions these two factors cross one another on the performance curves. Others, including one of the leading makers of cotton ropes, say that centrifugal stress does not exist in practice, but that the rope is gripped in the rim grooves sufficiently to hold it there and it suffers no tension from centrifugal stress, but that action helps the rope to leave its groove when the time comes for it to do so. And by this reasoning speeds of 7000 ft. per min. are held to be safe. But there must be some velocity at which the rope will not grip in its groove all the way around the wheels, and a still higher velocity where it will not even touch the grooves. These are for the mathematician to work out. It appears certain that at 7000 ft. the danger point has not been reached.

Manila, in this country is not favored as a driving rope. It is claimed that it is rough fiber and will grind itself away quickly. But I well remember that the late C. W. Hunt would not agree to this. He contended that a manila rope properly laid up of selected fiber and lubricated internally was superior to cotton. Manila is a stronger fiber, but cotton is very much more elastic and pliable. Every one of its many threads is of identical size and in the system of laying the threads parallel and untwisted in little strands there is obtained the maximum of strength. An ordinary rope will be rated at 50 hp. at 5000 ft. per min. This implies a working stress of only 350 lb. in a rope apart from the slack-side tension and the centrifugal tension. Ropes are run with a wide margin of strength and this is considered to pay in the long life it gives to the ropes.

Ropes run best when they always expose the same part uppermost, for they bed well to their grooves. But occasional ropes in a set will rotate about their longitudinal axis and they are thus constantly kneaded between the sloping faces of the grooves. They then become fluffy and short lived. Nobody seems able to discover why they rotate, though it is possibly due to the failure, when splicing, to lay them out and take off all superfluous twists tending to form kinks. New ropes should be hung with heavy weights to stretch them before splicing and a splice should be very long, at least 72 rope diameters in length. A well spliced rope joint cannot be distinguished after a short run.

Rope driving is being displaced in some cases by electric driving and greater production is claimed, but no very startling evidence has been forthcoming. The electric drive is usually compared with some old drive of 30 or 40 years' standing. Many present electric drives are little else than the use of a motor on the mill shafts instead of the rope pulley and no shafting is abolished, or where shafts are abolished any stock motor has been dumped down on the floor and set to drive a machine through a short stiff belt. These crude methods have done harm to the electric drive which should be made to do things that cannot so well be worked out mechanically.

As regards driving efficiency it was customary to say that where good gearing would drive a given mill with 1000 i.hp. a belt would demand 1050 i.hp. and ropes 1100 i.hp. But the ropes went with quicker running engines and higher pressures and they introduced of necessity such large rim wheels that the flywheel effect was many times that of the old spur rim, and the machinery ran without perceptable variation of hum, and less waste and bad work resulted. And ropes did not cause heavy breakdowns. If a rope fails it fails gradually, stretches abnormally, relieves itself of its share of work, and if badly neglected it throws out short wisps and loose tails of yarn which strike a safety bar and with a mere touch will stop the engine before the rope can run amuck.

W. H. Booth.

London, S. E., England.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Nonreturn Stop Valves

The editorial on this subject in the Nov. 5 issue was indeed interesting. I agree with everything said, because my experience with nonreturn valves corroborates the editorial.

In our plant we generate nearly 25,000 boiler horsepower in four boiler houses, which are all connected by a common steam main. If these boilers were not protected by nonreturn valves, and a tube should blow out, one can easily imagine the loss in time which would result. It might not only shut down the boiler house where the accident occurred, but even one of the other houses, if the valve in the line was not quickly closed. These conditions were considered sufficient reason for installing nonreturn stop valves on every boiler in the plant. This was done four years ago.

Besides the possible saving in time, there is, as the editorial pointed out, an additional reason for their use in that they are insurance against accident to workmen and hence are of value to the company. If a fireman should be burned, he might, after suffering pain and loss of time, recover damages from his employer. While the use of nonreturn valves will not keep tubes from blowing out, they minimize the danger attending such accidents. Consequently, they would undoubtedly cut down the number of accidents, and the number of resulting suits for damages, thus proving a saving in another way besides that of time.

An experience of mine illustrates this. A boiler was out of service for cleaning. In the drum, the manhole of which faced the adjacent boiler, was a boiler washer, and an inspector was working in the furnace. The boiler next to it blew out a tube and, although the men were badly frightened, they suffered no injury. If this boiler had not been equipped with a nonreturn valve, which operated instantly, these two men would undoubtedly have been boiled to death before the stop valve could have been closed.

These valves are reliable, too, for we have had five tubes blow out since they were installed, and each time the valve closed instantly. Again, when firing up a boiler there is nothing to worry about when it comes to cutting the boiler in on the line. When the boiler pressure equals the pressure in the header, the valve opens automatically.

Aside from all the obvious advantages of the nonreturn valve that mean a probable saving in time and money, and which certainly justify the expense as insurance there is a more important consideration, that is, the humanitarian side. Wherever it is possible to safeguard workers by installing safety devices it should be done. The automatic nonreturn stop valve comes in this category, and so its use along with other safety devices, should be required by law.

J. F. MOWAT.

Joliet, Ill.

Mr. Hawkins' letter and the editorial on the above subject is timely and should receive the approval and support of every engineer. I have had as many cases to deal with as Mr. Hawkins mentioned and a shutdown was the result in each of them.

The most serious accident was to a large tandem-compound engine in a power plant containing five other engines rated from 150 hp. to 1000 hp. running both direct- and alternating-current machines for street-car service and lighting.

The trouble started with a hot crosshead and before the guide could be loosened and the engine stalled, which was serious as there were no spare machines to carry the load. The crosshead parted at the wristpin and split the cylinders from end to end, broke the yoke connection between the two cylinders, "punched" out the cylinder head from the high-pressure cylinder, broke the end of the low-pressure steam chest and, worst of all, broke off the flange from the upper end of the throttle valve. The only means left to stop the flow of steam that followed the breaking of the flange was to close the stop valve on the engine steam pipe, located near the steam main, some 20 ft. from the floor. The break in this 6-in. pipe quickly filled the room with steam.

The oilers and assistants closed the throttles on the other five engines. One man had to crawl on his hands and knees to reach the engine-room door, as the windows were all covered with heavy wire screens. This shows how quickly a large room can fill with steam.

There were seven 72-in. by 18-ft. boilers in operation carrying a steam pressure of 115 lb. All the safety valves were fitted with a 3-ft. piece of pipe pointing toward the roof, which was about 12 ft. from the top of the boilers. Five of the boilers were connected at the front nozzle to an equalizing main from which steam was taken to run the condenser's auxiliaries and feed pumps. The stop valves on the risers were never packed up to the time of this accident since the plant started five years before. The wheels of these valves could hardly be turned with a 24-in. wrench, although the stems were leaking. The boiler room was already filled with steam above the boilers. But the flow of steam had to be stopped and the assistants, oilers and everybody in the plant except the firemen were working hard to close the valves. The valves were finally closed, but almost immediately the safety valves on two other boilers started to blow to add to the already uncomfortable heat of the room. Luckily the firemen had the fires banked just as the valves opened. The plant was idle for about three hours. Some machines were badly affected by moisture from the steam and were run some time without load to dry them out.

One year later the connecting-rod and crosshead on the high-pressure side of a 500-hp. direct-connected railway unit dissolved partnership by breaking the strap and the same experience was repeated. This was the third engine wreck in this station, but a nonreturn valve was unheard of at that time. Numerous other accidents have happened with some narrow escapes from injury and death.

This is only one plant out of thousands which are experiencing just such troubles, and while the public is suffering the effects of delays in service, the management busies itself by repairing and replacing the old valves only to meet another shutdown, when under a heavy overload some engine "takes a drink" from a foaming boiler and "punches" out the head or bursts a receiver and many other numerous accidents.

Many public-service corporations are installing nonreturn valves in their new plants and replacing the old valves with them as rapidly as repairs are necessary, but in some smaller plants where the danger to life and property is even greater a suggestion on the part of an engineer that nonreturn valves should be installed in place of a broken stop valve, is usually met with a retort that the valves have given satisfaction for 20 years and such accidents may never happen, and it is only by chance that such breaks do occur.

But these accidents are becoming too frequent of late years with the increase of boiler pressures, and like the old lap-seam boiler and the old style safety valves the stop valve must give way to be superceded by a nonreturn stop valve. Engine steam pipes should also be equipped with a nonreturn stop valve beween the throttle and the steam main which would cut out an engine as soon as the cylinder loses a head.

It is the engineers' duty to put this fact up to the managements of their companies and if they refuse to heed the warning then the license laws should be amended and legislation sought, which would prevent a new plant being put into service unless such stop valves were provided and that no stop valve on any boiler now in service, when once removed for repair should be put in such service again but be replaced by a nonreturn stop valve instead.

I hope the state of Massachusetts will soon have such an amendment to its boiler laws. I wish to thank Mr. Hawkins and POWER for bringing this important subject to the engineers' attention, for there are many in the business who have not yet seen the necessity of using the nonreturn stop valve on boilers.

R. A. CULTRA.

Cambridge, Mass.

Diagram for Quickly Finding M. E. P. at Various Cutoffs

The table of mean forward pressures for various cutoffs and initial steam pressures, given by Wallace S. Fowler, on page 632 of the Oct. 29 issue, suggests that I might add a time saver of my own along the same line. It is incorporated in the accompanying diagram, which I plotted some years ago, because I found such a diagram to be of material aid in my work.

But little explanation is required, except to say it is based upon the regular mean pressure formula

$$P_m = P_1 \frac{1 + hyp.\ log.\ R}{R}$$

with which everyone is familiar.

As an example of the use of the diagram, let the initial steam pressure be 164.7 lb. abs., the back pressure 4 lb. abs., and the cutoff ⅓. Also let the diagram factor be 0.8, all as given in Mr. Fowler's example. From the curve we have the ratio of absolute mean pressure to absolute initial pressure, for ⅓ cutoff, given as 0.7. So the absolute mean pressure is 0.7 of the absolute initial, or

$$0.7 \times 164.7 = 115.29\ lb.$$

Deducting the same absolute back pressure of 4 lb., leaves a theoretical mean effective pressure of 111.29 lb., and multiplying by the assumed diagram factor of 0.8 gives 89 lb., agreeing with Mr. Fowler.

One of the greatest conveniences of such a curve is the possibility of working the problem backwards, to determine the necessary cutoff. The application of the "hyp. log. formula" is difficult for this.

Let us assume we must have 89 lb. m.e.p. with 164.7 lb. abs. initial and 4 lb. abs. back pressure. With a diagram factor of 0.8, as before; what must the cutoff be?

The theoretical mean effective pressure must be

$$89 \div 0.8 = 111.25\ lb.$$

With 4 lb. back pressure, the mean absolute forward pressure must be

$$111.25 + 4 = 115.25\ lb.$$

If the absolute initial pressure is 174.7 ib. the ratio of the absolute mean to the absolute initial pressure must be

$$\frac{115.25}{164.7} = 0.7$$

A glance at the curve shows that this ratio of absolute mean to absolute initial pressure corresponds to a cutoff of 0.333 or ⅓. This agrees with the original problem.

Further, let the problem be to determine the necessary initial pressure so that, with ⅓ cutoff and 4 lb. back pressure and 0.8 diagram factor, we may have 89 lb. m.e.p. As before, the theoretical mean effective pressure equals

$$89 \div 0.8 \times 111.25\ lb.$$

and the absolute mean forward pressure equals

$$111.25 + 4 = 115.25\ lb.$$

The curve shows that ⅓ cutoff gives the ratio of absolute initial to absolute mean pressure of 0.7, as before, so the absolute initial pressure must be

$$115.25 \div 0.7 = 164.7\ lb.\ abs.$$

as before. If this initial pressure is not satisfactory, it might be desirable to alter the cutoff, so one can glance at the curve and find that with, say ¼ cutoff, the ratio of the pressures is 0.596, so the initial pressure would then have to be

$$115.25 \div 0.596 = 193\ lb.\ abs.$$

Or, if a lower initial pressure is desired, a cutoff of say 0.4 can be selected, giving an initial pressure of

$$115.25 \div 0.766 = 150\ lb.\ abs.\ (about)$$

It was for just these "backward jugglings," to determine the best conditions that I originally plotted the curve, as a curve gives all the intermediate values, which is rather impracticable in a table. And, again, a curve gives one a clear mental picture of the various relations.

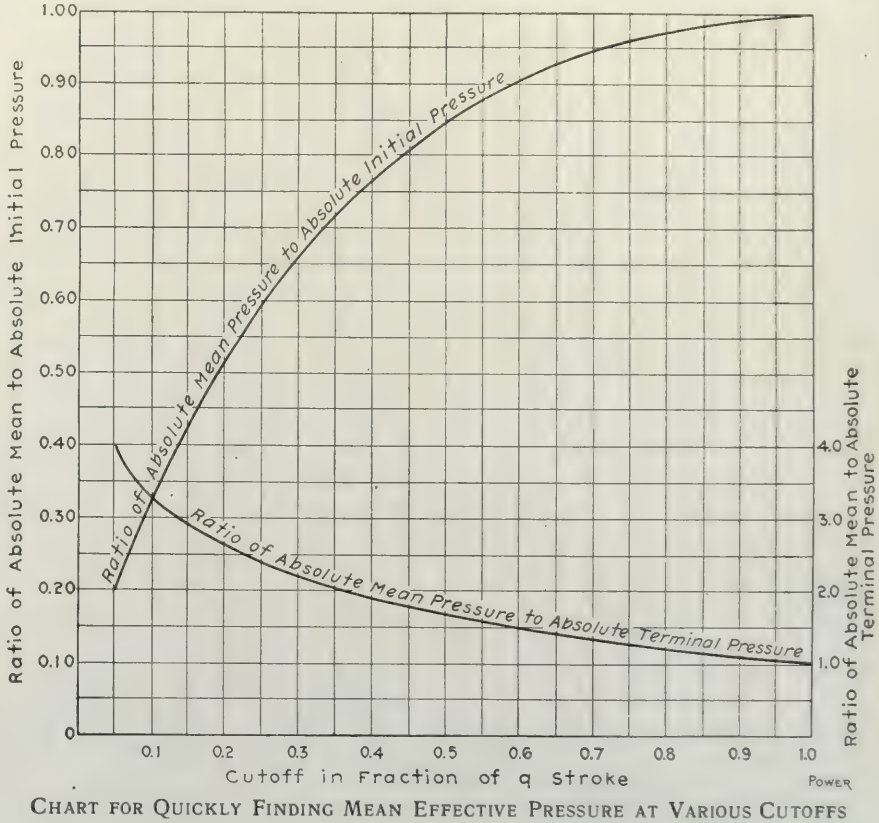


CHART FOR QUICKLY FINDING MEAN EFFECTIVE PRESSURE AT VARIOUS CUTOFFS

Another use for the curve is its application to compressed air. Isothermal air compression is the same as theoretical steam expansion if exhaust occurs at terminal pressure. For obtaining the efficiency of compression of an air compressor referred to the isothermal, it is necessary to compute the isothermal horsepower. Let us assume compression from atmosphere to 100 lb. gage. The ratio of compression is

$$\frac{114.7}{14.7} = 7.8$$

This corresponds to the ratio of expansion in a steam cylinder, so the cutoff is its reciprocal or

$$\frac{1}{7.8} = 0.1282$$

The curve gives the ratio of absolute mean to absolute initial pressure for this cutoff, as 0.391. As in air compression, the final, or discharge pressure corresponds to the "initial" pressure of a steam diagram, the mean pressure is

$$0.391 \times 114.7 = 44.9$$

Also, as in air compression the initial pressure corresponds to the back pressure of a steam diagram, we must deduct the initial pressure, which in this case is atmospheric. So 44.9 — 14.7 = 30.2 lb. m.e.p. for isothermal air compression from atmosphere to 100 lb. gage.

This last problem may also be worked from the lower curve of the diagram. A cutoff of 0.1282 corresponds to a ratio of absolute mean pressure to absolute terminal pressure, of 3.055, and as in air compression the initial pressure corresponds to the terminal pressure of a steam diagram, we have:

$$\textit{absolute mean pressure} = 3.055 \times 14.7 = 44.9 \textit{ lb.}$$

and subtracting the atmospheric pressure, as before, we have

$$44.9 - 14.7 = 30.2 \textit{ lb. m.e.p.}$$

It is difficult to read the curves as closely as this, but the diagram is very handy for approximate calculations, and if desired it may be replotted to a larger scale.

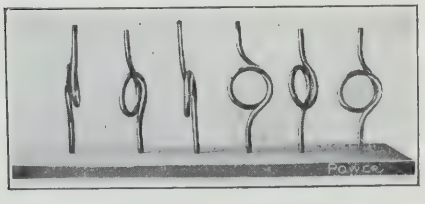
R. S. BAYARD.

New York City.

Rosin for Bending Pipe

C. R. Summer's letter in the Sept. 3 issue gave me an idea of how to make siphons for steam gages. He said he had never seen rosin tried for bending pipe but did not see why it would not work as well as for bending boiler tubes.

It does work well. I made six as perfect siphons, or goose necks as they are



SAMPLES OF BENDS MADE WITH ROSIN-FILLED PIPE

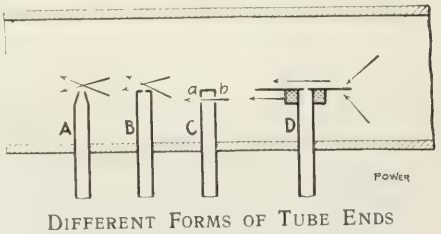
sometimes called, as any engineer would wish to have his steam gage fastened to, without a failure. I used ¼-in. black iron pipe and bent it cold over a 2½-in. pipe held in a pipe vise, threading both ends of the ¼-in. pipe before bending. I put a cap on the pipe before filling it with melted rosin, and before bending. The other end was capped to prevent spoiling the threads. A pipe about a foot long, which will go over a cap, will help in bending. A bunsen burner was used to melt the rosin and clean the finished siphons. The siphons were without a sign of a flat place; they are shown in the accompanying engraving.

H. S. SWEETSER.

Medford, Mass.

Pitot Tube Ends

On page 390, Sept. 10 issue, I noticed a discussion as to the merits of different forms of pitot tube ends, by G. G. Crewson. This is all very interesting to me as I have had some experience along the same lines. I tried the static tube ends as shown at *A*, *B*, *C* and *D* in the accompanying sketch and found that *A* and *B* would not give duplicate readings for conditions that were apparently constant, because of cross-currents, as indicated by



DIFFERENT FORMS OF TUBE ENDS

the arrows. The end *C* was very satisfactory but the least variation of the line of the hole *ab* from the line of air movement gave an appreciable error. However, the disk on the end of the tube, directs the air perpendicularly over the tube end and serves to eliminate the effect of cross-currents. The two last check each other well when *C* is carefully set. The ease with which *D* may be set (assuming, of course, that it is possible to get the disk in the duct at all) makes this modification very satisfactory.

J. D. HOFFMAN.

Lincoln, Neb.

Scale in Steam Turbines

The editorial in the Nov. 19 issue of POWER regarding scale on turbine blades furnishes an interesting topic for discussion. On first thought it would seem impossible for scale to form when the steam is properly superheated and a separator is installed in the line. Nevertheless, it seems to be a fact that this sometimes happens, and there must be some explanation.

There are only three ways that scale could get into a turbine under any circumstances, viz., through the steam inlet, through the journals and through the exhaust outlet. The latter can certainly be eliminated as a source, for there is a constant current of steam in the opposite direction. Leakage of scale-forming water through the journals is a possible cause of scale.

It is easily understood how scale could form on a turbine blade provided saturated steam were used which had not passed a separator or where the separator was of an inefficient type. Let us now consider what might happen in case superheat were carried.

As it leaves the boiler proper, even under normal conditions, steam will contain sometimes 3 to 5 per cent. of water. Bad, scale-forming water it probably is, too, due to concentration in the boiler. This water is in a finely divided state suspended in the steam. As the steam enters the superheater, each small particle of water it carries is evaporated, leaving a microscopic particle of scale not nearly heavy enough to drop out of the rapidly moving current. What happens then? These minute dust-like scale particles come in contact with everything which the steam hits and are for the most part carried out with the steam to the turbine.

What effect will a separator have on these particles? All common forms of separators are intended to separate liquids from gases. They are based on the principle that the momentum of a water or oil particle will carry it against the baffle while the steam current passes around the sides. They are not built to completely remove dust from a gas current.

We thus have these extremely minute scale particles passing by the separator to a large extent, and hitting, with the steam, against the turbine blades. Some of them stick and in time form a dense hard scale as described in the article to which reference was made. I believe that if the baffle-plate of the separator were inspected it would be found to be covered with the same dense hard scale which was found on the turbine blades. This would, of course, make it inefficient for removing water from saturated steam, as the ribs might be entirely plastered over. Ordinarily a separator is self-cleaning as the water removed from the steam is continually washing its surfaces.

Now, supposing the above explanation is the correct one, and it certainly seems plausible, what is the remedy? Naturally to put an efficient separator between the boiler and superheater, thus preventing any scale-forming water from leaving the boiler. Care should be taken to install one which is known to be efficient and the maker should be consulted as to the requirements. With such an arrangement, I believe the difficulty would be solved. It would at least be interesting to know what results are secured where such an arrangement exists.

EDWARD H. ROBIE.

Philadelphia, Penn.

On Thursday evening, Nov. 14, the Engineers Orchestra, of Worcester, comprised principally of members of Worcester No. 4 and H. E. Stone Association No. 32, N. A. S. E., gave its first public concert of the season. The program was well rendered and the concert was well attended.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Rotary Pumps

What is the advantage of rotary over centrifugal pumps used for private fire service?

R. S.

Rotary pumps can be operated under variable heads and speeds and are therefore more desirable as fire pumps in cases where the power is likely to be supplied at an uncertain speed.

Friction of Shafting

Will it take more power to drive 50 ft. of shafting from one end with a load on the other end than to drive the same shafting with the same load near the driven end of the shafting ?

H. P. A.

Yes; more power will be absorbed by the shafting when transmitting power from one end to the other than when a part of it runs idle, as the transmission of a load tends to throw the shafting out of line, thereby increasing the friction of the bearings.

Packing for Gasoline Joints

What is a good material for packing glass gage fixtures of oil or gasoline tanks?

J. H.

Ordinary rubber gage-glass packing rings, shellac varnished and dried before use, will be found to resist the solvent action of oil or gasoline; and when thus protected the rubber retains its elasticity and remains a very reliable packing material for a considerable length of time. Rubber rings used in this manner should be of proper size to fit snugly around the glass, but without being stretched so as to separate the shellac coating.

Cost of Coal vs. Wood as Fuel

What would have to be the price of average good bituminous coal per ton to equal the cost of dry wood at $4.25 per cord of 3800 lb., used as fuel for steam boilers?

H. D. B.

Cord wood that has been dried for about one year and containing 20 to 25 per cent. moisture, will have about 45 per cent. as much heating value per pound as bituminous coal of average quality. A cord of such wood weighing 3800 lb. therefore would have as much heating value as $3800 \times 0.45 = 1710$ lb. of coal, and the cost of it being $4.25 would be equivalent to coal costing

$$\$4.25 \times \frac{2000}{1710} = \$4.97 \text{ per ton of } 2000 \text{ lb.}$$

Electrical Transmission

What would be the number of brake horsepower required for driving a direct-current generator suitable for a delivery of 40 brake horsepower by electrical transmission 150 ft. from the generator?

H. L. H.

The average efficiencies of generators and motors of the sizes required would be about 90 per cent. for each. Assuming that transmission of current is by wiring of ample cross-section for the voltage, the efficiency of transmission would be about 97 per cent., and the power required for driving the generator would be about

$$\frac{40}{0.9 \times 0.9 \times 0.97} = 50.91 \text{ b.hp.}$$

Size of Pump Air Chamber

What should be the size of the air chamber for a double-acting, single-cylinder, 6x10-in. boiler-feed pump?

R. E. B.

The content of the air chamber should be not less than three times the pump displacement for a single stroke. The area of a 6-in. diameter piston being

$$6 \times 6 \times 0.7854 = 28.274 \text{ sq.in.}$$

the displacement per single stroke would be

$$28.274 \times 10 = 282.74 \text{ cu.in.}$$

and the content of the air chamber should be

$$282.74 \times 3 = 848.22 \text{ cu.in.}$$

If the air chamber is made cylindrical 8 in. in diameter with the upper end hemispherical, then as the volume of a sphere may be found by the formula

$$\text{vol.} = \tfrac{1}{6} \times 3.1416 \times \text{cube of the diameter}$$

for a hemisphere 8 in. dia.

$$\text{vol.} = \frac{\frac{1}{6} \times 3.1416 \times (8 \times 8 \times 8)}{2}$$

$$= 134.04 \text{ cu.in.}$$

This would leave

$$848.22 - 134.04 = 714.18 \text{ cu.in.}$$

to be contained by the cylindrical part of the air chamber. The diameter being 8 in., the cross-sectional area would be

$$8 \times 8 \times 0.7854 = 50.265 \text{ sq.in.}$$

hence the length of the cylindrical part should be

$$\frac{714.18}{50.265} = 14.2 \text{ in.}$$

Pumpage per Hour

A duplex pump having double-acting water pistons 10 in. in diameter and making a 12-in. stroke has piston rods 1¾ in. in diameter. What would be the net discharge in gallons per hour, making 52 double strokes per minute, allowing 5 per cent. slip ?

E. R.

The outer side of each piston would have an area of 78.54 sq.in. and the net area of the piston-rod side of each piston would be the same less the cross-sectional area of its 1¾ in. diameter piston rod, i.e.,

$$78.54 - 2.40 = 76.14 \text{ sq.in.}$$

Therefore, the piston displacement for one double stroke would be

$$(78.54 + 76.14) \times 12 = 1856.16 \text{ cu.in.}$$

or

$$\frac{1856.16}{231} = 8.035 \text{ gal.}$$

per double stroke, which with 5 per cent. slip is reduced to

$$8.035 \times 0.95 = 7.633 \text{ gal.}$$

and with the pump making 52 double strokes per minute, the net discharge would be

$$7.633 \times 52 \times 60 = 23{,}815 \text{ gal. per hr.}$$

Dimensions for 15 Gal. Tank

What should be the length and thickness of material of a 15-gal. gasoline tank made of copper if 12 in. diameter and provided with one inward and one outward bumped head?

W. F. B.

One U. S. gallon is 231 cu.in. and 15 gal. would be

$$231 \times 15 = 3465 \text{ cu.in.}$$

The cross-sectional area of 12 in. diameter would be

$$12 \times 12 \times 0.7854 = 113.0976 \text{ sq.in.}$$

therefore 1 in. in length of the tank would contain practically 113.1 cu.in. and to contain 15 gal. or 3465 cu.in., the length of the tank would have to be

$$3465 \div 113.1 = 30.63 \text{ in.}$$

or practically 30⅝ in. To be laid on its side and supported in saddles the tank could be made of No. 14 gage copper, with joints brazed and copper-riveted.

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Percentage

When we are told that the price of certain articles is so much "per dozen." "per ton" or "per pound" we readily understand what is meant. The word "cent" is an abbreviation of the Latin *centum*, which means one hundred. Thus we have the century, one-hundred years; cent, the one-hundredth part of a dollar; centimeter, the one-hundredth part of a meter, etc.

Per cent., therefore, means by the hundred or hundredths; a certain number of parts out of each one-hundred parts. Five per cent. of a dollar equals 5 cents; $\frac{5}{100}$ of a dollar = 5 cents.

The following table shows clearly how percentage values may be expressed:

1 per cent. = $\frac{1}{100}$ or 0.01
5 per cent. = $\frac{5}{100}$ or $\frac{1}{20}$ or 0.05
10 per cent. = $\frac{10}{100}$ or $\frac{1}{10}$ or 0.10
20 per cent. = $\frac{20}{100}$ or $\frac{1}{5}$ or 0.20
50 per cent. = $\frac{50}{100}$ or $\frac{1}{2}$ or 0.50
75 per cent. = $\frac{75}{100}$ or $\frac{3}{4}$ or 0.75
100 per cent. = $\frac{100}{100}$ or 1 or 1
125 per cent. = $\frac{125}{100}$ or $1\frac{1}{4}$ or 1.25
$\frac{1}{2}$ per cent. = $\frac{1}{2}$ of $\frac{1}{100}$ or $\frac{1}{200}$ or 0.005
$\frac{3}{4}$ per cent. = $\frac{3}{4}$ of $\frac{1}{100}$ or $\frac{3}{400}$ or 0.0075

In examples in percentage we have three quantities to consider:

The Base.

The Percentage.

The Rate.

The *base* is the quantity upon which the percentage is calculated, or to which the rate per cent. refers. In the statement "18 is 6 per cent. of 300," the base is 300, because it is the number to which the rate 6 per cent. refers.

When you say 6, 8 or 10 or any per cent. of any quantity or number is so much, that quantity or number spoken of is the *base* because it is the number that is supposed to be divided into 100 equal parts.

Suppose the coal consumption in a plant increased 15 per cent. and that the consumption originally was 100 tons. Then, there is now being consumed 15 tons for every 100 tons or 115 tons. If the consumption had increased 100 per cent., then the amount consumed after the increase would be 200 tons.

The term per cent. is expressed % in writing; 15% is read 15 per cent. In computation or in "figuring" examples in percentage the sign % is not used and the numbers are expressed in decimals instead of in fractions. Thus instead of writing 25% or $\frac{25}{100}$ it is written 0.25.

The *percentage* is that part of the base that is indicated by the rate per cent., and is found by multiplying the base by the rate. Thus in the statement, "18 is 6 per cent. of 300" the *percentage* is 18. Thus in our coal problem: 0.15 × 100 = 15, the percentage is 15.

The *rate per cent.* is the number of 100ths of the base that is taken or considered. As we said above, this figure is always expressed in 100ths in calculations. In the statement "18 is 6 per cent. of 300," 6 is the rate and "6 per cent. of" means 0.06 of the base 300.

If any two of these quantities, that is, the base, percentage or rate are known, the third is easily found. The percentage being the product of the base and rate is easily found by multiplying the base by the rate, while either the rate or base may be found from the percentage by dividing the latter by whatever quantity is known, thus:

$$Base \times Rate = Percentage$$

$$\frac{Percentage}{Base} = rate$$

$$\frac{Percentage}{Rate} = base$$

The chief thing to remember is that the per cent. or rate is to be expressed in 100ths.

To apply what we have learned a few problems will be worked out.

Rule: *Percentage equals base multiplied by rate expressed in 100ths.*

Example: A battery of boilers burned 12 tons of coal per day; after stopping leaks in the settings and changing the manner of firing a saving of 4 per cent. was made. What was the saving in coal per day?

Remember that the 12 tons is supposed to be divided into 100 equal parts.

That a saving of 4 per cent. was made means simply that of every 100 parts of the 12 tons, 4 parts were saved, i.e., the saving is $\frac{4}{100}$ or 0.04 of the amount originally used. Here 12 is the base and 4 is the rate and we want to find the percentage or saving. Then applying the rule:

$$12 \text{ tons} \times 0.04 = 0.48$$

or nearly ½ ton saved per day. Ans.

When the percentage and base are given, to find the rate:

Rule: *Divide the percentage by the base; the quotient will be the rate expressed in hundredths.*

Example: A battery of boilers required 12 tons of coal to evaporate a given amount of water. After the scale and soot were removed, 0.48 ton less would evaporate the same amount of water. What was the saving in per cent.?

Here 12 tons is the base and 0.48 ton is the percentage. Now to find the rate.

$$\frac{0.48}{12} = 0.04$$

or 4 per cent. is coal saved. Ans.

When the percentage and rate are given, to find the base.

Rule: *Divide the percentage by the rate, expressed in hundredths; the quotient will be the base.*

Example: Changes in operating a battery of boilers resulted in a saving of 0.48 ton of coal per day, which was a saving of 4 per cent. What was the coal consumption?

If 0.48 ton is 4 per cent., 100 per cent. will be

$$\frac{0.48 \text{ ton}}{4} = 12 \text{ tons.} \quad Ans.$$

Gain or Loss Per Cent.

When it is desired to find how much the base has been increased or diminished by the addition or subtraction of a certain percentage we may calculate it in either of two ways.

Example: A certain grade of coal contains 11 per cent. of ash. How many pounds of combustible are there in one long ton or 2240 lb. of such coal?

First, we may calculate the percentage of the ash by the method already given, and by deducting the amount of ash from the total or 2240 lb. find the amount of combustible, thus:

$$0.11 \times 2240 = 246.4 \text{ lb. of ash.}$$

$$2240 \text{ lb.} - 246.4 \text{ lb.} = 1993.6 \text{ lb.}$$

of combustible. Ans.

Second, we may say that since the 2240 lb. equals 100 per cent. and 11 per cent. is ash, then

$$100 - 11 = 89 \text{ per cent.}$$

of the coal is combustible.

Now

$$2240 \text{ lb.} \times 0.89 = 1993.6 \text{ lb.}$$

of combustible. Ans.

Example: A boiler was "doing" 350 hp. and later another engine was installed which increased the load on the boiler by 15 per cent. How much power did the boiler develop after the increased load?

We may find the additional load by finding 15 per cent. or

$$0.15 \times 350 = 52.5 \text{ hp.}$$

and then add this 52.5 to the original load or 350 hp. and so find the total

$$350 + 52.5 = 402.5 \text{ hp.}$$

is the total load. Ans.

Or, we may say that as 350 hp. is 100 per cent., the total load after the engine was added was 115 per cent. or

$$1.15 \times 350 = 402.5 \text{ hp.}$$

Perhaps the most frequent error in calculations involving percentage is that due to failure to distinguish between the

sum or difference of the percentage and the base.

Suppose you are buying coal that contains 11 per cent. of ash and you wanted to buy enough so that you would have just a ton, 2240 lb. of combustible. How many pounds of coal would you buy?

At first you are apt to take 11 per cent. of 2240 lb. as the additional number of pounds required.

$$1.11 \times 2240 = 2486.4 \text{ lb.}$$

But the ash is not 11 per cent. of the combustible required, but of the unknown quantity of coal which is the *base* from which the percentage is to be computed. The 2240 lb. is the *difference* between the base and the percentage, i.e., between the total amount of coal and the ash that it contains.

The whole quantity or 100 per cent. is made up of ash and combustible, and if the ash is 11 per cent., then the combustible must be 100 — 11 = 89 per cent. Then the required amount of combustible must be

$$\frac{2240 \text{ lb.}}{0.89} = 2517 \text{ lb.} \quad Ans.$$

Price and Discounts

The prices of power-plant machines, material and supplies as given in catalogs or price lists are nearly always subject to discounts, that is, the listed price is subject to reduction; a part of it is not to be paid or counted. Usually one cannot correctly estimate costs from a price list without knowing what discounts are allowed as the listed price gives no accurate indication of the actual cost.

Discounts are given in per cent. and expressed, 40 per cent. and 10 per cent. off or "40 and 10" as is usually said. One is apt to think this means 50 per cent. off, but it does not. It means that a discount of 40 per cent. is first made and a discount of 10 per cent. is made on the remainder. If the list price is $2.50 with "40 and 10" off, then 40 per cent. of $2.50 = $1. Now we deduct $1 from $2.50 = $1.50. Then 10 per cent. of $1.50 = $0.15, or 15 cents, and $1.50 — 0.15 = $1.35 = the actual cost.

This same procedure holds true no matter how the discounts are given, i.e., whether it is "40 and 10" or "40, 10 and 5."

Examples for Practice

(1) A pound of a certain coal contains 14,500 B.t.u., but this is 15 per cent. more than is actually used to generate steam in the boiler. How many B.t.u's. then, are used in generating steam?

(2) A plant used 4 tons of coal in one day and 9 lb. of water was evaporated per pound of coal. The feed-water consumption on this day was 25 per cent. greater than on the preceding day. What was the consumption on the preceding day?

(3) An engine was bought for $8000 and later sold for $6000, what was the loss in per cent.?

(4) If 26 tons of coal is 18 per cent. more than was received last week, how many tons were received last week?

(5) The listed price of a 16-in. siphon condenser, circulating pump and engine is $1225, but a discount of "20 and 10" is allowed. What is the actual price?

(6) The condenser auxiliaries require 2 per cent. of the total steam used by the main engine; the engine uses 3600 lb. per hour. How much do the auxiliaries require?

(7) An engine runs at 230 r.p.m. at no load and its governor is supposed to regulate the speed within 2 per cent. from no load to full load. The engine makes 228 r.p.m. at full load; does the governor regulate within 3 per cent.? If not, how "close" is the regulation?

Answers to Last Week's Questions

(1) 641,686 oz.

(2) 66.094 in.

(3) 20 in.

(4) 2.54 cm.

(5) 30.49 × 25 = 762.24 cm.

(6) (a) 6 miles; (b) 9.653 km.

(7) 23 land miles.

(8) 123 circ. in.

(9) 115.14 sq.m. = area.

(10) (a) 27.320 liters; (b) 1,848,000 cu.in.

(11) 1728 cu.in.

(12) 366.2 long tons.

(13) (a) 185 ft. (b) 13.6 ft.

Correction: The answer to No. 3 of the "Example for Practice" in the Engineers' Study Course III, as given in the Nov. 26 issue, should have been 4750 $\frac{75}{761}$ lb. instead of 2790 $\frac{5}{1805}$ lb.

[In order to make the Engineers' Study Course still more interesting, John Ketcham, of Diamond, Ohio, suggests that those following the course communicate with others who are following it. In this way papers could be exchanged and questions asked and answered by those who club together.

Mr. Ketcham thinks that an exchange of papers and ideas in this way would make the course very helpful to those who find it difficult to master the subject of mathematics.

POWER will be pleased to print the names and addresses of those who would like to take up Mr. Ketcham's suggestion. Those who desire to do this should send in their names and addresses and from the printed list select the names of the persons with whom they wish to communicate.—EDITOR.]

Hardened Copper

Our best knowledge of early man and the materials and facilities with which he labored indicate that before he learned to make, harden and sharpen steel, he was mainly dependent upon copper for cutting tools. We think of copper as one of the softest of the metals and one of the last that we should expect to take a cutting edge, and much speculation has been indulged in upon the "lost art" of making it do so.

As a matter of fact, there is no known nor conceivable way to make pure copper other than as it is, nor to doctor it so that its natural ductility will not assert itself when it is subjected to pressure or impact. The cutting tools of the ancients, which apparently did take very fine edges, or which were adequate for such heavy work as stone cutting, etc., were made not of copper but of an alloy of copper, tin and perhaps other elements. The tip of a copper soldering iron will sometimes become so hard that a file will not cut it, through its absorption of other metals with which it comes into contact in its heated condition.

The writer hereof has been using for some time a pair of desk shears, presented to him by S. R. Dawson, of 72 Broadway, New York, and made out of what Mr. Dawson calls "hardened copper," but which he acknowledges is a copper bronze consisting of copper, tin and other ingredients. Their color is that of yellow brass and they hold both it and their edge nicely. Mr. Dawson manufactures a long line of cutting tools, including razors and surgical instruments from this bronze and they take and apparently hold smooth, fine edges.

It is, however, more with reference to its use in engineering applications than in the cutlery trade that our readers will be interested in hardened copper and these applications promise to be numerous. It is claimed that it makes an excellent bearing material, that connecting-rod brasses made of it may be run directly upon the pins without danger of cutting or freezing and several of the large railroads and manufacturing companies are using it successfully in places where metals must move in contact at considerable speeds or under considerable pressure. The inventor would be pleased to communicate with any designer or manufacturer who is involved in a difficult bearing proposition, and he has been able to satisfactorily solve the problem in a number of cases.

Oklahoma City's new $500,000 federal building, which was recently opened, is electrically equipped throughout. The lighting load amounts to 45 kw. and there are about 50 hp. in various motors used for driving, heating and ventilating systems, vacuum cleaners, pump, air compressors, elevators, etc.

Mercury Column Measuring Pressure

By A. O. Doane

Where great accuracy is needed mercury columns are useful. This article describes the operation of U tube and cistern types and their application in measuring steam and gas pressures.

Before the spring type of pressure gage was introduced the mercury column was generally used to indicate pressures. The convenience and cheapness of the spring gage has resulted in the almost complete disappearance of the mercury instrument, except in laboratories and for scientific work, and for use as a vacuum gage.

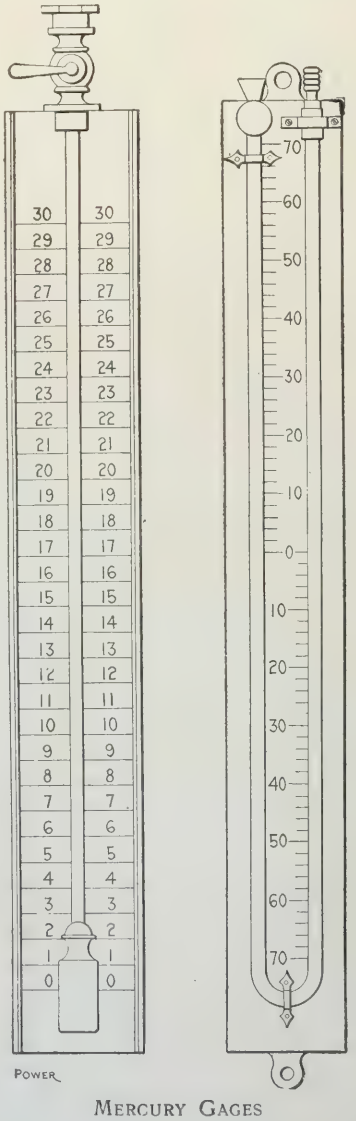
Where great accuracy is necessary, such as in engine, waterwheel and pumping-engine trials, the mercury column has a useful field. It is very sensitive and having no mechanical devices to get out of adjustment, its accuracy is not subject to the variation which is an undesirable feature of the spring gage when used for fine work.

The two principal forms of the mercury instrument are the double-column or U-tube and the single-tube or cistern type. The former is the simplest, consisting of a glass tube bent in the form of a U, or as sometimes made, two glass tubes connected by a hollow iron base piece. The double-tube form is largely used as a vacuum gage in connection with condensing engines.

Mercury is poured into the instrument until it rises in the tubes to the proper height for the range of pressure to be indicated. The scale is commonly graduated with the zero at the middle and in inches above and below this point. The scale is adjustable and the zero mark is set opposite the tops of the mercury columns in the tubes, which are at the same level when there is no pressure on the gage. In operation, one column goes down and the other up, and the difference in level between the tops of the columns is read from the scale in inches of mercury.

The cistern type of instrument consists of a closed reservoir of mercury from which a single tube rises, the bottom of the tube being very close to the bottom of the reservoir. The scale is divided so that the zero shall be the surface of the mercury in the reservoir. When pressure is applied the mercury rises in the tube, and the length of the column above the surface of the mercury indicates the pressure. As the mercury is forced up in the gage tube, the level of the mercury surface in the reservoir falls, and this causes a variation in the elevation of the zero point on the scale, depending upon the amount of the pressure and the ratio of the areas of the reservoir and the gage tube at the point where the upper surface of the column is observed.

There are several ways of dealing with the difficulty caused by this variation in the position of the zero point. One is to compute the interval of the scale, to allow for this variation; another to provide some means, such as glass windows in the walls of the reservoir or a gage-glass fixture attached to the pot in such a way that the level of the mercury surface may be observed. The scale then has an adjusting device, so that the zero mark may be placed level with the mercury surface. If the pressure varies considerably and frequently, this way is undesirable and the first is preferable.



MERCURY GAGES

Still another way is to use a fixed zero point at the level of the mercury surface when there is no pressure on the gage. The scale is graduated in both directions from this point and the readings showing the elevations of the upper and lower mercury surfaces are added to obtain the total length of the column. A gage-glass or other device for observing the elevation of the mercury in the reservoir must be provided.

The length of the graduations of the scale, taking into consideration the drop of the mercury surface in the reservoir as the mercury rises in the gage tube, may be computed by using the following formula:

$$X = \left(\frac{1}{S} \times \frac{1}{\left(1 + \frac{a}{A}\right) - \left(\frac{a}{A} \times \frac{1}{S}\right)} \right) \times 12$$

in which

X = Length of the interval on the scale in inches corresponding to 1 ft. of water;

S = Specific gravity of mercury;

a = Internal area of the glass-gage tube in square inches;

A = Internal area in square inches of the mercury reservoir minus the transverse area of metal in the iron pipe projecting into the mercury in the reservoir and connecting with the glass-gage tube.

The specific gravity of mercury is taken as the ratio of the weight of a unit volume of mercury at the temperature of the column to the weight of an equal volume of water at the temperature of the water, which is the source of the pressure the gage is to measure. As the temperature of both water and mercury is subject to variation, it is well to assume temperatures which will represent as nearly as possible average conditions, and, if greater refinement is desired, to correct the readings for the actual temperatures prevailing at the time the observations are taken.

If the length of scale interval for 1 lb. pressure is desired, the length for 1 ft. of water is multiplied by the quotient obtained by dividing 144 by the weight in pounds of a cubic foot of water at the temperature used in obtaining the specific gravity of the mercury.

The best authorities give the specific gravity of mercury at 32 deg. F. compared with water at the point of maximum density 39.2 deg. F. as 13.5956, and the coefficient of cubical expansion per degree Fahrenheit as 0.00010055. From this the specific gravity at any other temperature may be computed.

For ordinary ranges of temperature the specific gravity will vary from 13.56 to 13.59. Absolute accuracy in practical work is difficult to obtain, as the mercury may contain impurities or the density of

the water may vary, owing to dissolved minerals or absorbed air or other gases.

If it is desired to measure steam or gas pressure, the specific gravity is taken at the temperature of the column of mercury, and the scale may be graduated in pounds per square inch. For steam, due allowance must be made for the effect of the column of condensed water which is generally present in the steam-pressure pipe. The pressure per square inch, due to the height of this column, must be subtracted from the total pressure balanced by the column of mercury, to determine the true steam pressure.

If high pressures are to be measured, the height of the mercury column may prove troublesome as the surface of the mercury in the gage-glass should be at a convenient height above the floor for observation. This may sometimes be provided for by putting the reservoir under the floor, or in a pit or pipe sunk in the ground at the proper elevation and connected with the gage-glass with ordinary ⅛-in. iron pipe. Only sufficient glass tubing need be used to cover the range of pressure expected, with some additional height to provide for surges in pressure. If connected to water-works piping, subject to water rams or large variations in pressure, an overflow pipe should be provided from the top of the glass tube, with a receptacle at the bottom of sufficient capacity to hold the entire amount of mercury in the reservoir. No metal except iron, pure aluminum, solid nickel or platinum should be used where mercury can come in contact with it.

The glass tube may be secured to an iron socket, made to screw on the top of the iron pipe, with a cement made by mixing litharge and glycerin to a paste. This should be allowed to set about 24 hours before putting the gage in service.

If the mercury reservoir is set below the surface of the floor, or in any other location difficult of access, or if no window or gage-glass is provided, the elevation of the surface of the mercury may be determined in the following way:

Take a piece of ⅛-in. iron-pipe-size brass pipe, or plain iron pipe may be used for temporary work, long enough to reach from the surface of the mercury in the reservoir to a convenient point for observation. A rubber-covered copper or aluminum wire is then passed through the pipe, allowing about ¼ in. of bare wire to project from the bottom of the pipe. The wire is securely sealed into the lower end of the pipe, so that no water can enter, and must be well insulated from the pipe. A mark is made near the upper end of the pipe which is at a known distance from the point of the wire. A stuffing-box is provided on the cover of the mercury reservoir, and the pipe containing the wire works through this and may be raised or lowered until the point of the bare wire is in contact with the surface of the mercury. To indicate this contact one or two cells of dry battery are connected in a circuit containing an instrument for indicating a flow of current, such as a voltmeter, battery tester or an electric bell. The wiring is arranged so that the circuit is closed when the point of the wire touches the mercury surface. If a copper wire is used the mercury will cling to the point and the surface at the point be raised a short distance when the wire is raised. With a little practice it is possible to determine the elevation of the mercury surface to within 0.01 in. Knowing the distance from the mark to the point of the wire, the elevation of the mercury may be easily found.

This operation may be carried out under water pressure sufficient to raise the mercury to approximately the normal level in the glass tube, in which the top of the mercury column is observed. The distance from the lower mercury surface to the mark on the pipe plus the distance from this point to the upper mercury surface in the glass tube will give the length of the mercury column balancing the pressure.

By making a mark opposite the top of the mercury column and computing what the measured length of column represents in the units used for the graduations, the scale may be conveniently and accurately adjusted.

If the pressure varies considerably while these observations are being made, the pressure valve should be shut off, leaving the mercury in the reservoir under pressure. All joints and the stuffing-box must be tight if this is done; otherwise the mercury in the tube will rapidly drop.

A pressure pipe from a water main, or discharge pipe of a pumping engine should never be connected to the top of the pipe but always to the side or lower part, to avoid the entrance of air. All summits on the pressure pipe leading to the mercury reservoir should have air cocks, which must be opened from time to time to see if air is collecting.

With considerable fluctuation in pressure, using a small pressure pipe, ¼ or ⅜ in. is advisable. Throttling a valve or cock closely, to damp the movement of the column should not be attempted, but a large diameter glass observation tube should be used; one ¾-in. internal diameter gives good results in most cases. If any throttling is necessary, a cock gives better results than a valve.

In cleaning the glass tube, no metal should be allowed to touch the interior surface, as it will cause the tube to crack. A few drops of white-wine vinegar on top of the mercury in the tube will greatly assist in keeping the glass clean and bright.

If the mercury gage is correctly installed and given the small amount of care it requires, it will be found very accurate and satisfactory for work to which it is especially adapted.

Causes of Notable Boiler Explosions

Probably the most disastrous explosion which ever occurred was that of a horizontal tubular boiler in a large shoe factory at Brockton, Mass., on the morning of Mar. 20, 1905. One part of the building became a mass of ruins, which took fire, and 60 of the 400 persons employed in the factory were either immediately killed or buried alive, and over 100 were more or less seriously injured.

Investigation revealed that the boiler burst as a result of a defect known as a lap-joint fracture. This is one which extends along the longitudinal seams of lap-jointed boilers, and is due to imperfect rolling of the plates when curving them. Sometimes the last few inches at each end slipped off the rolls and did not become bent to the true form. Then they had to be forced to the desired radius, which was often done with sledge hammers, the plate being placed over a "set."

Such obviously severe treatment seriously stressed the plate, and as, in addition, the true form was not obtained even after treatment, the subsequent riveting of the ends together tended to set up fracture. Some ignorant or reckless boiler maker would drill or even punch the holes prior to bending, and, under such circumstances, it is no wonder fracture developed. Unfortunately, the lap-joint fracture usually occurs on the inside of the overlapping plates, where it cannot be detected, and hence may develop while working until the boiler is weakened to the bursting point.

The boiler which failed at Brockton was periodically examined by an insurance company, and at the last inspection no defects were found, so that the boiler was apparently safe for the stipulated working pressure. This explosion did much toward doing away with the lap joint, and in the present best practice only butt joints are used for the longitudinal seams. With such joints the true cylindrical form of the shell is retained and any tendency to straining, either in constructing or working, is avoided. Where lap joints are still used, rolls are employed which are capable of rolling the plates clear to their ends.

Another disastrous explosion was that which occurred on board

H. M. S. "THUNDERER"

in 1876, when 45 were killed and 33 seriously injured. This accident was due to the safety valves becoming inoperative, through faults in workmanship, such that when the wings expanded with the heat they became jammed. Probably this explosion would have been averted.

if the boiler pressure gage had had a stop pin at the high-pressure end of the dial. At the time important steam trials were being conducted, and in the excitement and bustle, it escaped notice that the pressure-gage pointer went completely around the gage and on past the zero mark again, so that it registered an apparently low pressure, when the pressure had reached the bursting point.

An account of a terrible boiler explosion appeared in the *Vulcan* (the monthly organ of the Vulcan Boiler & General Insurance Co., of Manchester, England) for August, 1905. It stated that one of the boilers on board the

U. S. Gun-boat "Bennington"

which was lying in the harbor of San Diego, Calif., burst and was immediately followed by the explosion of the powder magazine. All on board were either killed outright or horribly maimed, the dead and missing amounting to 60 and the injured to 75. According to this report, one of the port boilers had become defective and been recently repaired, and was supposed to have failed first, causing the others to fail and igniting the powder magazine.

Shamokin Explosion

Many instances are on record of whole ranges of boilers exploding, but in no case were the results so disastrous as those above related. On Oct. 11, 1894, at Shamokin, Penn., 27 boilers out of a range of 36 burst, but fortunately only five persons were killed and a few injured. These were very long plain cylindrical boilers externally fired. This type is especially likely to fail from a defect commonly known as a seam rip. Being externally fired, the bottom of the shell is directly exposed to the intense heat of the furnace gases, while the top remains comparatively cool. The bottom consequently expands much more than the top, excessively straining the circumferential seams, which gives rise to fractures or seam rips extending for various distances round the seams. Fortunately, seam rips usually occur when the boilers are being cooled down for cleaning or inspecting, but occasionally they occur without warning when the boilers are at work, and not uncommonly the fracture extends completely around the shell; then unless the boilers are well stayed longitudinally, a violent explosion results.

Explosion at Redcar, England

On June 14, 1895, a similar explosion occurred at Redcar, in Yorkshire, England, when 11 out of 15 boilers exploded, killing 12 persons, and injuring nine others. These boilers were nearly 70 ft. long ,and the company which insured them had reported quite a number of seam rips and advised cutting the boilers into two parts and lashing the ends together by longitudinal stays to minimize the risk of failure. Unfortunately, the advice was not acted upon. The danger of these long externally fired boilers has been long recognized and happily this type is now fast becoming obsolete.

Another Multiple Boiler Explosion

occurred at Friedenhutte, Germany, in July, 1887, when a whole range of 22 boilers gave way, killing 12 people and injuring 35 more. These boilers were of the ordinary elephant type, fired by blast-furnace gas. It is supposed that an explosive mixture of blast-furnace gas and air collected in the flues and became ignited and that the shock from this explosion caused the boiler to let go.

A Water-tube Boiler Explosion

Water-tube boilers are commonly supposed to be immune from serious explosion, but this is fallacious. In December, 1907, at St. Louis, a whole range of seven such boilers exploded, six of them being blown to pieces, while the remaining one had a large number of tubes blown out. Eight people were killed and 21 seriously injured. According to *Locomotive,* one of the tubes of the first boiler burst, but whether as a result of scale, overheating, or other defect it is not known. This tube bursting probably tore out the others, until the whole boiler was blown to pieces, which doubtless led to the explosion of the next boiler, and so on.

At present boiler explosions are less frequent than before the advent of the various boiler-insurance and inspection agencies, and the more vigilant inspecting of boilers, both during construction and working. Really disastrous explosions are comparatively rare, and such as do occur, are usually from carelessness and neglect, or even recklessness, as was the case with the explosion at York, Penn., Aug. 10, 1908. This explosion, a full account of which was given in Power, Aug. 25, 1908, was due to sheer recklessness. The boiler was an old one, the plates were badly wasted and cracked, neither pressure nor water gage was attached, and the safety valve was practically useless, having been known to blow off but once in 18 years. A boiler worked under such conditions should be expected to explode. Steam boilers now are rarely worked without suitable fittings, and generally are periodically examined by experts, so that defects are discovered and corrected before they become serious.

Every year, knowledge of both the construction and working of steam boilers becomes more and more thorough, and with the exercise of care and skill on the part of those operating the plant, together with periodical and vigilant inspection by experts, boiler explosion should soon become almost a thing of the past.

A little book entitled "The Distington Multiple Boiler Explosion and Other Multiple Boiler Explosions," recently published by the chief engineer of the National Boiler & General Insurance Co., Manchester, England, gives a full account of an investigation into the cause of the explosion of four boilers at a large iron works, together with an account of 17 other multiple explosions. Those interested in the subject will find it well worth reading.

Worcester Associations Celebrate N. A. S. E.'s Thirtieth Anniversary

On Saturday, Oct. 26, the members of Worcester Association No. 4 and H. E. Stone Association No. 32, both of Worcester, met in the State Mutual Building banquet hall and with their invited guests celebrated the thirtieth anniversary of the National Association of Stationary Engineers and the tenth anniversary of H. E. Stone Association No. 32. Under the direction of a joint committee, Messrs. Tolman, Lambert and Stockwell, Worcester No. 4, and Messrs. Wilder, Seed and Miller, No. 32, a delightful banquet was provided, with music.

About 100 members and guests attended, among those present being Mayor O'Connell, of Worcester; State President Maloney, Fall River; Past President Wheeler, New York; Past State President H. E. Kelsey, Lowell; State Inspector of Boilers James B. De Shayo and State Inspector Morton, of the Worcester district; H. E. Stone, consulting engineer, and Ole Petersen, state secretary.

After the repast, Toastmaster Seed introduced Mayor O'Connell, who proposed the toast, "The N. A. S. E." and spoke briefly. Speeches were also made by Past National President Kelsey, State Secretary Ole Petersen, State Inspector De Shapo; State President Maloney, Mr. Hayes, of the State Educational Committee, and C. J. Wilder.

The meeting was one of the most enjoyable in the history of the N. A. S. E. in Massachusetts.

Brush Bar Belt Dressing

"Brush Bar" belt dressing, made by the Wayo Manufacturing Co., Buffalo, N. Y., is put up in cylindrical form, one full pound of dressing to each bar. The bar is 2 in. in diameter. Each stick of dressing is made with a 2-in. leather brush end for removing dirt from a belt while running. Then the stick is reversed and the dressing is evenly applied to the belt. The brush serves as a handle to the bar of dressing, thus permitting of using the entire stick without waste.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

The Aktiebolaget Scania-Vabis, of Stockholm, Sweden, has decided to enlarge its plant. Good! We congratulate the—er—them, or it. From our ignorant point of view, to enlarge the name would be a mistake.

Ezra Semple's Silas, who is firing in a "juice factory" down to New York, writes Ezry that he's tickled because the courts have swatted the "bathtub trust." Si asseverates that New York is so ripped up and dusty that a bath, even in cold weather, is an absolute necessity. The city folks haven't anything on our Si.—*Whangville Banner.*

A newspaper calls this " a little story full of interest," though it makes a noise like a "movy": A Chicago engineer, out of work, hungry and discouraged (orchestra plays shivery music), saw a hurrying citizen drop a real cent and "scorn to pick it up"—just like that. Did the engineer pick up the cent and buy a turkey with it? NO! He bought a paper, "scanned the want ads, picked a job as a hotel engineer and went to work without saying anything." This paper should have spelled that "cent" with an "s" before the "c." Is it for this that we have engineers?

"Our leading townsman Zebulon Tucker has no equal in Four Corners when it comes to practicing economy," says the *Trumpet.* "He took his electric fan apart last week for cleaning purposes. When he got it together again he had 23 pieces left over."

Every now and again some trouble-hunter tries to upset business by prophesying hard times for us working people. Old man Trouble won't trouble us much if we take the trouble to cock one eye up at old Sol and keep a-inchin'. Ding bust a pessimist anyhow!

(The bean-oil industry in South Manchuria has greatly developed in recent years.—Consular Report.)

Don't you remember those sweet Alice days, Ben Bolt, when we oiled our "bean" before hieing away to Alice? How each spear was oilingly cajoled to make a showing? How redolent was the odor we disseminated as we entered the vine-clad cottage? Them was the happy days, Ben! Today, alas, we regretfully gaze on that ingrowing forehead and sigh: "Oh, where are the spears of yestere'en?" And oil did it, though we can't prove it.

NEW PUBLICATIONS

STEAM TURBINE DESIGN. By John Morrow. Published by Longmans, Green & Co., New York. Cloth; 471 pages, 5¾x9 in.; 150 illustrations; plates; tables. Price, $4.50.

This book deals chiefly with the design of reaction turbines with especial reference to marine application by British ship builders. There are, however, numerous articles on turbines for electrical and other land purposes.

The author first takes up the fundamental principles of design, showing the application of entropy and other heat diagrams to the subject; this is done in an exceptionally clear manner and with the absence of complicated mathematics. This is followed by chapters on design of the various parts which make up the machine; formulas are derived for calculating these parts together with numerous examples of existing installations in which the applications of the formulas are illustrated.

The Rateau and other types of regenerators are described and their application to low-pressure turbines discussed.

Under the design of propellers for turbine-driven ships methods are given for proportioning propeller and turbine speeds so that a maximum efficiency for the unit will be obtained.

Several methods of speed reduction are described, including: Melville-McAlpin gears, hydraulic and electrical transmission systems and their points of advantage are discussed.

Chapters are included on the general principles of surface condensation, condenser design, and the work is concluded by a brief description of the impulse turbine.

Anyone interested in turbines will find this book is well worth reading. It is especially valuable to students and designers of marine turbines.

LIGHT, ITS USE AND MISUSE. A primer of illumination prepared by the Illuminating Engineering Society, 29 West Thirty-ninth St., New York. Paper; 20 pages, illustrated.

This little pamphlet has been prepared primarily for the users of artificial light. With this purpose in view technicalities have been avoided and the text confined to a simple exposition of the general principles of illumination. With the aid of a number of well chosen photographs the proper and improper distribution of light is shown for different kinds of service, both in the home and in the factory, and special attention has been called to the evils attending bad lighting.

BOOKS RECEIVED

EXPERIMENTAL ENGINEERING. By R. C. Carpenter and Herman Diederichs. John Wiley & Sons, New York. Seventh edition, rewritten and enlarged. Cloth; 1132 pages, 6x9¼ in.; 636 illustrations; indexed. Price, $6.

COAL. By E. E. Somermeier. McGraw-Hill Book Co., New York. Cloth; 175 pages, 6x9½ in.; illustrated; tables. Price, $2.

PRACTICAL MATHEMATICS. Part I and Part II. By Claude Irwin Palmer. McGraw-Hill Book Co., New York. Cloth, 291 pages in the two volumes; each 4½x7 in.; tables; 150 illustrations. Price, 75c. each.

HYGIENE FOR THE WORKER. By William H. Tolman. American Book Co., 231 pages, 5x7½ in.; illustrated.

THE "MECHANICAL WORLD" POCKET DIARY AND YEAR BOOK FOR 1913. Published by "The Mechanical World," Manchester, England. Cloth; 439 pages, 4x6½ in.; illustrated. Price, sixpence.

TOOTHED GEARING. By George T. White. D. Van Nostrand Co., New York. Cloth; 226 pages, 4½x7½ in.; 136 illustrations; tables. Price, $1.25.

THE THEORY OF ENGINEERING DRAWING. By A. A. Adler. D. Van Nostrand Co., New York. Cloth; 312 pages, 5½x9 in.; 273 illustrations. Price, $2.

Large Suburban Contract at Minneapolis

The Minneapolis General Electric Co. has closed a 10-year contract for selling electric current covering the requirements of the Northern Power Co., which operates under 25-year franchises in Excelsior, Deephaven, Tonka Bay, Wayzata and a number of other communities on the shores of Lake Minnetonka, near Minneapolis.

To serve this desirable business, including at present a population of about 12,000 in summer and 5000 during winter, a nine-mile continuation of the transmission line now being extended to Hopkins, Minn., will be necessary. The Northern Power Co. is rapidly extending its lines to all of the summer resorts around the lake. It will close down its steam-generating station at Excelsior, 18 miles from Minneapolis, and many small isolated lighting plants will be displaced by electricity, as well as gasoline plants heretofore used for pumping water.

PERSONAL

D. G. Baker, formerly works manager of the Olds Motor Works, at Lansing, Mich., and of the Blake & Knowles Steam Pump Works, at Cambridge, Mass., has been appointed works manager of Carels Fréres, Ghent, Belgium, the largest exclusive builders of the Diesel engines in Europe at the present time. The plant is being doubled. Mr. Baker has just returned from a trip to Ghent, where he concluded arrangements, and will sail in a few days with his family to Europe.

The Engineers Club, of Boston, is progressing at a rate which meets the expectations of the most sanguine of its organizers. With the election of the candidates, whose names are now before the membership, the resident membership of the club becomes nearly complete. The club house is nearly finished and it is hoped that it will be in readiness for the members' use about the first of January.

Moments with the Ad. Editor

The most necessary quality in a good ad writer is that much-misunderstood quality—*imagination.*

Imagination is quite the opposite of a common conception thereof. It has nothing to do with fancifulness, with "make-believe," with dreaminess.

Imagination means insight. The great scientist who, after gathering data for years, with but a vague idea of the tendency of his investigation, suddenly, with his facts still incomplete, realizes the underlying meaning and unity of it all, as Darwin did in his formulation of the theory of evolution,—*he* has that precious quality, imagination.

The engineer who, studying for months the problem of spanning a great canyon, suddenly sees the finished bridge in his mind's eye—beautiful, complete,with his great problem solved through a new combination of engineering principles—he, too, has imagination. So the statesman who foresees the mental temper of a nation, or the great general who, from a study of circumstances that would baffle, confuse, and make the ordinary leader temporize, sees the one right stroke to win his battle, and strikes.

Imagination in the advertisement writer means insight—vision—appreciation—of the point of view of another. To imagine the "lead" that will get your special audience's attention, or the attention of the greatest number—the point of view that will attract, and the line of argument that will convince, is the one indispensable gift.

The salesman, too, imagines his buyers' view point, and adopts different styles of attack for each case.

The ad writer is compelled to do even more. He must get the *combined* point of view of many—that is, the one element in which thousands of the possible number out of thousands are at one *for his purpose.* He cannot adapt himself tactically to each individual case. He must strike the individual appeal that is most universal at the moment. In other words he, if he be successful, talks to thousands in a way that gives each a feeling that he talks to him personally.

When you see an advertisement in "Power" or some other paper that "gets you" and that is simultaneously "getting" a thousand or ten thousand other men, differing widely from you in a thousand ways, remember, that though it may have been written rapidly and easily by its writer, and may seem a "cinch" for almost anyone to do, yet would probably be the most difficult if not impossible thing for the man without the gift and the training, even if he knew the product in question from A to Z.

Like all simple things into which a special, individual, rare ability goes, it seems easy for the outsider, and is anything but that. There's an incident of a man who had a circular letter to be written. Finding that he was unequal to the task himself, he went to an advertising man, who wrote the letter and charged him $15 for the job

Quite enraged at this impudent and unwarranted piece of highway robbery, he at once appealed to one of the leading advertising authorities of New York, a man who had spent hundreds of thousands of dollars in advertising.

"Say," said he, "how long should it take a man to write a circular letter?"

"Well," replied the big advertiser, "A man of exceptional ability might possibly complete the letter in a day's time, a man of ordinary ability could write it in half a day, and any damn fool could write it in half an hour."

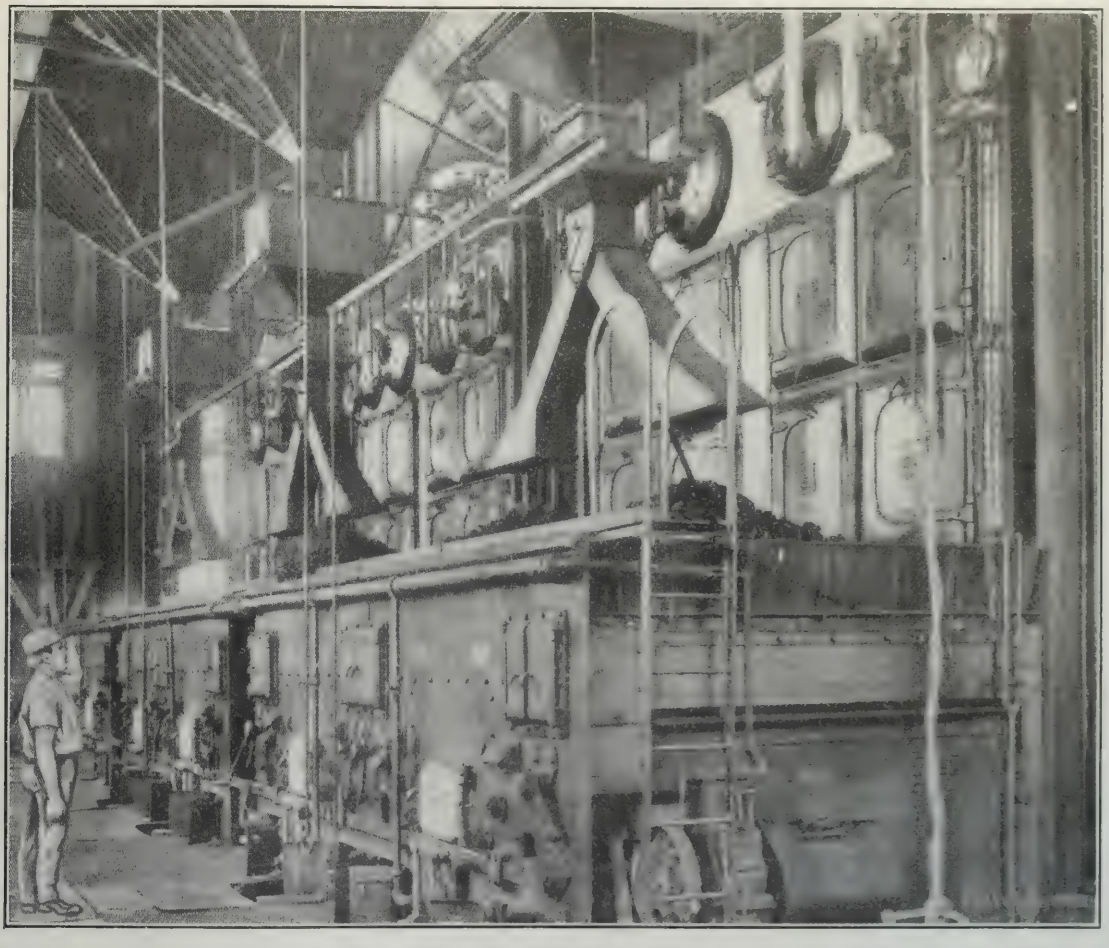
Nine-tenths of the work involved is plain "digging"—digging deep down into the proposition to get at the point of contact between product and purchaser. This is where the gift of imagination—insight—plays its part.

If you find the ads in "Power" interesting and informative, it may be, perhaps, because the men who write them have the gift and the training.

POWER

Vol. 36 NEW YORK, DECEMBER 17, 1912 No. 25

The Water Level



Did you ever experience that hair-raising, knee-weakening, heart-stopping sensation that comes when the water level in the boiler has gone out of sight in the gage-glass—gone so far below the bottom gland-nut that you didn't know where it was?

Maybe you opened the water column drain and water came. Looked good, didn't it? And you made up your mind to be more careful.

Probably there was a reason why the water got so low before you discovered it. Gage-glasses will get coated with lime, etc., and if badly so, when as far away as in the picture, it becomes difficult to ascertain the water level.

It is easy to "lose" the water when carrying heavy loads. The gage-glass is the only tell-tale. Keep it clean and renewed when dirty. It pays.

Nashawena Steam Turbine Plant

By Warren O. Rogers

One of the most uptodate power plants in New Bedford. The mill has 126,000 spindles and a slight variation below normal in the speed of the generating units is greatly multiplied at the spindles. Motor drive is utilized and the loss of output has been reduced to a minimum, due to the constant electrical energy maintained by the turbine units.

One of the latest cotton mills to be added to the large number in the city of New Bedford, Mass., is the Nashawena, which at present has 126,000 spindles and a floor space for a total of 165,000.

The plant necessary to provide power and heat for this mill consists of sixteen 300-hp. vertical boilers set in two rows of eight each, Fig. 1. These boilers are in batteries of two and set in two rows forming a wide alley between them, thus providing space for steam and water mains, also the main smoke flue, which runs above the boilers, Fig. 4, to the outer end of the boiler room to a 260-ft. concrete chimney.

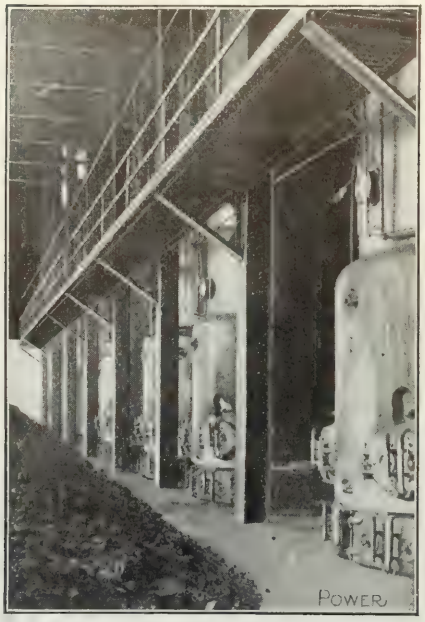


FIG. 1. ROW OF EIGHT BIGELOW-MANNING BOILERS

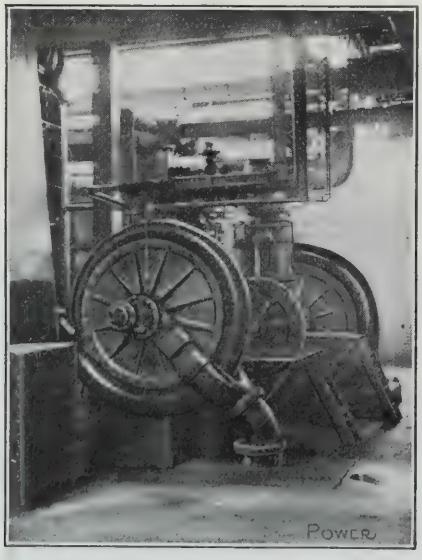


FIG. 2. CIRCULATING PUMPS

PIPING

Two 14-in. steam mains running to a header through this alley, are suspended at a height to bring the 19-in. steam header, which runs at right angles with the two 14-in. mains, low enough for the branch pipes leading to the turbine to come just below the turbine-room floor; the branch pipes are then brought up and connected with the turbines.

A 7-in. steam pipe from each boiler drops from a side connection in the steam main. An angle, nonreturn valve serves to cut each boiler out of the line when necessary.

Brass main and branch feed pipes are used. The main 6-in. feed pipe is suspended midway between the boiler rows, and 2-in. branches connect with the boilers. This arrangement of piping has the advantage that the same structural work is used to support both the steam mains and the feed line. The 2½-in. boiler blowoff pipes connect with a 4-in. main blowoff line placed in a trench, between the boilers. It extends through the boiler room to the water front. Fig. 3 shows a general plan of the plant piping.

A grated platform runs in front of the boilers on the firing side, and a wide grating has been built on the opposite side between the two rows of boilers. It is supported by the same metal work from which the piping is suspended. The main steam pipes are fitted with stop valves, operated from the floor by chains and sprocket wheels.

At one side, occupying a corner formed by the walls of the turbine and boiler

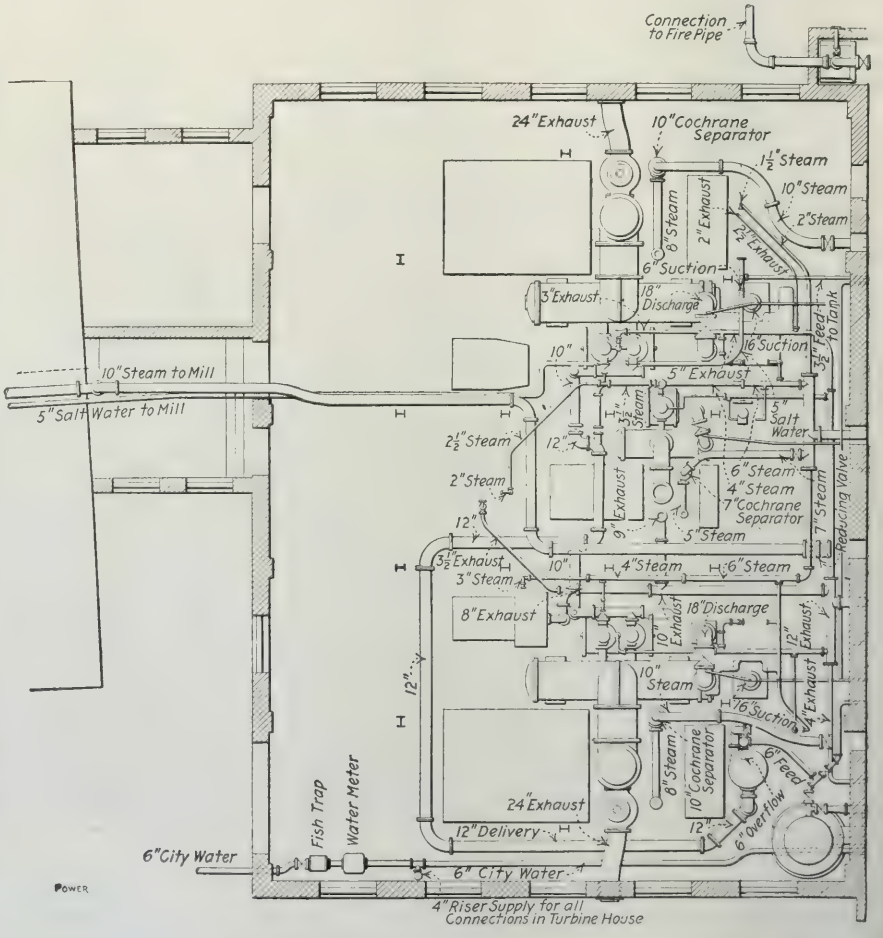


FIG. 3. PLAN OF THE POWER PLANT OF THE NASHAWENA

rooms, is the pump room in which are two 14 and 10 by 18-in. outside-packed, boiler-feed pumps, and one injector. These boiler feeders are so piped that water can be taken from the hot-water tank under a water-weighing machine through which all the return and make-up water flows, or it can be taken direct from the city 6-in. main. The returns from the condensers come into a hot-water tank through a 12-in. pipe, and are pumped into a 3000-hp. open feed-water heater in the boiler room. The hot water goes from the heater to the water weigher, feed pumps and boilers in succession. The returns from all traps and auxiliaries are piped to the hot well from which they are pumped to the heater from which it flows to the water weigher by gravity.

Electrical Units

Three turbines generate the electrical energy. Two of them are of 3000 kw. capacity, at a speed of 1800 r.p.m., and one is a 500-kw. unit, at 3600 r.p.m. All generators deliver three-phase, 60-cycle current at 600 volts. These units receive exciting current from a noncondensing 100-kw. turbine, directly coupled to a direct-current generator, which runs at 2400 r.p.m. and is used for starting up in the morning. When the main turbines are up to speed, a 150-hp. induction motor, directly coupled to a direct-current generator carries the load during the running period.

There is also a 20-kw. turbine directly coupled to a direct-current generator. It is used with the small 500-kw. turbine unit. Fig. 5 shows two views of the turbine room.

Condensers

The surface condensers are in the basement between the turbine foundations. The two larger ones have a heating surface of 7000 sq.ft.; the small one, 1650 sq.ft. The condenser circulating pumps and air pumps, Fig. 2, can be seen from the turbine floor through an opening.

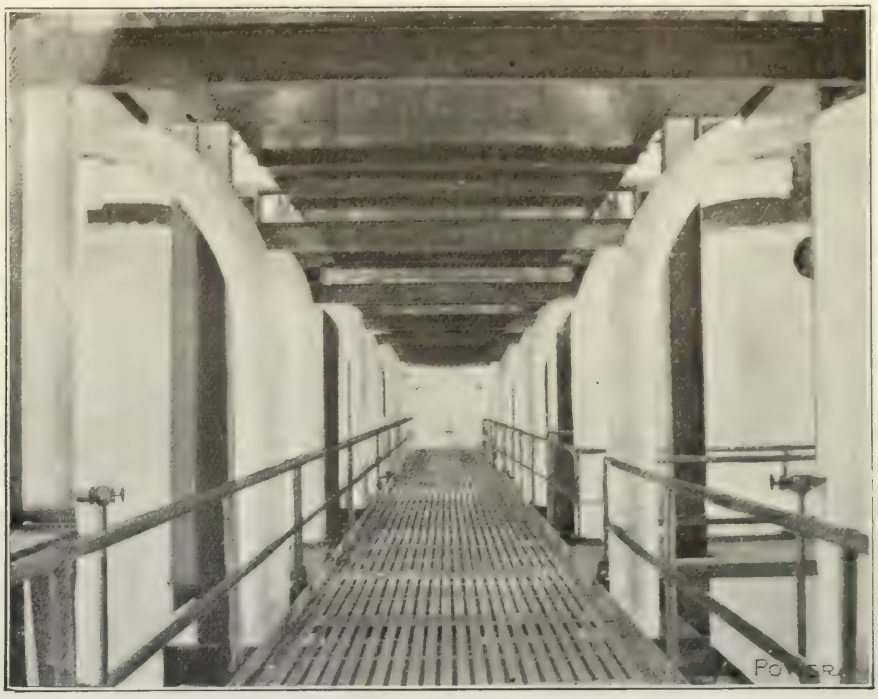
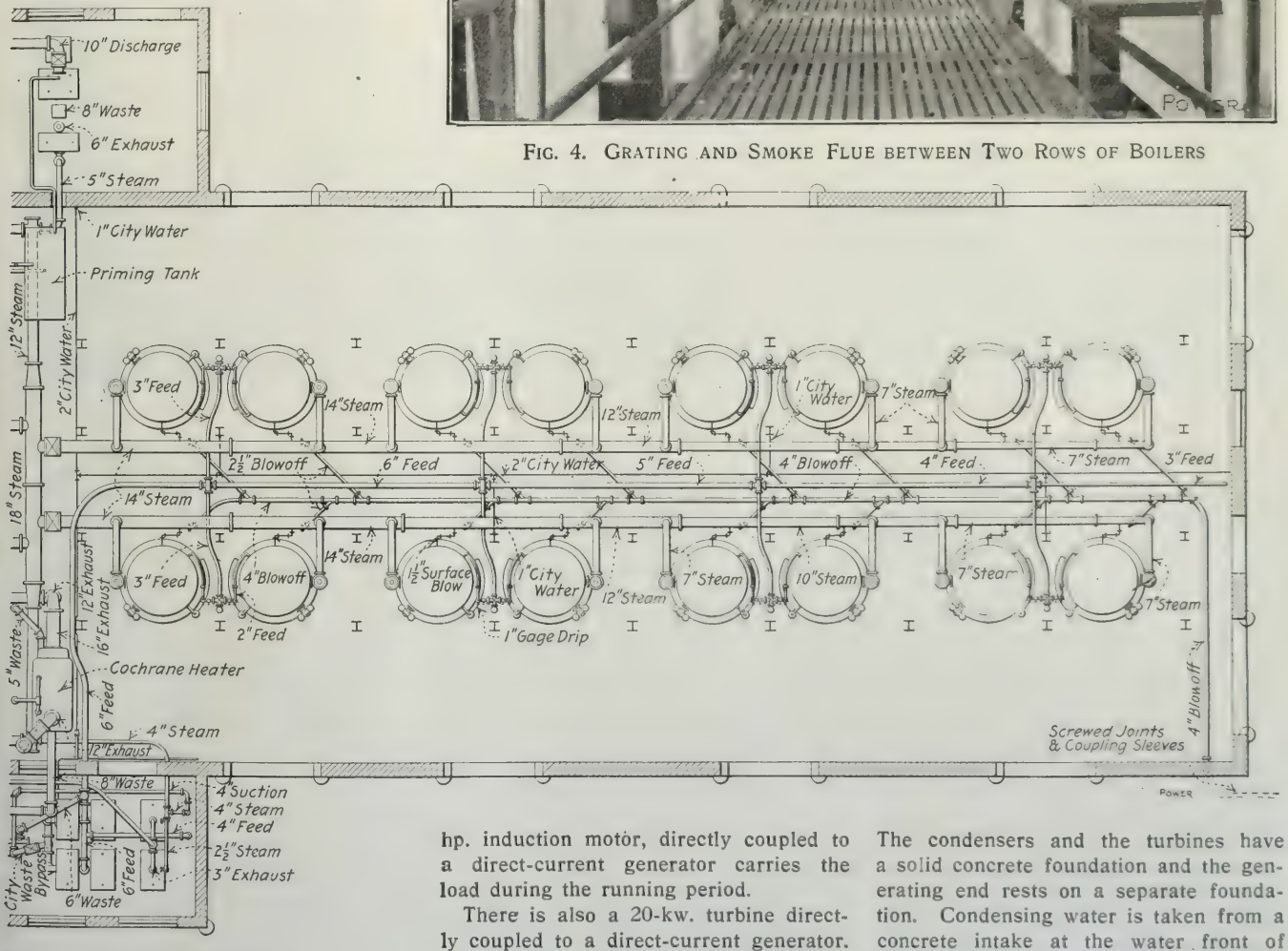


Fig. 4. Grating and Smoke Flue between Two Rows of Boilers



Mill, New Bedford, Mass.

The condensers and the turbines have a solid concrete foundation and the generating end rests on a separate foundation. Condensing water is taken from a concrete intake at the water front of the harbor, by the circulating pumps in the basement, through a 16-in. suction pipe.

A 15-ton, hand-operated crane travels the length of the turbine room. A removable floor at one corner of the room, permits of hoisting parts of machines from the basement, which is on the ground level.

EQUIPMENT OF THE NASHAWENA MILL POWER PLANT

No.	Equipment	Kind	Purpose	Kw.	Volts	Amp	Phase	Cycles	R.p.m.	Size	Pressure Lb.	Hp.	Manufacturers
16	Boilers	Vertical	Steam gen.								150	300	Bigelow Co.
2	Pumps	Warren	Boiler feed							14x10x18″	150		Warren Steam Pump Co.
1	Weigher	Willcox	Water										Willcox Engineering Co.
1	Heater	Cochrane	Feed water										Harrison Safety Boiler Works
1	Turbine	Horizontal	Gen. unit	500					3600		150		Allis-Chalmers Co.
2	Turbines	Horizontal	Gen. units	3000					1800		150		Allis-Chalmers Co.
1	Generator	A. C.	Gen. unit	500	600		3	60	3600				Allis-Chalmers Co.
2	Generators	A. C.	Gen. units	3000	600		3	60	1800				Allis-Chalmers Co.
1	Turbine	Curtis	Exciter	100					2400		150		General Electric Co.
1	Generator	D. C.	Exciter	100	125	800			2400				General Electric Co.
1	Motor	Induction	Exciter				3	60				150	General Electric Co.
1	Generator	D. C.	Exciter	100	125	800							General Electric Co.
1	Turbine	Curtis	Exciter	520							150		General Electric Co.
1	Generator	D. C.	Exciter		125								General Electric Co.
1	Condenser	Surface	Turbine							1650 sq.ft.			Wheeler Condenser & Engineering Co.
2	Condensers	Surface	Turbines							7000 sq.ft.			Wheeler Condenser & Engineering Co.
3	Pumps	Edwards	Air										Wheeler Condenser & Engineering Co.

The floor of the turbine room is of hard maple, and strips of rubber matting are placed where walking is necessary. The walls are white vitrified brick with nile-green trimmings.

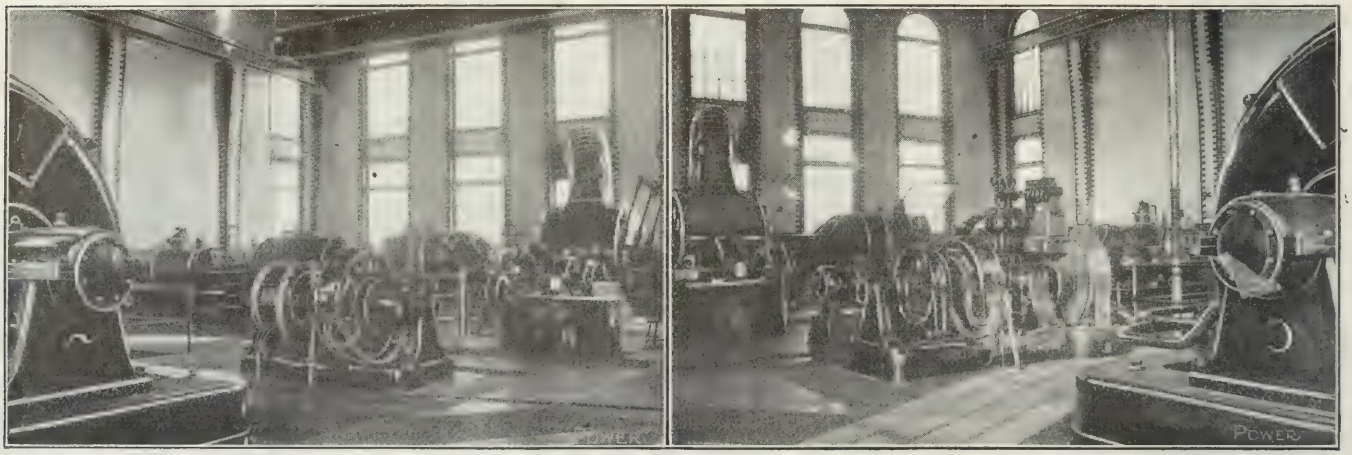


FIG. 5. TWO VIEWS OF THE TURBINE ROOM

The switchboard runs lengthwise of the turbine room; the cables drop down to the basement and lead to the factory building through a tunnel, which also contains the steam and water pipes.

Corrosion

In commenting upon the memorandum annually addressed to the Manchester Steam Users' Association by its engineer, C. E. Stromeyer, which this year deals largely with the subject of corrosion, *The Engineer,* of London, broadens out into the following consideration of the general subject.

But the more we study corrosion the more we are forced to the conclusion that it is a very complex and very variable phenomenon. It can only in rare instances be dealt with in the laboratory and then generally only in cases where the chemical reactions are so pronounced as to leave no doubt about their nature. It must be remembered that the user of a vessel of iron or steel for industrial purposes calls all the eating away of such vessel corrosion. He does not distinguish between the direct dissolution of the metal, say by a strong acid, and the slow oxidizing effect produced by water and air. In the first case the chemist has not much difficulty in putting his finger on the trouble, while in the latter, as, for example, in the unequal corrosion of ships' bottoms or propellers or condensers, he is faced by problems so complex that no results worthy of consideration are to be expected from the test-tube experiments. In such cases the only hope is to work on a large scale and under conditions as nearly as possible the duplicate of those actually existing. For example, the condenser with which trials are now being made by a committee of the Institute of Metals will certainly teach us more about the corrosion of condenser tubes than the whole series of tests made by hanging up little bits of alloys in beakers on a laboratory shelf.

The chemist takes a piece of pure iron first for his experiments. It is washed free from every trace of grease, ignited to drive off occluded gases, and handled with consummate skill and care, is introduced into some pure reagent, the action of which it is desired to investigate. The steel of bridges and boilers is stained by a thousand hands, is battered by hammering and bending till it is full of strains and stresses, and above all it is a most complex structure of iron and other things added voluntarily or involuntarily.

We doubt if anyone who has not made the attempt for himself can conceive the irregularity of even the purest iron. By using a delicate galvanometer it has been found that two identical fragments of almost pure Swedish iron would give a marked deflection in ordinary tap water. A touch on the surface of one piece is sufficient to deposit grease from the hands which reverses or still further upsets the balance. The presence of a little oxide on one while the other is clean is strongly indicated. With ordinary commercial iron and steel the results are far more marked and far more uncertain. Let anyone cut a small piece of steel from a boiler plate, divide it in two, plunge the parts in ordinary water at ordinary temperatures and couple them up to a fairly delicate galvonometer. They will undoubtedly give a deflection.

Whether a galvanic action is inseparable from corrosion or not is a debatable point which may here be neglected for we know, at least with certainty, that where there is galvanic action there is wasting away, and that is all we require to know. In a boiler a score of things are present that upset the balance and initiate corrosion. To begin with, the plates have natural variation in composition, then here and there mill scale—

a very powerful positive element more active than copper in a copper-zinc couple—adheres to the plates; again, every rivet hole is surrounded by a border of highly stressed material which probably gives a marked electrical reaction with neighboring less stressed portions. The flanged ends, particularly if they are flogged into place—as too many are—with sledge hammers and dollies, are full of stresses at the corners, stresses which there is little doubt upset the electrical stability. Finally, when under steam, there is the breathing and working of the plates, all offering conditions which invite the production of corrosion. It is with a complexus like this that the investigator of boiler corrosion has to deal, and we say without fear of contradiction that no laboratory experiment can be conceived that will give any information that is not better won from tests and examination and careful thinking about boilers themselves. When, moreover, the tests are carried out on little specimens hung in beakers, they are more than ever valueless for practical application. Plates of at least a foot square should be used so that the uncertainty caused by the difficulties of removing oxides before weighing may be at least reduced.

Melting Points of Firebrick

In bulletin No. 10, C. W. Kanolt, assistant physicist of the Bureau of Standards, describes a series of tests recently conducted to determine the melting points of different kinds of firebrick. Fifty-four samples were used, including fireclay, bauxite, silica, magnesia and chromite brick. The melting points were determined in an electric vacuum furnace and the temperatures were measured with an optical pyrometer. The results are given in Table 1. All the bricks tested were made in the United States. Each melting point given is the mean of at least two determinations. The mean of the melting points of the 41 samples of fireclay bricks is 1649 deg.

TABLE 1. MELTING POINTS OF FIREBRICKS

Sample	Melting Point	Sample	Melting point
Fireclay brick	Deg.	Fireclay brick—con.	Deg.
1	1630	37	1650
2	1635	38	1660
3	1605	47	1725
4	1605	48	1705
5	1705	49	1715
6	1705	*53	1635
7	1700	*54	1635
8	1700	*55	1685
9	1675	*56	1605
10	1710		
11	1660	Bauxite brick:	
12	1555	19	1760
13	1635	20	1740
14	1630	44	1720
15	1655	45	1785
17	1650	46	1720
21	1615	50	1665
22	1640	51	1565
23	1660	52	1590
24	1660		
25	1715	Silica brick:	
26	1695	39	1700
27	1600	40	1705
28	1695	41	1700
29	1595		
30	1560	Chromite brick:	
31	1600	42	2050
32	1650		
33	1655	Magnesia brick:	
34	1570	43	2165
35	1650		
36	1590		

*Samples 53, 54, 55 and 56 are of the same brands as 4, 9, 7 and 17, respectively, but from different lots.

TABLE 2. MELTING POINTS OF MATERIALS ENTERING INTO FIREBRICK

	Deg		Deg
Kaolin	1740	Bauxite	1820
Pure alumina	2010	Bauxite clay	1795
Pure silica	*1750	Chromite	2180

*This is not the true melting point but represents approximately the temperature at which silica flows distinctly.

The melting points of materials important in the manufacture of firebrick were also determined and are given in Table 2.

A heavy cylinder oil should not be fed at the same rate, in drops per minute, as a lighter oil, for there is more substance in a drop of heavy than in a drop of light oil.

The Flow of Steam Through Pipes

By H. V. Carpenter*

By the use of the accompanying charts the amount of steam which will flow per minute through a pipe of given diameter with a given loss of pressure; or the loss of pressure occasioned by allowing steam of a given pressure to pass through a pipe of given size with a given velocity and kindred problems may be solved without calculation. A little study of the charts in connection with the examples given in the article will make their use easy.

*Professor of mechanical and electrical engineering, State College of Washington.

There is perhaps no phase of power-plant design in which the rule-of-thumb methods are still adhered to so commonly as in the determination of the proper size of steam pipes.

This is due, apparently, to several reasons; the commonly accepted formulas are complex, are not any too well substantiated by experimental data, nor based on sound theory; the tables given by various writers are incomplete and inconvenient. The engineer who installs pipes which are too large will seldom be criticized, for the mistake shows but slightly in the first cost of the entire plant and the large radiation losses remain unnoticed. Undoubtedly, also, many designers overestimate the importance of keeping the pressure drop in the lines low. Loss of pressure in a pipe line carrying any fluid is due to friction, of course, and results in the transformation of energy of motion of the fluid as a whole into molecular energy or heat. In a water line this heat is usually a total loss, but with steam flowing in well covered pipes most of this heat is carried on with the steam, raising its temperature or its quality above that which would otherwise result. Thus instead of being able to figure the per cent. loss of power as being equal to the per cent. loss of pressure, as is usually true for water or electricity, we find the loss is materially reduced by the return of the heat to the steam. Or, stated more definitely, the total energy of each pound of steam just after entering the pipe is,

heat energy + pressure × volume energy + energy of velocity along the pipe

Just before leaving the pipe we find that the heat energy has been drawn upon to increase the volume and also to increase the velocity of the steam, while at the same time the friction has been absorbing a part of the energy of velocity and returning it in the form of heat energy. So, if the pipe could be perfectly insulated all the energy entering the pipe would be delivered at the other end but in a slightly less available condition.

In view of the complexity of these relations when radiation losses must be considered, and the extreme difficulty of getting accurate and complete experimental data, it is not surprising that we have no very well established formulas. The following, stated first by Unwin and accepted after extensive tests by R. C. Carpenter and G. H. Babcock, is probably correct enough for ordinary conditions:

$$V = 16{,}050 \sqrt{\frac{Pd}{Ly\left(1 + \frac{3.6}{d}\right)}}$$

or

$$W = 87.5 \sqrt{\frac{P_{Vd}{}^5}{L\left(1 + \frac{3.6}{d}\right)}}$$

Where

V = Velocity of the steam in ft. per min.;

P = Drop in pressure in the length L;

d = Diameter of pipe, in.;

y = Density of the steam, lb. per cu.ft.;

W = Weight of steam delivered, lb. per min.

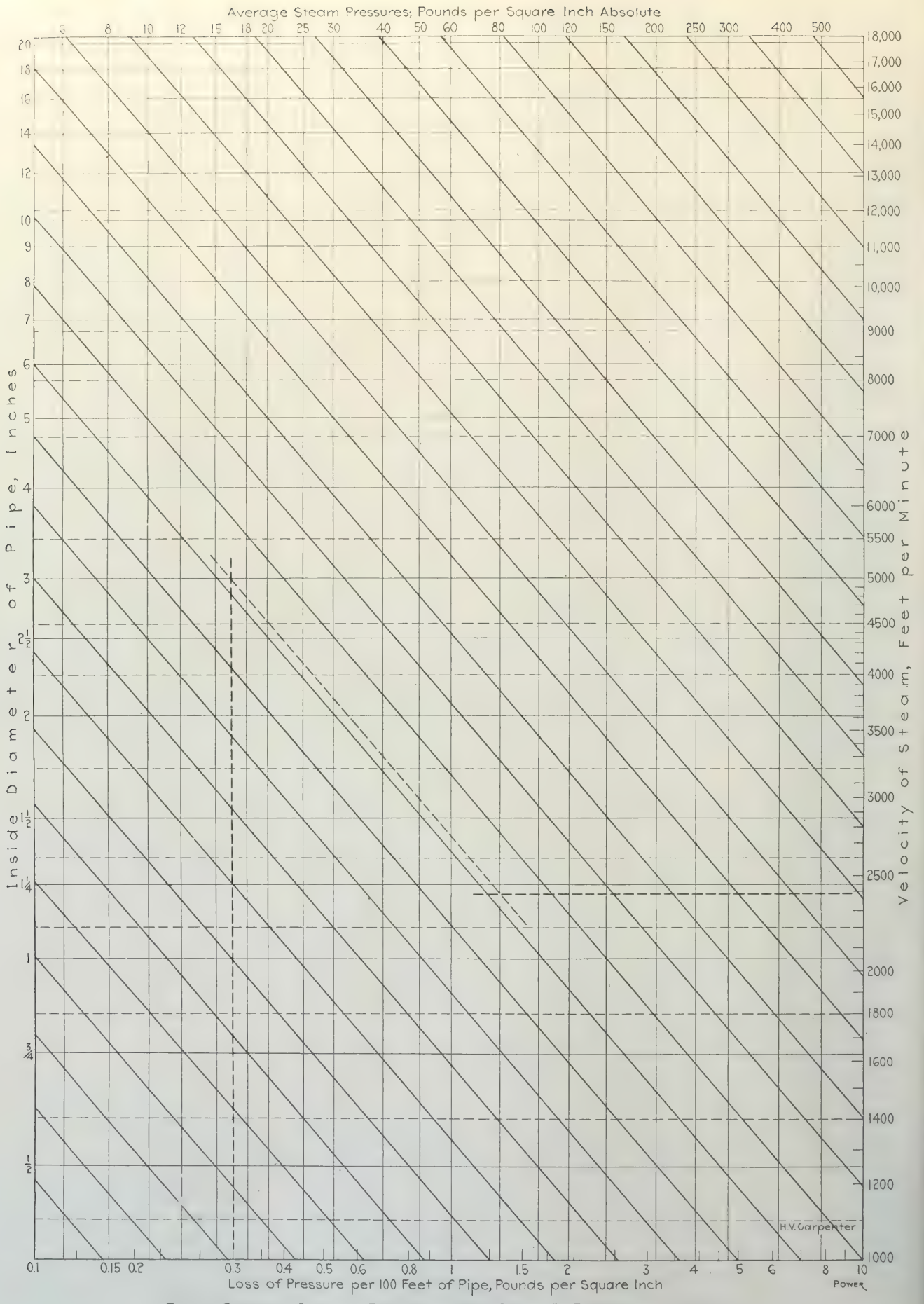


CHART SHOWING LOSS OF PRESSURE WHEN STEAM IS DELIVERED THROUGH A GOOD PIPE OF GIVEN SIZE AT A GIVEN VELOCITY

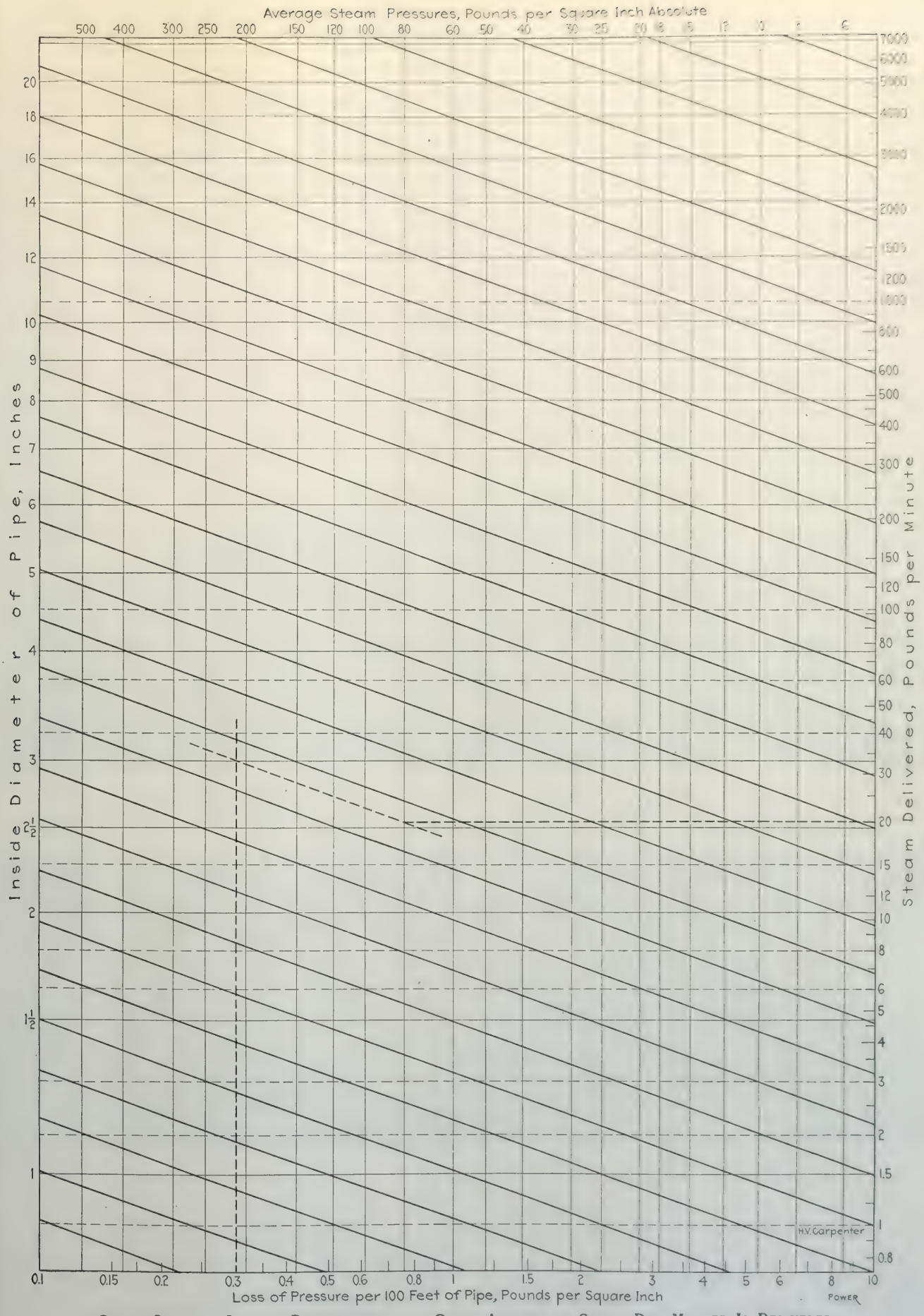


CHART SHOWING LOSS OF PRESSURE WHEN A GIVEN AMOUNT OF STEAM PER MINUTE IS DELIVERED THROUGH A PIPE OF GIVEN SIZE

In order to simplify the use of these formulas the accompanying charts are presented, the first representing the formula for V just given, and the second the formula for W. The velocity chart is given because many engineers have become accustomed to designing for an assumed velocity, usually about 6000 ft. per min. The chart shows the velocity which may be expected for any given set of conditions. For example, following the heavy dotted lines, a drop of 0.3 lb. per 100 ft. of pipe, in a 3-in. pipe, with an average pressure in the pipe of 80 lb. per sq.in., will give a velocity of 2380 ft. per min. Or, a velocity of 6000 ft. per min., with an average pressure of 250 lb. per sq.in., in a 10-in. pipe will require a drop of slightly over 1 lb. per 100 ft. of pipe. For the same conditions, however, with a 2-in. pipe the loss is more than 10 lb. per 100 ft.

These examples bring out the striking effect of the pipe size and the pressure on the loss, and show that the 6000-ft drop in pressure, and weight of steam passing per minute. For example, following the heavy dotted lines as before, with a loss of 0.3 lb. per 100 ft. in a 3-in. pipe at an average pressure of 80 lb., the quantity of steam delivered will be 21 lb. per minute. Or, with 1 lb. per 100 ft. drop a 10-in. line will deliver 860 lb. per min. when the average pressure is 60 lb. per sq.in., or a little over 1500 lb. per min. when the pressure is 200 lb. per sq.in. Or, if a 20-in. pipe is delivering 4000 lb. per min. at an average pressure of 250 lb. per sq.in., the drop in pressure will be 0.15 lb. per 100 ft. of pipe.

Regarding the accuracy of the charts, they represent the formulas exactly, except for the inaccuracies in drawing, and in reading the scales. These errors are far within the limits of accuracy needed in practice so the charts may be used with the same degree of confidence as the formulas.

As to the accuracy and range of the formulas, it seems that all of the published experiments were made with pipes of from 1.85 to 4 in. in diameter. There is little doubt that the formulas may be applied with entire safety over a much wider range than this, but the practical limits are unknown.

rule cannot be taken as safe or desirable where there is a very wide range of cases.

The second chart will ordinarily be found the more useful since it shows the relations existing among the essential quantities; size of pipe, average pressure,

Locomotive Boiler Explodes

By A. D. Nichols

A Colorado & Southern R.R. freight locomotive exploded recently while making a heavy grade, near Bunker Hill, Colo., killing the engineer and fireman.

The locomotive was an eight-driver consolidation type. The boiler was blown to one side of the track, as shown in Fig. 1, a distance of nearly 200 ft.

In Fig. 2 is illustrated a view of the boiler from the front end, showing the crown-sheet blown down through the firebox.

A close view of the crown-sheet and firebox is shown in Fig. 3. The initial rupture took place where the crown-sheet and flue sheet joined, making a clean tear in the sheet and leaving the riveted joint intact.

With but two exceptions the crown and side sheets stripped from the stay-bolts, leaving them in the outside sheets. To all appearances the cause of the trouble was low water. There was, however, a heavy coating of scale, in some parts an inch thick, on the crown-sheet. Fig. 4 is a view of the interior of the boiler taken from the firebox end. The flues were badly bent as were also some of the stays.

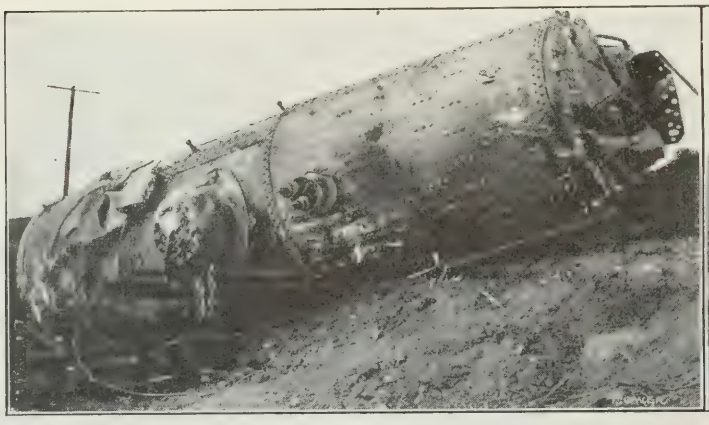


Fig. 1. Boiler Blown to One Side of Track

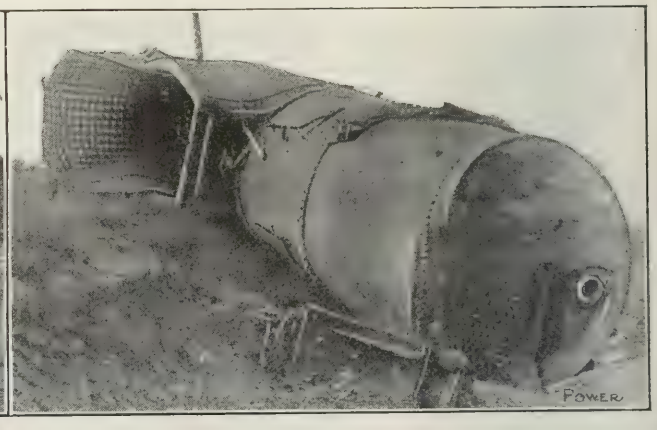


Fig. 2. Crown Sheet Blown through Bottom of the Firebox

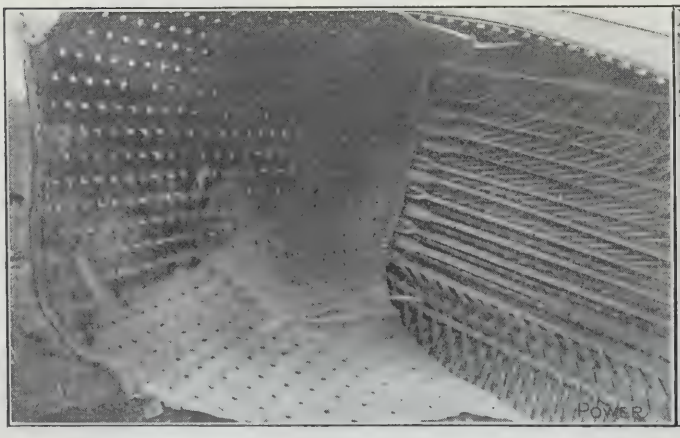


Fig. 3. Near View of Crown Sheet

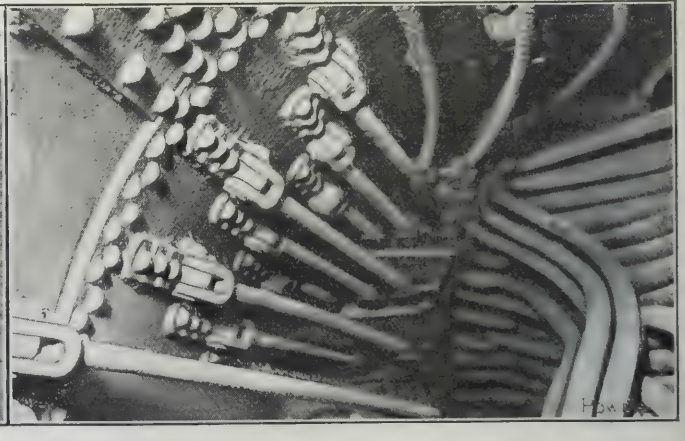


Fig. 4. Interior of Firebox

Importance of Furnace Efficiency*

By Joseph Harrington

Furnace efficiency is the controlling factor in combined efficiency and is independent of rate of combustion. A heat balance on the basis of coal as fired is necessary for a proper understanding of efficiency losses.

*Abstract of paper read before the Western Society of Engineers, Chicago.

The code of the American Society of Mechanical Engineers specifies the combined efficiency of boiler and grate as being the ratio of the heat actually used in converting water into steam, to the total amount of heat energy in a unit of dry coal, and until recently combined efficiency has been the basis for judging the performance of a boiler and furnace. It is proposed to illustrate the insufficiency of that kind of an analysis and to show that the efficiency of the elements themselves should be determined.

With the development of the modern mechanical stoker the process of burning coal is now considered the more important of the elements entering into the stoker-boiler unit. An inspection of the heat balance under the heading of "Boiler Losses," in the test-record sheet of the Green Engineering Co., will show three items. No. 61 is the heat loss due to the temperature of the gas above the temperature corresponding to the steam pressure. A study of Fig. 1 will show the rise in flue temperature corresponding to an increased boiler capacity. Inasmuch as the steam pressure was constant throughout these tests, it is evident that the rise in flue temperature is not proportional to the rate of combustion, or the developed capacity of the boiler. This signifies that the true boiler efficiency is not greatly affected by differences in the rate of heat absorption. Ability to absorb heat controls boiler efficiency, while furnace efficiency is affected by several considerations. The ability of a tube to transmit heat, when in a standard condition of cleanliness, is practically the same at all times under similar conditions.

The accompanying table shows half a dozen heat balances, which will indicate a fairly constant boiler efficiency at a fair range of ratings. The grate and furnace efficiency, however, vary to an appreciable extent.

For a proper appreciation of the disposition of the heat contained in a pound of fuel, the analysis must be on the heat contained in the fuel as it is actually fired, rather than upon the basis of either dry coal or combustible. What the engineer wants to know is how the heat contained in the actual coal he is firing is disposed of; and as the moisture contained in the coal has an influence on the efficiency of the entire process, it must be taken into consideration in an accurate analysis.

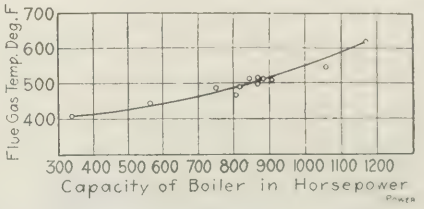


FIG. 1. RELATION BETWEEN BOILER CAPACITY AND FLUE GAS TEMPERATURE

The Middle Western coals will frequently contain as high as 15 per cent. of moisture, and lignites almost invariably carry from 25 to 40 per cent. In the latter case, a considerable portion of the total heat value of the coal is utilized in evaporating this moisture, which robs the boiler of just that much heat and reduces the net evaporation by that amount. Eastern coals exhibit the more accurate relation between the actual and dry analyses, but boiler tests with these coals should be analyzed on a basis of coal as fired just the same. The test sheet previously referred to takes into account the heat balance based on coal as fired, in which the actual heat in a pound of fuel is accounted for, the various losses being given in terms of British thermal units. An expression of these values in the form of percentages might perhaps give a clearer notion of the significance of the item. This test sheet has been developed in logical order, so that the various items can be figured in rotation, the data for the calculation of every item having been previously figured.

On account of the heat balances involving the ultimate analysis, some question may be raised as to the difficulty of obtaining this analysis in connection with an ordinary test. If the test is to be worked out in all its refinements, such analysis is essential, but it has been found that with a given coal, the ultimate analysis is so nearly constant that the heat balance is not affected materially by the slight variations that exist. If the exact origin of the coal can be established, an ultimate analysis can be easily determined. The United States Geological Survey has published a great many, identifying each analysis with a certain coal.

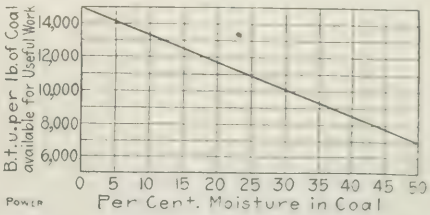


FIG. 2. EFFECT OF MOISTURE IN COAL ON AVAILABLE HEAT

In Fig. 2 is shown how moisture in the coal affects the heat available for useful work. This is plotted on the assumption that the net coal in a pound of fuel (disregarding ash for the time being, or considering the ash percentage constant) is the difference between the percentage of moisture and 100. For ex-

HEAT BALANCE PER POUND OF COAL FIRED-B.T.U.

	B.t.u. per pound	Absorbed by boiler. Water	Steam	Necessary Losses. Moisture and H_2O from H	By dry gases up to temp.	Available for unit	Highest theor. efficiency	Furnace and Grate Losses. Comb. in ash	Excess air to T_p	CO	Cinders	Available for boiler	Boiler Losses. Furnace and grate efficiency	By temp. gases above T_p	Air Leakage	Radiation and unacc't for	Boiler efficiency	Comb. efficiency
July 6, 1911..	11,027	9534	866	391	655	12,981	92.6	319	89	0	290	12,283	94.6	483	854	546	84.7	74.1
Jan. 18, 1912	13,092	8668	765	450	696	11,946	91.3	405	193	56	190	11,102	92.9	941	406	322	84.9	72.0
June 1, 1911..	14,103	9089	831	447	713	12,943	91.8	332	523	155	290	11,643	90.0	680	311	732	85.2	70.3
May 31, 1911	13,389	8557	803	427	667	12,295	91.8	557	339	121	290	10,888	89.4	576	196	856	85.2	69.9
June 15, 1911	13,370	8671	749	471	624	12,275	91.8	479	460	0	290	11,046	90.0	658	273	695	85.2	70.4
Jan. 29, 1912.	13,362	8706	734	487	707	12,168	91.1	594	175	110	290	10,999	90.4	424	349	786	85.9	70.7
July 6, 1911..								2.39	0.64	0.0	2.08		94.6	3.47	6.13	3.92	84.7	74.1
Jan. 18, 1912.								3.09	1.47	0.43	1.45		92.9	7.19	3.10	2.46	84.9	72.0
June 1, 1911..		Heat	balance	in per cent.				2.35	3.71	1.10	2.06		90.0	4.82	2.21	5.19	85.2	70.3
May 31, 1912								4.16	2.53	0.90	2.17		89.4	4.30	1.46	6.39	85.2	69.9
June 15, 1911								3.58	3.43	0.0	2.17		90.0	4.90	2.04	5.20	85.2	70.4
Jan. 29, 1912.								4.45	1.31	0.82	2.17		90.4	3.17	2.61	6.88	85.9	70.7

TEST RECORD SHEET

Report of Test on ..... Boilers Equipped With .....

Grates at ..... Owned by .....

Purpose of Test .....

Fired by ..... of ..... Date 19

1	Run from ..... to		11	Steam gage pressure ..... lb.	
2	Duration of test		12	Atmospheric pressure ..... lb.	
3	No. of boilers used		13	Absolute steam pressure ..... lb.	
4	Type of boiler		14	Draft suction at uptake ..... in.	
5	No. of tubes		15	Draft suction over fire ..... in.	
6	Diameter of tubes		16	Draft pressure under fire ..... in.	
7	Water heating surface ..... sq.ft.		17	Average temperature boiler room ..... deg.F.	
8	Superheating surface ..... sq.ft.		18	Average temperature flue gases ..... deg.F.	
9	Grate surface ..... sq.ft.		19	Average temperature feed water ..... deg.F.	
10	Ratio grate surface to water heating surface ..... 1 to		20	Average temperature steam Tp ..... Ts ..... deg.F.	

21	Name of fuel	
22	Size of fuel	

Proximate Analysis						Ultimate Analysis			Ash		Flue Gas			
H_2O	V.M.	F.C.	Ash	S	B.t.u. Per Pound	C.	H.	S.	% Comb. in Refuse	Refuse % Coal Fired	CO_2	O	CO	N. etc.
					Fired					Refuse % Dry Coal	1st Pass.			
					Dry						Flue			
					Comb.									

23	Coal burned total run actual ..... lb.			Heat Balance per Lb. Coal Fired	
24	Coal burned per hour actual ..... lb.		48	Heat per lb. Coal fired ..... B.t.u.	
25	Coal burned per hour dry ..... lb.		49	Heat absorbed by water in boiler ..... B.t.u.	
26	Coal burned per hour per sq.ft. grate surface actual ..... lb.		50	Heat absorbed by steam in boiler (superheat) ..... B.t.u.	
27	Total weight of refuse ..... lb				
28	Total weight of combustible ..... lb.			Necessary Losses	
29	Water evap. total run actual ..... lb.		51	Heat absorbed by moisture and H_2O from burned H up to Tp ..... B.t.u.	
30	Water evap. per hour actual ..... lb.		52	Heat absorbed by theoretical amt. dry gases up to Tp ... B.t.u.	
31	Factor of evaporation		53	Heat available for unit ..... B.t.u.	
32	Water evaporated per hour (f. & a 212 deg.) ..... lb.		54	Highest theoretical efficiency ..... %	
33	Quality of steam ..... %				
34	Horsepower builder's rating			Furnace and Grate Losses	
35	Boiler horsepower (Mean of Test)		55	Heat loss due to combustible in ash ..... B.t.u.	
36	Boiler horsepower (Max. hr.)		56	Heat absorbed by excess air up to Tp ..... B.t.u.	
37	Per cent. rating (Mean of Test)		57	Heat loss due to production of CO ..... B.t.u.	
38	Sq ft. water heating surface per hp. (Mean of Test)		58	Heat loss due to production of cinders ..... B.t.u.	
39	Water evaporated actual per lb. coal as fired, ... lb.		59	Heat available for boiler ..... B.t.u.	
40	Water evaporated actual per lb. dry coal ..... lb.		60	Furnace and grate efficiency ..... %	
41	Water evaporated f & a 212 deg. per lb. coal as fired ..... lb.				
42	Water evaporated f & a 212 deg. per lb. dry coal ..... lb.			Boiler Losses	
43	Water evaporated f & a 212 deg. per lb. combustible. ..... lb.		61	Heat loss due to temp. of gases above Tp ..... B.t.u.	
44	Wt. of gas per lb. Coal Fce ..... lb. Flue ..... lb.		62	Heat loss due to air leakage through boiler setting ..... B.t.u.	
45	Percentage of excess air Fce ..... % Flue ..... %		63	Heat loss due to radiation and unaccounted for ..... B.t.u.	
46	Cinders, (a) weight ..... (b) % coal fired		64	Boiler efficiency ..... %	
47	Carbon in Cinders per cent coal fired ..... %		65	Combined efficiency ..... %	

Ta = Temp. of air in boiler room.

Tp = Temp. corresponding to steam pressure.

Tf = Temp. of gases in flue.

Cb = Carbon burned per lb. actual coal

Sp. Ht = Specific heat of the steam.

Ts = Temp. of steam as superheated.

Hs = Total heat in superheated steam.

Ht = Total heat in dry steam at observed pressure.

ample, wet coal containing 20 per cent. of moisture would contain only 80 per cent. of combustible. In this way, the diagram has been worked out within the range of ordinary moisture content. Lignite coals often contain 35 per cent. of moisture, which would mean 9350 B.t.u., instead of 15,000 if the coal had not contained any moisture.

Add to the losses due to the moisture contained, the unavoidable losses due to combustible in refuse, the losses due to excess air, the losses due to radiation and unaccounted for, and the other losses which may be charged against the particular method of operation in vogue, and a point will very soon be reached where a wet coal ceases to be of any value. It has been found by experimentation that a moisture content of 25 per cent. is about the limit of practical usefulness with the ordinary form of furnace in commercial service. Lignites carrying moisture in excess of this, require a specially designed furnace wherein the heat generated by the combustion of the fixed carbon can be more efficiently brought into contact with the incoming fuel.

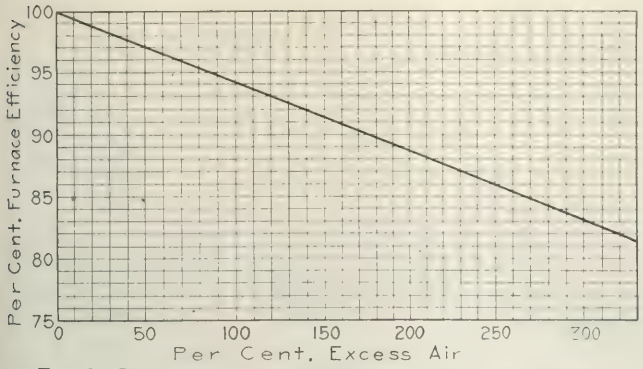


FIG. 3. RELATION BETWEEN FURNACE EFFICIENCY AND EXCESS AIR

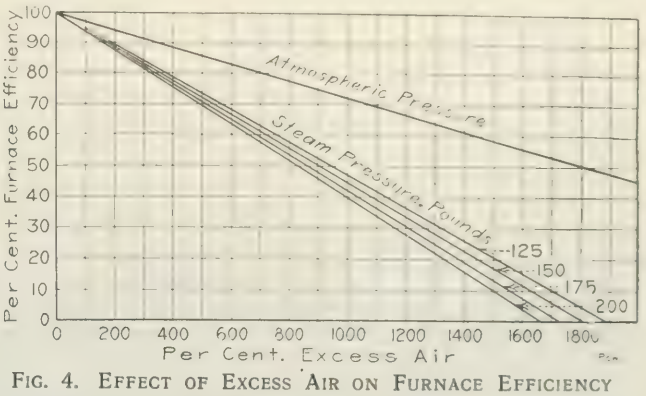


FIG. 4. EFFECT OF EXCESS AIR ON FURNACE EFFICIENCY AT DIFFERENT STEAM PRESSURES

In addition to the influence of moisture, furnace efficiency is affected by the amount of air used in burning the fuel. In many cases with Western fuels, a gas analysis of 10 per cent. of CO_2 would be considered fair practice. To maintain this figure, a fairly uniform fuel bed is required, and there must not be frequent or extensive opening of firing doors for stoking or slicing. A gas analysis of 13 per cent. of CO_2 is about the limit of economical operation with Western coals, because when the CO_2 is carried much beyond this point, CO will develop and the furnace efficiency on this account will not increase, the loss due to imperfect combustion being greater than the gain effected by reducing the excess air. A study of these various effects is given in Figs. 3 and 4.

In Fig. 3 is shown the effect on furnace efficiency of an excess of air. This curve has been carried to the ordinary extent of dilution, and illustrates the result of a leaky fuel bed or porous setting.

Fig. 4 shows the effect of varying steam pressure on the relation between furnace efficiency and excess air. This is not materially important, but it is interesting to notice that the lower the steam pressure, the less is the detrimental effect of excess air.

The design of the furnace to effect the necessary mixing, varies widely with different fuels and drafts and rates of combustion, and in the absence of any means for determining this mathematically, it appears to still remain largely a question of experience. The product of incomplete combustion is CO, wherein only about one-third of the potential heat is developed. It is, therefore, obviously important that CO must not be present in appreciable amounts.

It has been stated that there are other hydrocarbon compounds which are present in flue gases and which are carried away with the escaping gases whenever CO is present. These gases, not being of such a nature as to render it possible to detect them by the usual means, have added an element of uncertainty in the ultimate analysis of efficiency losses.

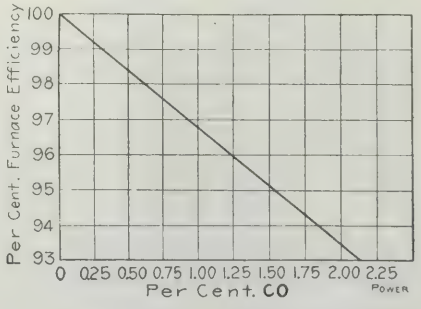


FIG. 5. RELATION BETWEEN FURNACE EFFICIENCY AND LOSS FROM CO

The author's experience, in analyzing a large number of heat balances, is that the presence of these intangible combustible compounds is quite apparent. A portion of the heat unaccounted for is undoubtedly due to this. It is doubly important, therefore, that CO in furnace gases should be held down to an absolute minimum. It is not always the case that heavy reduction in air excess must be secured before CO will appear. This is largely a question of the proper mixing in the furnace which is dependent on the size and design of the combustion space. Both CO and excess air may be present at once and the extent of this is purely a question of the capacity of the furnace to mix and retain the gases until the combustion is complete. It is, however, our experience that when CO appears, it is time to stop the reduction of air supply, even though it is quite obvious that the mixing ability of the furnace is deficient. Fig. 5 illustrates the effect of CO on furnace efficiency.

In connection with the paper the author gave the mathematics showing the method of determining the excess air and CO curves. The deductions therefrom are that each 100 per cent. of excess air affects the efficiency 5.62 per cent., and that each 1 per cent. of CO reduces furnace efficiency by 3⅓ per cent. If this were carried to the ultimate, the efficiency would be zero when the per cent. of CO reached 30, assuming that such was possible.

Lunkenheimer "Puddled" Semi-steel Valves

To meet the demand of high pressures and superheated steam, the Lunkenheimer Co., Cincinnati, Ohio, has got out a line of "puddled" semi-steel valves.

This metal is an extremely high-grade iron and steel alloy with an average tensile strength of 35,000 lb. The valves are made in two combinations C and D, to suit various conditions of superheat and to meet the specifications of engineers who differ as to the metal that should be used in the trimmings. Combination C is guaranteed for a temperature of 600 deg. F. and D for 550 deg. F. Both combinations are suitable for 250 lb. pressure per sq.in.

An English company holding a concession in Russia for constructing a hydroelectric station on the River Terek has under advisement constructing a tramway system to connect Vladikavkaz with Nalchick and Kislovogsk.

Eynon-Evans Blowoff Valve

A new type of blowoff valve is shown herewith, consisting of a yoke and body cast in one piece, Fig. 2. A nickel ring *B*, Fig. 1, is secured in the iron body, the interior diameter of the ring being the same as that of the bronze shield *C*, thus forming a continuous surface to receive the packing ring *D*. This ring is so placed in the ring *B* and shield *C* that it prevents leakage from the inlet to the outlet of the valve, and around the stem *E*. It can be adjusted or compressed by the nuts *F* to the desired density while the valve is in service.

The handwheel *G* withdraws the pistons *H* and *J* with the packing from the ring *B* into the shield *C*. The latter is operated by the handwheel *K*. This construction draws the packing into the shield before any grit, scale or other foreign matter is permitted to pass through the valve from the boiler. The lower end of the shield also acts as a valve and permits removing and inspecting the packing while the blowoff valve is in service under full water pressure.

The valve is really two-in-one, and when operated the wheel *G* is rotated, causing the packing *D* and piston *H* to rise or open until the shoulder on the follow-up piston *J* strikes the bottom of the hub *L*. The packing *D* is thus inside the shield *C* and fully protected from injury. Rotating the wheel *K* causes the shield *C* to rise until the shoulder just below the threaded section strikes the bottom of the bushing *M*. In this position the lower valve is wide open. Fig. 3 shows the valve partly open. In closing the valve the shield *C* is screwed in until it strikes the seat *B*, when the handwheel *G* is turned, forcing the packing *D* and piston *H* to the position shown in Fig. 1.

This valve is manufactured by the Eynon-Evans Manufacturing Co., Fifteenth and Clearfield streets, Philadelphia, Penn.

New Mechanical and Electrical Time Recorders

The Industrial Instrument Co., Foxboro, Mass., has added to its line of recording instruments, a mechanical time recorder, or operation recorder. The instrument is mounted in a round form of case. The pen arm is attached to and supported by a shaft or axis. An arm attached to the shaft passes through the case and an extension with adjustable clamps serves to facilitate the adjustment of the available motion to secure the desired pen travel on the chart. Clock movements are supplied to revolve the charts in 10 and 30 min., 1, 2, 3, 4, 6, 12 and 24 hours and seven days as desired.

The recorder is useful in recording the frequency, duration and extent of any mechanical motion, such as opening and closing of gates, valves or doors, etc.

An electrical time recorder has also just been placed on the market by the same company. It is operated by electric contact and consequently is adapted for use in many places where it is desirable to put the recorder at a distance from the source of motion or operation. When contact is made the current actuates an armature to which the pen arm is attached and the chart records the frequency and duration of contact. These electrical time recorders are regularly supplied with any number of pens up to six for recording independent operations simultaneously on the same chart.

Our Vast Coal Reserve

The known coal fields of the United States embrace a total area, according to the United States Geological Survey, of 310,296 square miles, to which may be added something over 160,000 square miles of which little is known, but which may contain workable coals, and about 32,000 square miles where the coal lies under heavy cover and is not considered available under present conditions. The supply of coal before mining began is estimated to have been 3,076,204,000,000 short tons, of which 1,922,979,000,000 tons were considered to be easily accessible and 1,153,225,000,000 short tons to be either so deep or the beds so thin that they are accessible only with difficulty. Classified according to the character of the coal, the original supply consisted of 21,000,000,000 short tons of anthracite,

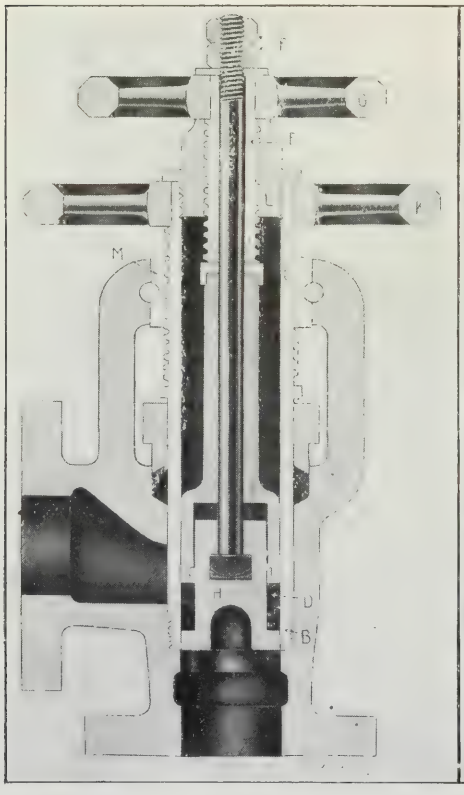


FIG. 1. VALVE CLOSED

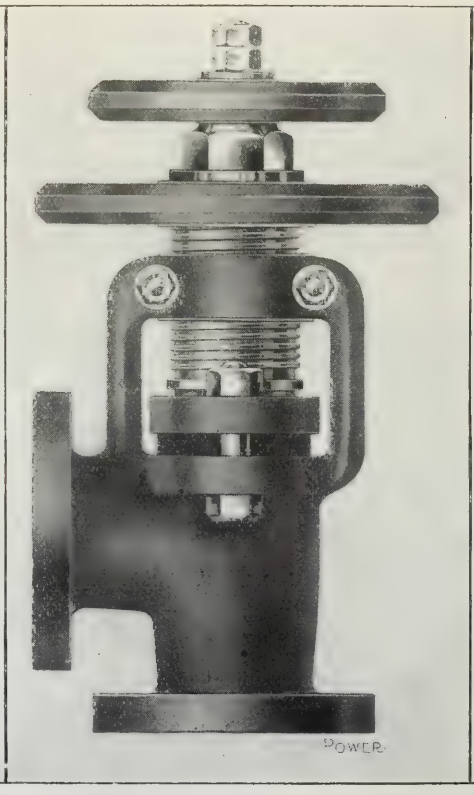


FIG. 2. EXTERIOR VIEW

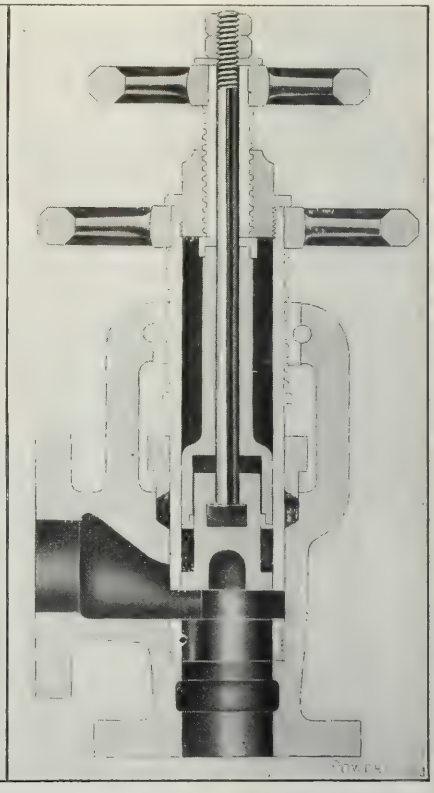


FIG. 3. PARTLY OPEN

1,661,457,000,000 tons of bituminous coal, 650,157,000,000 tons of sub-bituminous coal, and 743,590,000,000 tons of lignite.

The total production of coal to the close of 1911 has amounted to 2,270,798,737 short tons of anthracite and 6,468,773,690 tons of bituminous coal, or an aggregate of 8,739,572,427 tons. This total production to the close of 1911 represents, including the waste of coal in mining, an exhaustion of the beds equal to 14,181,980,000 short tons, or somewhat less than 0.5 per cent. of the original supply. In other words, the quantity of coal still remaining to be mined amounts to 3,062,022,020,000 short tons, or a little more than 99.5 per cent. of the original supply. The annual rate of exhaustion at the present time as represented by the production in 1910 and 1911 is 0.025 per cent, of the supply. The quantity of coal still in the ground at the close of 1910 was 6000 times the production of that year, or, estimating a half ton of coal lost for every ton recovered, the supply is equivalent to 4000 times the present annual rate of exhaustion.

Injector Troubles and Their Remedies

By R. Ewart Cleaton

These troubles are often due to scaled tubes and leaking pipes and joints. Many of them are designated and the remedy given, also the method of testing for leaks.

As the proximity of heat is detrimental to a vacuum, an injector should be placed well away from the boiler; the steam connection should be at least 9 or 12 in. long from the boiler shell. If the injector still gets warm, a hose played upon it will remedy the troubles temporarily, and a piece of asbestos millboard placed between the boiler and the injector, will also serve to keep the latter cool. The steam outlet should be connected to a high point of the boiler to insure a maximum amount of dry steam.

In most portable plants, the water supply is obtained on a lift, and care should be taken that it is not greater than that for which the injector is designed; the shorter the better, as the longer the lift the greater is the decrease in the capacity of the injector.

Water Supply

One of the most common causes of trouble arises from a dirty water supply. This can be obviated in several ways. Where the water contains a large amount of suspended matter, the use of a settling tank is advisable, from which the water can be drawn off into another tank, or, as is more often the case on contractors' plants, a barrel into which the suction pipe is introduced.

Care should be taken to see that fibrous, scaly or greasy matter adhering to the sides, is removed before inserting the suction pipe, and that the end of the latter is 12 to 18 in. from the bottom. This is especially desirable when a settling tank is not employed, for should it be only 2 or 3 in. above the bottom of the barrel the force of the suction will cause the mud to be drawn into the injector.

The suction pipe should have a fine-mesh brass or copper strainer to arrest any suspended substance in the water. A foot valve will also materially add to the ease of starting. Should the mesh become dirty and clogged, it can be cleaned by blowing a jet of steam through it, previously disconnecting the foot valve from the suction pipe.

Having made sure of a clean water supply, next in importance is the piping.

Piping

Half the trouble experienced with injectors may be attributed to either using the wrong sort of piping, or defective material.

Rubber piping should be avoided entirely, as it deteriorates very rapidly, and small pieces of rubber are detached, thereby undoing all the good that has been obtained by insuring a clean water supply. Galvanized-iron pipe should be used in preference to ordinary black pipe, as the chance of rust forming and detaching itself is eliminated. Fittings, such as elbows, etc., should be curved, not square, and preferably of malleable iron instead of cast iron, as a better thread can be obtained, thus insuring tighter connections and consequently preventing leaks.

An easy way to locate a leak in the piping is to plug up the suction end of the pipe as tightly as possible, open the water-regulating valve to its full extent having also wedged the overflow valve on the injector so that it will not open, and then turn on the steam; any leak will be plainly discernible by the jet of escaping steam. Assuming that the piping system is perfect, it will often be found that the gland in one of the valves is defective and requires repacking. Unless every gland and joint in the entire system is perfectly tight, the injector cannot be expected to give satisfaction.

When the plant is in a district having a chalky or limestone subsoil, a great deal of lime is bound to be in solution in the water, and when steam is raised this is precipitated not only in the boiler, but also in the pipe conveying steam to the injector. This in time will cause serious obstruction to the steam, thereby diminishing its velocity to a large extent, which in turn will lessen the vacuum necessary for the working of the injector. When this happens, the pipes affected should be disconnected and allowed to soak in a 10 per cent. solution of muriatic acid, for at least 12 hr. This can be done over night, and not delay the working of the plant in the daytime.

An injector will not work efficiently, with the steam passing through a pipe smaller than the inlet nozzle, and this is the condition when the pipes become encrusted with scale. The suction and feed pipes should also be the same size as the injector connections.

A priming boiler is another common cause of trouble, as water or oil in the steam supply will destroy the necessary vacuum.

Valves

The valves should be brass, of the globe type in preference to the straight-way. Special care should be taken in connecting up the check valve on the feed line between the injector and the boiler. Although this may appear extremely simple, trouble is often caused by this valve being carelessly put wrong end on. If this is done no rectification of the other defects will enable water to be forced into the boiler. If the overflow valve continuously leaks when the injector is working, it should be reground with a little flour of emery.

Capacity

The capacity of an injector is affected in many ways, such as the distance the water has to be lifted, if the supply is below the injector; the height above the injector, if the water is taken from an overhead tank; the temperature of the water supply; water pressure, if taken from a main, and the steam pressure. These conditions should be considered when buying an injector and the makers consulted as to the most suitable size. Besides stating the above, the size of boiler used should also be given.

To test the capacity of an injector, close the globe valve between the boiler and the check valve in the delivery pipe and take off the cap of the check valve and remove the check. Then place a pail under the valve and allow the injector to run for exactly one minute, timing it with a watch. The amount thus collected in the pail multiplied by 60 will then give the hourly capacity of the injector in gallons.

When to Install a Fuel Economizer

By Everard Brown

Figures are given in an assumed case showing the gain that may be expected, with fuel at different prices, by installing an economizer. Other advantages are also shown.

Under what conditions it is advisable to include a fuel economizer in the design of a new power plant is a problem demanding careful consideration. A general assumption is that for every increase of 10 deg. in the feed-water temperature 1 per cent. in fuel will be saved and, as the function of an economizer is to raise this temperature, it is sometimes assumed that the fuel saving of 1 per cent. will hold good under all conditions. This is not the case, however, as will be shown.

Assume an ordinary boiler plant of 1000 hp. operating under 150 lb. steam pressure at its rated capacity, the flue gases escaping at 600 deg. F. and the feed water entering the economizer at 150 deg. The average coal consumption for everyday practice is, say, 4 lb. per boiler horsepower, which corresponds to an evaporation of approximately 8.5 lb. of water from and at 212 deg. Running 24 hours per day, six days per week and 50 weeks per year, the coal consumption will be

$$\frac{4\times 24\times 6\times 50}{2000} = 14.4 \textit{ net tons of coal per b.hp.}$$

Under these conditions it may be assumed that any good economizer will easily raise the temperature of the feed water 100 deg. so that the percentage of heat gained will be,

Water entering economizers at 150 deg. contains 117.9 B.t.u. above 32 deg. F.,

Water leaving economizer at 250 deg. contains 218.5 B.t.u. above 32 deg. F.,

Steam at 150 lb. gage pressure contains 1193.4 B.t.u. above 32 deg. F.

$$\frac{218.5 - 117.9}{1193.4 - 117.9} = 9.35 \textit{ per cent. heat gained}$$

By taking 9.35 per cent. of 14.4 net tons of coal burned per boiler horsepower per year, the gross fuel saving is 1.35 net tons per boiler horsepower per year or 1350 tons for 1000 boiler horsepower.

From this saving must be deducted all charges incident to the operation of the economizer as well as interest on the investment. Under ordinary conditions a good economizer can be installed for about $6.50 per horsepower. This figure includes the apparatus itself, foundations, piping, flues and the increased height of stack necessary for an induced draft system.

These charges may be segregated as follows:

Interest on investment	5 per cent.
Depreciation (assuming life of apparatus to be 20 years)	5 per cent.
Repairs and maintenance	2 per cent.
Attendance	2 per cent.
Power for operating scrapers and, increased pumping head due to economizer	3 per cent.
Miscellaneous	2 per cent.
Total	19 per cent.

On an investment, therefore, of $6500, the estimated cost of a 1000-hp. plant, the total charges would be $1235, which must be deducted from the gross saving of 1350 tons of coal. Assuming, then, that coal can be obtained for $1 per ton, the gross saving would be $1350, from which must be deducted $1235, thus leaving a net saving of $115, which is only 1.77 per cent.

At the following prices of coal the savings on the investment would be:

Cost of coal per ton	Gross savings	Net savings	Per cent. interest on investment
$1.00	$1350.00	$115.00	$1.77
2.00	2700.00	1465.00	22.54
3.00	4050.00	2815.00	43.30
4.00	5400.00	4165.00	64.00

On a basis of a 10-hour day for the same number of days per year as the foregoing, the results would be:

Cost of coal per ton	Gross savings	Net savings	Per cent. interest on investment
$1.00	$562.50	$380.00 loss	5.84 loss
2.00	1125.00	182.50	2.80
3.00	1687.00	745.00	11.46
4.00	2250.00	1307.50	20.11

These figures are based on a total charge against the economizer of 14.5 per cent. obtained by dividing each of the percentages for repairs, attendance, power and miscellaneous by two, but leaving those for interest and depreciation stand, as they would naturally be the same whether the plant operated 10 hours per day or 24.

Such an installation may be considered typical as to what can be expected of a fuel economizer and, while there is probably no actual condition where this example will apply as given, nevertheless, it may be used as a guide and, by the proper substitution of figures for a particular plant, a fairly accurate idea can be obtained as to the results that should follow. As the temperature of the feed water increases or the flue-gas temperature decreases, the advantages of an economizer diminish.

For ordinary practice the general statement may be made that an economizer will be a paying investment where the price of coal is over $2 per ton and where the flue-gas temperatures are over 500 deg. F., or, say, about 100 deg. more than the temperature of the steam within the boiler. It must be borne in mind, however, that often high temperatures of escaping gases may be attributed to the improper proportion of grate to heating surfaces, etc., which might be remedied so as to bring about a considerable saving without installing an economizer.

In considering a fuel economizer only one saving can be readily calculated, the saving in fuel. There are several other advantages, however, that strongly favor its installation under suitable conditions. For example, it will increase the capacity of the boiler plant from 10 to 20 per cent., according to the rate of evaporation per square foot of heating surface in the boiler and, because of the relatively large amount of reserve heat that the economizer is capable of storing, excessive peak loads can be carried more easily.

The life of the boiler is prolonged because the temperature of the feed water may be raised to that of the steam within the boiler, thereby minimizing the excessive strains resulting from unequal expansion and contraction. Also, because of this comparatively high temperature of the feed water, considerable scale-forming matter in the water is precipitated and can be blown off at the economizer; consequently the boiler will be cleaner and have a more efficient heating surface. While it is rather difficult to determine in dollars and cents just how great these advantages are; nevertheless, they should receive proper attention and be added to the saving effected in fuel when making a decision.

After a decision is reached favorable to installing an economizer, it should be remembered that the shortest and least impeded route into the economizer makes for not only greater efficiency but also less draft requirements. It is also important that the economizer tubes be kept clean both inside and out, as the same principle applies as with the boiler.

There were only two engine failures on the Oklahoma division of the Santa Fé R.R. in June, according to a newspaper report. Neither failure was the fault of the boiler department, but happened in the machinery; and both failures were unavoidable. This is probably the best record on the system. It is due to the team work that is in constant practice by the employees of the Santa Fé shops in Arkansas City. In April and June there were no boiler failures charged to the Oklahoma division, and this is a brilliant piece of work that is rarely accomplished on any division of the Santa Fé, if it is accomplished at all. —*The Boiler Maker.*

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

Motor Field Windings

Three general classes of field windings are used on electric motors; shunt, series and compound. The compound field winding is of two kinds, "cumulative" and "differential." These are clumsy words, but the only ones that describe the windings accurately.

The shunt winding is connected across the brushes, as illustrated in Fig. 108. Consequently, it is subjected to the full electromotive force of the supply circuit. Shunt-wound motors are supplied always from constant-potential circuits.

The series field winding is connected directly in series with the armature, as indicated by Fig. 109. Consequently, the full armature current passes through the field winding, except under special conditions when part of the current is "shunted" out of the field winding by a wire or other conductor, as in Fig. 110. Series-wound motors may be used on either constant-potential or constant-current supply circuits, but they are almost always supplied from constant-potential lines.

A compound field winding consists of a shunt winding and a series winding, as represented by Fig. 111. In almost all cases compound windings for motors are of the "cumulative" kind; that is, both of the windings magnetize the cores in the same direction. In a "differential" winding the series winding opposes the shunt winding, but it is much weaker and therefore does not neutralize the shunt winding entirely.

Fig. 112 illustrates the principle of the differential winding. The coil *Sh*, consisting of, say, 30 turns, when excited by its normal current will magnetize the core *C* with the polarity indicated by the letters *N*, *S*. The coil *D* when excited by its normal current will magnetize the core in the opposite direction. If the current in the coil *Sh* be 2½ amperes, that coil will have a magnetizing force of 2½ × 30 = 75 ampere-turns. If the current in the coil *D* be 5 amperes, its magnetizing force will be 5 × 3 = 15 ampere-turns; these will neutralize 15 ampere-turns of the 75 in the coil *Sh* and leave 60 to act on the core, which will be magnetized with the polarity *N*, *S*, but not so strongly as it would be if the coil *D* were absent or unexcited.

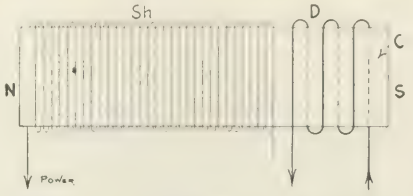


Fig. 112. Differential Winding

The Shunt-wound Motor

Shunt-wound motors, supplied from constant-potential circuits, are used for general service where no special requirements exist. They run at almost constant speed at all loads because the field excitation and the impressed electromotive force are practically constant. The absorbed impressed electromotive force.

A shunt-wound motor operating on a constant-potential circuit has practically constant field flux because the field winding is connected directly to the constant-potential circuit, as illustrated in Fig. 113. The number of armature wires, of course, is constant. The counter electromotive force is equal to the impressed electromotive force minus the drop in the armature circuit:

C.e.m.f. = *Imp. e.m.f.* — *drop*

Consequently, any increase in the arma-

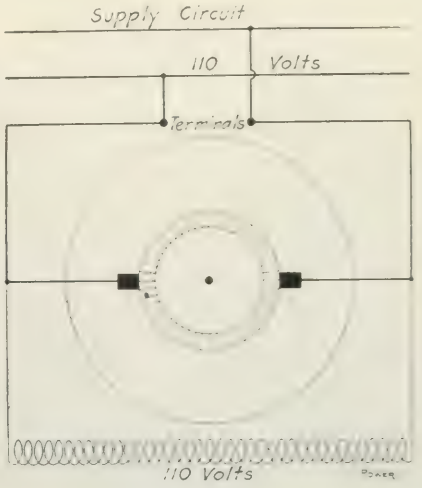


Fig. 113. Elementary Connections of Shunt-wound Motor

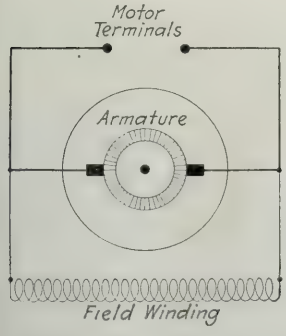


Fig. 108. Shunt-wound Machine

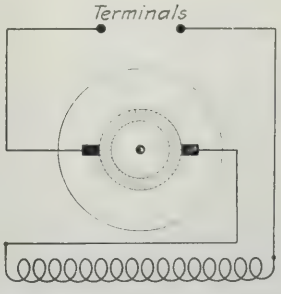


Fig. 109. Series-wound Machine

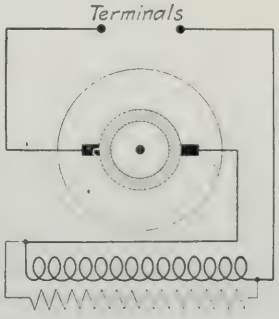


Fig. 110. Shunted Series Field Winding

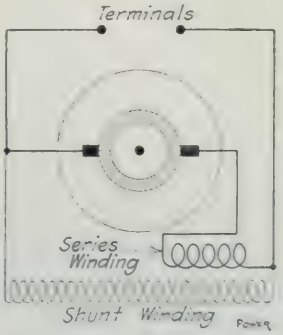


Fig. 111. Compound-wound Machine

factor that causes the armature to change its speed when the load changes is the drop in the armature circuit due to resistance. As explained in the preceding lesson, the speed of a motor armature depends on the field-magnet flux, the number of armature wires and the counter electromotive force that the armature must generate to balance the ture current will cause a decrease in speed, because it increases the drop in the armature circuit and thereby reduces that part of the impressed electromotive force that must be opposed by the counter electromotive force.

An increase in the armature current is caused automatically by an increase in the load on the motor. The reason is

that increasing the load tends to stop the armature, and the rate of speed is reduced until the counter electromotive force falls to a point where it allows the passage of enough current to equal the backward "pull" of the load. Here is where "torque" comes in.

For example, suppose a 110-volt motor had a total field flux of 2,500,000 maxwells (magnetic lines of force), an armature winding of 360 wires connected in two paths and an armature resistance of ¼ ohm. Also assume that the power required to drive the armature alone, without doing any external work, was such as to take 2 amp. from the supply circuit. The drop in the armature circuit at no load would be

$$\tfrac{1}{4} \times 2 = \tfrac{1}{2} \text{ volt}$$

leaving 109½ volts to be balanced by counter electromotive force. The speed at no load, therefore, would be

$$\frac{6{,}000{,}000{,}000 \times 2 \times 109\tfrac{1}{2}}{2{,}500{,}000 \times 360} = 1460 \text{ r.p.m.}$$

Now suppose a load were thrown on the motor which increased the backward drag to a torque of 9½ lb.-ft. With the current of 2 amp., which passed through at no load, the torque of the armature would be

$$\frac{2{,}500{,}000 \times 360 \times 1}{852{,}230{,}000} = 1.056 \text{ lb.-ft.}$$

which could not keep the speed up against the load torque of 9½ lb.-ft. The armature speed, therefore, would be pulled down by the excessive drag of the load; but as soon as the speed began to fall, the resultant fall of the counter electromotive force would allow more current to flow, thereby increasing the armature torque. When the speed diminished to 1437½ r.p.m. there would be no further decrease because the armature current would have increased to 9 amp., producing enough torque to balance the load drag. Thus, the counter electromotive force at 1437½ r.p.m. would be

$$\frac{2{,}500{,}000 \times 360 \times 1437\tfrac{1}{2}}{6{,}000{,}000{,}000 \times 2} = 107\tfrac{3}{4} \text{ volts}$$

leaving 2¼ volts to be taken up by resistance. The resistance being ¼ ohm, the current would be

$$\frac{2\tfrac{1}{4}}{\tfrac{1}{4}} = 9 \text{ amp.}$$

The torque produced by 9 amp., under the conditions described, would be

$$\frac{2{,}500{,}000 \times 360 \times 9}{852{,}230{,}000} = 9\tfrac{1}{2} \text{ lb.-ft.}$$

From this example it is evident that a shunt-wound motor supplied from a constant-potential line alters its speed automatically when the load changes, the speed variation being opposite to the load variation and just enough in extent to cause the armature current to increase or decrease to the exact value demanded by the load. It is also evident that the speed of such a motor does not vary in proportion to the load variation, the speed change being very much smaller than the load change. In the case just cited, for example, the speed dropped only 22½ revolutions in 1460, or 1½ per cent. in order to take care of a load increase of more than 200 per cent.

Starting a Shunt-wound Motor

When the armature of a motor is at standstill, the only opposition to the flow of current from the line is the resistance of the armature circuit. In the case just considered, if the motor were connected

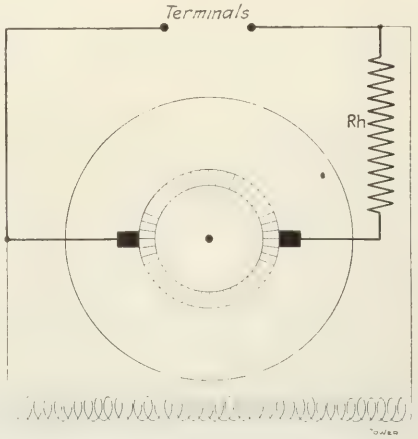


Fig. 114. Elementary Connections Including Starting Resistance

directly to the line while motionless, there would be an enormous rush of current through the armature. The resistance being only ¼ ohm, the tendency would be for a current of

$$\frac{110}{\tfrac{1}{4}} = 440 \text{ amp.}$$

to rush through. It takes time, however, for the current to reach the maximum value determined by the electrical resistance of a circuit. In the case of the motor, therefore, the armature would start with a tremendous jerk and begin generating counter electromotive force before the current reached 440 amp., but the rush would undoubtedly rise to between 300 and 400 amp. before the armature got under way.

To prevent such excessive current rushes, a shunt-wound motor is provided with an auxiliary apparatus commonly termed a "starting box"; the correct name is "starting rheostat." A starting rheostat consists of a few coils of wire —usually of iron—mounted on noncombustible insulators and arranged so that the wire can be cut out of circuit, section by section. This is connected in series with the armature of the motor as illustrated by Fig. 114, where *Rh* is the wire of the starting rheostat. When the connections between the armature and the supply circuit are first closed, all of the rheostat wire is in circuit, and its resistance is high enough to prevent a heavy rush of current through the armature, even if the latter should be prevented from revolving. For example, if the full-load armature current of a 250-volt motor is 25 amp., the resistance of the starting rheostat should be not less than 5 ohms, and would preferably be about 8½ ohms. With a 5-ohm rheostat the current rush would be slightly less than 50 amp., because the resistance of the armature winding and connections is added to that of the rheostat, making the total resistance slightly more than 5 ohms.

The usual method of cutting out the starting rheostat is illustrated in Fig. 115. A pivoted arm *A* is arranged to swing over a row of contact buttons located on the arc of a circle and connected to the resistor* *Rh* as shown. A contact finger on the end of the arm rubs on the faces of the contact buttons. When the circuit between the motor and the supply line is first closed, current flows through the field winding but not through the armature, because the arm is not in contact with any of the buttons. The arm is then moved to the first button, thereby closing the armature circuit through the whole resistor. When the armature has got a good start, the arm is moved to the second button, thereby leaving the first section of the resistor out of the circuit and increasing the current in the armature. After the resulting increase in speed,

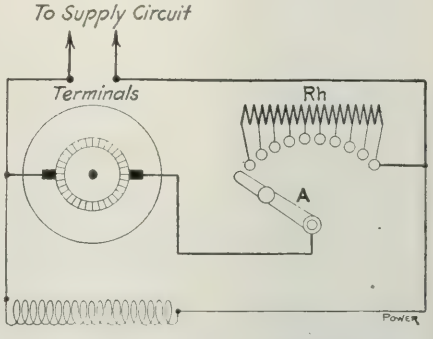


Fig. 115. Shunt-wound Motor and Starting Rheostat

the arm is moved to the third button, and so on, until the last button is reached and all of the resistor is cut out, leaving the armature connected directly to the supply circuit.

Horsepower

Just as a steam engine has a "horsepower constant," by which the mean effective pressure may be multiplied to find the indicated horsepower, so a shunt-wound electric motor has a horsepower constant by which the armature current may be multiplied to ascertain the horsepower.

The horsepower constant may be derived in two ways: either from the impressed electromotive force and armature resistance or from the magnetic flux.

*A resistor is a conductor used solely for adding to the resistance of a circuit.

armature wires and speed. The former is easier. The mechanical power (in watts) developed by the armature is equal to the watts taken from the line minus the watts lost in driving the armature and overcoming the resistance of the armature circuit. The watts lost by armature-circuit resistance cannot be determined without knowing what the armature current is, but as the total armature losses are commonly between 4 and 10 per cent. of the input, an assumption of 6 or 7 per cent. will not be far enough out to cause important error. On this assumption, the horsepower constant is given by the simple formula:

$$\frac{Impressed\ e.m.f.}{800} = hp.\ constant$$

Multiplying this constant by the full-load armature current will give a fair approximation to the full-load brake horsepower of a shunt-wound motor, within the ordinary workshop range of sizes—say from 1 to 50 hp.

The formula is based on the fact that if there were no losses whatever, the horsepower of the armature would be equal to the armature input in watts divided by 746; and on the assumption that the average armature losses are 6¼ per cent. of the armature input. If 93¼ per cent. of the impressed armature watts divided by 746 equals the brake horsepower, then the impressed armature watts divided by 800 will equal the brake horsepower and the impressed electromotive force divided by 800 will equal the brake horsepower *per ampere of armature current.*

Care of Direct Current Switchboards

By G. B. Longstreet

As all the power delivered by the generators has to pass through the switchboard, it necessarily follows that strict attention should be given to the appliances mounted thereon to insure their working properly.

The switchboard should be located in a dry place and well insulated. If there are windows near it, they require careful watching in stormy weather. The board should be kept clean and neat both in front and back, and enough room provided in the rear to allow a person to work freely without coming in contact with any wire or busbar.

If a compressed air blast is to be had this should be used occasionally for blowing the dust from the board, a light blast being used without a metal nozzle attached to the hose. Where this is not obtainable the work can be done with hand blowers.

It is best to put the name of each circuit on a card and place this directly over the switch controlling that circuit. Receptacles made expressly for receiving these cards can be purchased at a small cost, also one or more drawings of the switchboard and connections are a help to the attendant at all times, and on large or complicated boards these are a necessity.

The switch blades should be kept bright to make good contact, and should work freely. A very small amount of vaseline applied occasionally to the blades will allow them to close easily and will not interfere with good contact.

Automatic circuit-breakers require considerable attention. They must work freely and accurately and both the carbon and the copper contacts should be examined frequently, especially after bad blowouts, to make sure that good contact is made and that there are no badly burned or blistered parts. If there are, the parts should be replaced.

Rheostats must also receive a like amount of attention. A good contact must be maintained between the movable finger and buttons or bars, as the case may be. A small amount of vaseline applied to them gives free movement without impairing contact. In the case of bar contacts special care should be taken that the intervening spaces do not fill with dirt.

Voltage regulators should receive attention according to the directions furnished by the makers. All meters should be calibrated at regular intervals and if found incorrect should be corrected. Busbars should be marked "positive" and "negative."

In plants where the service cannot be suspended to make repairs to circuit-breakers or switches, a jumper may be used. This is made of cable with a forked terminal soldered to each end. Before pulling the switch or circuit-breaker one end of the jumper should be fastened to the switch-post of the out-going cable and the other end connected to the switch-post of the out-going cable of the circuit chosen to take care of the load during repairs. Where possible it is preferable to do this when the load is light, but if this is impossible care should be taken to choose a strongly fused circuit upon which to tap. A good policy is to have an extra switch or circuit-breaker to be used for this purpose only.

CORRESPONDENCE

Motor Windings

Where a number of induction motors of different types and makes are in operation, it is convenient to have at hand diagrams indicating the method of connecting the phases, when new coils are to be installed.

After dismantling the machine so that the end connections can be seen, make a pencil drawing to a reasonably large scale in the following way: Draw a circle, and by diametrical lines divide into as many segments as there are coils in the stator. Close to each line and on opposite sides indicate by pencil the inner and outer terminals of the coils. Follow out and indicate on the drawing in different colors one phase at a time. After tracing for a distance of two or three poles, count back to the beginning and check for errors. A little practice will be required to trace these readily. When complete make a diagram on tracing cloth for filing in a loose-leaf notebook, preferably using different colors for the different phases as on the rough drawing, from which a print can readily be made for use on the job.

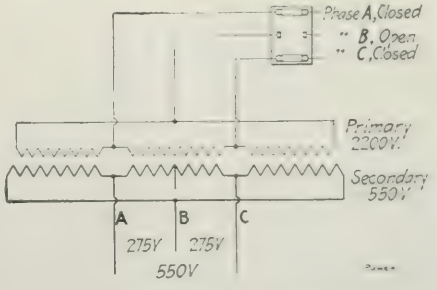
G. B. Cook.

Miami, Fla.

Blown Fuse

In the Nov. 5 issue, Mr. Koppel, in referring to the sketch therein given, wishes to know which of the primary fuses were blown, thereby causing the motor trouble. He states that upon making a voltage test at the motor the following readings were taken; *A* to *B*, 550 volts; *B* to *C*, 275 volts and *A* to *C*, 0 volts.

Phase *A* on the primary was the one that was open, but I wish to call attention to the test which Mr. Koppel made. Owing to the incompleteness of his sketch in which he failed to show the manner of test and the outside wiring on the secondary through starting box, it is hard to tell just why there was no reading between phases *A* and *C*.



Primary Fuse *B* Blown

The accompanying sketch shows the inside connections at the same transformers, delta connected. For simplicity, assume that phase *B* on the primary is open. This does not leave either transformer dead, although the two outside ones will be left in a weakened condition. Each outside transformer will be operating on one-half the line voltage, as one circuit gives two transformers in series. Then the voltages on the secondary side should read in relation to their respective position to the primary; that is, from *A* to *C* there will be full voltage and between *B* and *C* or *A* and *B* there will be one-half the normal voltage.

C. W. Cox.

Alameda, Calif.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gasoline Engine Economy*

By A. E. Potter

Carbureters differ greatly in their consumption of fuel, and some can be adjusted to operate on materially less gasoline than others. With a motor throttled to its lowest speed the mixture has to be materially richer in gasoline vapor than when run at maximum speed, for two reasons; first, to allow for dilution of the charge by air drawn into the inlet-valve chest past the valve stem, and, second, for the reason that upon opening the throttle, before the engine can pick up speed, the velocity of the air past the needle valve is temporarily slower, drawing in less gasoline and resulting for a short time in a slightly weaker mixture. If this mixture is made very much poorer the motor will not pick up power, often noticeable in a motor car on a hill, and unless the speed lever is dropped back in such cases the motor will be "stalled." If the air velocity is too great, it is likely to result in an accumulation of unvaporized gasoline in the inlet manifold, and upon opening the throttle this raw gasoline is drawn into the cylinders, choking the motor with a gas so rich that it will not ignite, or upon ignition will produce very little power, and as in the former case may "stall" the motor. In one case the mixture is too poor and in the other too rich, both to be avoided if possible, the latter being more important than the former, as it results in waste of gasoline. This latter condition is quite likely to be present in motors equipped with carbureters too large for the motor, always resulting in waste of fuel, no matter whether the mixture be too rich or too poor.

The first internal-combustion engines using gasoline fuel were of the hit-and-miss-governed type.

While no great skill is needed to design a vaporizer or constant-level carbureter that will operate an engine of this type, and that will allow the development of rated horsepower, to design such a device as will result in minimum consumption of fuel per horsepower-hour delivered is not so easy.

The usual design is a straight pipe with a spraying nozzle entering the lower side, a pump to supply gasoline to a cup or bowl with an overflow back to the tank. This pipe may be the same size as the engine intake or may be smaller, but never have I found a device of this kind employing the true venturi principle—a principle that will give the air its maximum velocity to completely vaporize the gasoline without reducing the volume of each charge to such a point as to result in a reduction of power.

This type of engine, if properly designed, has a vaporizing device that will deliver maximum horsepower at a minimum consumption of gasoline, and when called upon for explosive charges intermittently the air has at all times practically the same velocity, and as a result the efficiency should be practically the same at high, low and intermediate speeds.

But with throttle-governed engines, conditions are such that marked fuel economy under varying power requirements is the exception, rather than the rule.

It is only reasonable to believe that the higher the air velocity the more completely will the gasoline be vaporized and the more intimately mixed with the air, resulting in high efficiency. On the other hand, to induce this high velocity the volume of the incoming charge may be reduced to such an extent that any loss as a result of imperfect vaporization may be more than offset by a reduced output. So long as there is a safe margin of power above the greatest requirement, such a condition would result in economical operation, but if extra demands are made of such an engine, it would not be able to deliver any more power than an engine of relatively less piston displacement.

To overcome these and other disadvantages, and to render such engines more efficient at varying speeds, they are usually equipped with automatic carbureting devices. These may be broadly divided into three classes: those with proportional openings for air and gas; those that dilute a rich gas with auxiliary air; and those that adjust the amount of gasoline to an auxiliary supply of air.

When properly proportioned to the demands of the engine, each class should show high efficiency. With engines that are not subjected to sudden and intermittent power demands, carbureters with proportional openings give fair results, but they seem to operate best with a mixture slightly rich in gasoline and are not generally recognized as being especially economical.

In the other two classes if the velocity of the air is too high there is likely to result a loss of volume and a resulting drop in the horsepower delivered, while if too slow there is a strong possibility that the gasoline will not be thoroughly broken up and mixed with the air, with loss of efficiency, and the result is poor economy.

A series of tests of six different carbureters, conducted at Purdue University, and reported by Prof. George W. Munro in a paper read last June before the American Society of Mechanical Engineers, shows marked differences in fuel consumption and power developed.

While these tests were recorded at 300, 400, 500, 600, 700, 800, 900, 1000, 1100 and 1200 r.p.m., for present purposes the results at 600, 800, 1000 and 1200 only will be compared. At 600 r.p.m., the maximum horsepower developed was 19.53, the carbureter showing an efficiency of 17.4 per cent., this being the highest efficiency shown by any carbureter at that speed, while the minimum horsepower recorded was 14.89, with but 14.3 per cent. efficiency. The efficiency was based on an assumed fuel heat value of 20,700 B.t.u. per pound, which value is assumed in all the following comparisons.

At 8000 r.p.m. the carburetor showing also the greatest power at 600, delivered 25.01 hp., the same power with an efficiency of 15 per cent. that another showed with an efficiency of but 14.4 per cent.

At 1000 r.p.m. the efficiency of this last carbureter increased to 14.5 per cent., delivering the same horsepower as two others showing respectively 19.5 and 20.5 per cent. efficiency, while the carbureter showing the greatest horsepower at 600 and 800 showed 30.441 hp. with an efficiency of 17.5 per cent. Another showed the same horsepower at an efficiency of 19.3 per cent.

It was at 1200 r.p.m., about the speed at which the engine should have shown best results, that the greatest disparities were apparent, three different carbureters showing the same horsepower delivered, while their efficiencies proved to be respectively 15.5, 22.2 and 15.3 per cent., the consumption of gasoline being proportional to these different percentages, the best economy being 0.74 pint per horsepower-hour, and the most uneconomical nearly 1.07 pint per horsepower per hour.

The plausible theory to account for these varied results, is that the gasoline was less perfectly vaporized and mixed in one instance than in either of the other two.

*From a paper read before the National Gas Engine Association at Indianapolis, Dec. 5, 1912.

There is one other patent cause of waste of gasoline which deserves brief mention; this is the timing. To ignite a charge any later than will give maximum power means waste of fuel. The earlier the ignition can be set without knocking, the nearer closed can be the throttle; also the more synchronous the timing the earlier it can be set. A single spark occurring with absolute regularity, not early and late intermittently, as may occur in a single-cylinder engine with parts badly worn or with loose timing apparatus, will develop more power than a series of sparks which may or may not be timed to begin at absolutely the same instant every time when called upon to furnish ignition. In like manner a timer that does not have all its contacts spaced absolutely correct may cause the ignition to occur early in one cylinder, while one or all the others may get the spark materially later.

Gasoline-engine economy, therefore, seems to resolve into four factors:

1. The proper proportioning of the carbureter to the engine; that is, neither too large nor too small.
2. Proper carbureter adjustment.
3. The elimination of all carbureting devices that waste fuel.
4. Synchronous and as early timing as is consistent with intelligent operation.

The Junkers Engine—VI

By F. E. Junge

Balancing

First consider the inertia forces and tilting moments of a combination of three Junkers tandem engines with cranks set at 120 deg. The residual inertia forces, which were considered under the heading "Pressure on Bearings of Junkers Engine," (see Nov. 26 issue) for each engine plane taken as a whole, completely neutralize each other. This will be made clear by the following illustrations, of which Fig. 31 shows the constructive arrangement of the unit and Fig. 32 the pressure exercised by inertia. In spite of this neutralization a tilting moment exists, the nature of which is shown in Fig. 33. For the ratio of crank circle radius to length of connecting rod of 1 to 5, the maximum unbalanced inertia force for each tandem engine is two-fifths of the inertia force for a single engine at the dead center with an infinite connecting rod. These forces must be transmitted through the engine's bedplate, without perceptibly straining the latter, if an effective balance of the inertia forces is to be attained. Considering the small value of these forces in the Junkers engine, and their rapid rate of change, these conditions may easily be satisfied by a light bedplate. The maximum value of the tilting moment, is approximately 1.73 times the residual inertia force at dead center with an infinite connecting rod, multiplied by the distance between two successive center lines of the engine.

This couple tends to rock the engine on its foundation, as was shown for the reaction of the twisting moment (see page 715, Nov. 12 issue). However, since this moment is small in magnitude and is subjected to rapid changes, it may easily be met.

The inertia forces and tilting moments of a three-cylinder Diesel engine with cranks set at 120 deg. will next be considered. In this arrangement which is represented in Figs. 34 to 36, the inertia forces entirely neutralize each other, as in the Junkers engine. But, since the inertia forces occurring at dead center in the Diesel engine are three times as large as those existing in a Junkers engine under the same conditions, the unbalanced tilting moments in the former are decidedly larger than in the latter. Moreover, the changes take place only one-half as fast as in the Junkers engine. Therefore, to procure satisfactory inertia balance in the Diesel engine, it is necessary to design the engine frame for forces which are three times as large as those of the Junkers engine. Hence a heavy, expensive frame and substructure are necessary.

Now consider the six-line arrangement in Junkers engines. The tilting moment of the three-line arrangement of cylinders can only be completely neutralized by applying to the same bedplate an opposing moment, which, for all positions of any crank, will be in equilibrium with the couple in question. This is attained by the arrangement of a second similar engine on the same bedplate. In the Junkers three-line tandem arrangement the tilting moment runs through the same values every half revolution (see Figs. 31 to 33). As far as abolishing any free tilting moment is concerned, it is thus immaterial as to whether the compensating three-line tandem set be fixed at 0 deg. with respect to the crank of reference of the first engine, or at 180 deg. Since the setting of the cranks at 180 deg. produces a uniform twisting moment, the division of the crank-pin circle into six equal parts should be followed. The couple exercised by the one engine on the other, through the medium of the frame, is, as already mentioned, 1.73 times the unbalanced inertia force at dead center for an infinite connecting rod, times the distance between the center lines of two adjacent cylinders. Considering the rapid alternations of this couple and its small magnitude, a complete inertia balance may be obtained, in the Junkers engine, by providing a light connecting bedplate. To this is added the advantage of completely utilizing the six cranks to procure a uniformly combined crank effort or twisting moment.

The six-line tandem arrangement of the Junkers engine thus provides a perfect inertia balance and a highly uniform turning moment, notwithstanding that the engine and substructures are light and inexpensive.

Considering the six-cylinder arrange-

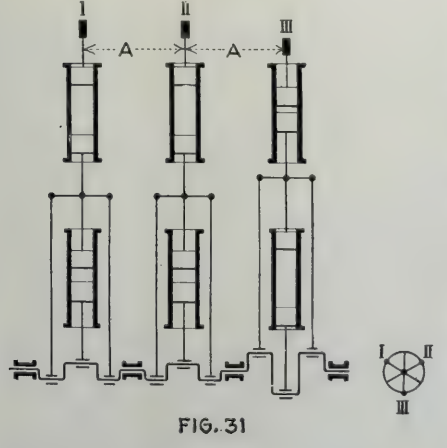
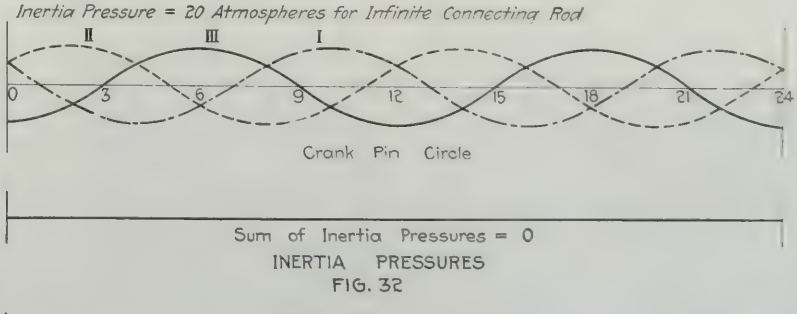


FIG. 31



INERTIA PRESSURES
FIG. 32

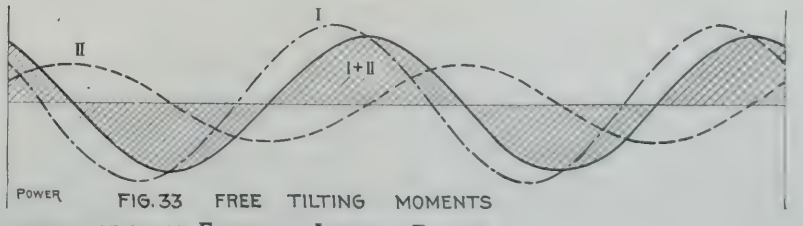


FIG. 33 FREE TILTING MOMENTS

Graphical Representation of Forces in Junkers Engine

ment in Diesel engines, should a balancing of the tilting moments be desired, this can be accomplished by placing two three-line units in juxtaposition, at 0 deg. relative displacement of the cranks. This restriction is essential because the unbalanced couple goes through one complete cycle per revolution. Moreover, the bedplate is subjected to a tilting moment 150 per cent. larger than that occurring in the Junkers engine, and the alternations are one-half as slow. Therefore, a satisfactory inertia balance in the Diesel engine can be attained only by a heavy, expensive bedplate. In consequence of the 0-deg. spacing, the twisting moment is only one-half as uniform as in the Junkers engine.

Therefore, to procure complete balance in the Diesel engine the frame must be so proportioned as to transmit forces and moments 200 per cent. and 150 per cent. larger, respectively, than those encountered in the Junkers engine, with alternations one-half as rapid. Yet, the six-

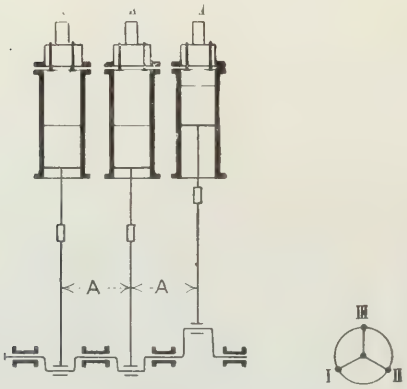


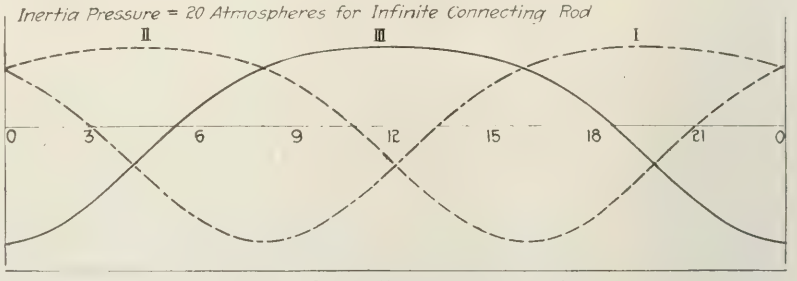
FIG. 34

cylinder arrangement does not provide a more favorable turning moment than an ordinary three-cylinder engine. Notwithstanding the additional expenditure, the twisting moment is only half as uniform as that peculiar to the Junkers aggregate. The result is a heavy expensive frame and substructure. If in the six-line Diesel arrangement a turning moment equivalent to that in the Junkers engine is to be attained, it can be done only by sacrificing the tilting moment compensation provided in the frame. A free unbalanced couple 300 per cent. larger than that in the Junkers three-tandem arrangement thus results.

Frame and Supports

From the foregoing considerations the following deductions apply to the bedplate and foundation: Owing to the excellent inertia balance in the Junkers engine its demands as to substructure are but small; hence the foundations may be light and inexpensive. Moreover, harmful vibrations in the building, vehicle or vessel are thus obviated. In all cases where perfect balancing and quiet running is required the Junkers engine is superior to all other reciprocating combustion engines. Consequent to the interlocking of the acting forces in the power transmitting mechanism itself, the cylinders and their connections with the main bearings are not subjected to any reacting stresses. In the normal horizontal Diesel engine the excessive reacting forces have to be taken up by the cylinder and the bedplate. The connection between the cylinder and the bedplate is thus strongly subjected to bending and, therefore, must be made as inflexible and as heavy as possible. Cast iron, however, is never to be relied upon in transmitting large forces. Hence this arrangement is always the weak point in the design of horizontal Diesel engines.

As to the main bearings in the normal Diesel engine they have to support extremely large loads, owing to the large piston forces exerted. Thus, to keep within the bounds of an acceptable specific pressure the bearings must be heavy and long, calling for careful superintendence. The contrary holds true in the Junkers engine. Here, owing to the compensating arrangements, the main bearings are well relieved from their loads, so that with relatively short bearing surfaces only low specific pressures are to be anticipated.



INERTIA PRESSURES
FIG. 35

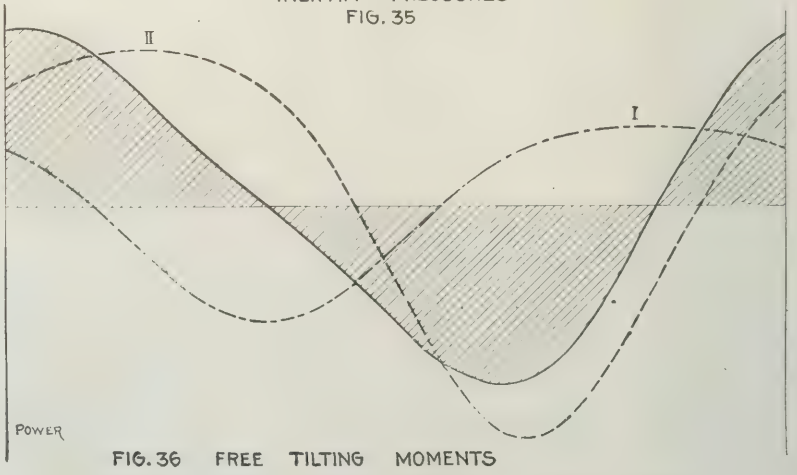


FIG. 36 FREE TILTING MOMENTS

Graphical Representation of Forces in Diesel Engines

CORRESPONDENCE

Operating Gas Engines

Mr. Brennan's article on operating gas engines in the Nov. 26 issue states, in the list of troubles common to the make-and-break system of ignition: "To ascertain at once if the existing trouble is in the wiring or the igniter, place the switch in position to close the circuit and then with a screwdriver form a circuit from the stationary electrode on the igniter to the cylinder. If a good spark results by moving the screwdriver, the circuit is complete and the trouble is in the igniter. Having traced the trouble to the igniter, it may be due to dirty points or to the actuating springs working improperly. On the other hand, if no spark is had it will be necessary to look over the battery connections and terminal points of all the wiring."

While the foregoing is true in a way, yet if the electrodes are not in contact there will be a spark at the screwdriver. On the other hand, if the electrodes are making contact there will be no spark unless the contact points are dirty or badly burnt.

The correct way is to try the screwdriver method by first holding the igniter in contact, and then holding the electrodes apart. If there is no spark in the first place at the screwdriver and there is a good spark when the electrodes are held separated, then the igniter and wiring are all right.

Hawley L. Scott.

Bangs, Ohio.

The second annual electrical show of the Peoria Show Association will be held in the Coliseum. Peoria, Ill., Jan. 18 to 25.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Repairs and Improvements in an Ice Plant

By W. S. Luckenbach

As the ice-making season is over, every owner and conscientious engineer should seriously consider ways and means of increasing the capacity of his plant without materially increasing the operating expenses or fixed charges. All changes or additions require careful consideration, but it will be a comparatively easy matter if the engineer has kept daily records covering the entire plant, viz., consumption of fuel, temperature of feed and cooling water, temperature of distilled or raw water for ice making, brine and liquid anhydrous temperatures, and in an absorption system the temperature of the poor liquor at the absorber. If he has also noted the cost of labor, oil, packing, etc., and the amount of ice produced, he is well equipped to sum up the situation. It will be a great help if he can compare his plant with others working under similar conditions.

The engineer in charge should make a note of all defects, large or small, from the boiler room to the dump tub and leave nothing to memory, as the failure to replace even a nipple or a fitting may cause trouble and expense during the busy season.

Have the steam boiler thoroughly gone over from grate bars to safety valve. In the writer's opinion one of the principal adjuncts is a first-class heater and purifier. This apparatus will remove a large percentage of the impurities and indirectly will increase the efficiency of the boiler by reason of cleaner tubes and less blowing down to remove the sludge. Every extra gallon of hot water blown into the sewer adds to the cost of fuel. It is good practice to insert a thermometer in the feed line so that the temperature of the water from the heater may always be known. The rear-end soot blower is also a fuel-saving device. In the old method of blowing from the front, the flue doors must be kept open; and when the tubes are over 12 or 14 ft. long it is impossible to thoroughly clean the back ends on account of blowing against the draft and into an inclosed chamber. When blowing from the rear the work can be done at any time and the tubes cleaned thoroughly, because the soot is blown with the draft and up the stack instead of into the combustion chamber, from which some may return a second time into the tubes.

Boiler-feed and cooling-water pumps need careful attention, especially if steam driven. The examination of pistons and slide valves must be made under steam pressure. Disconnect the valve stems from the rocker-arms and with the back head off test the valves in various positions, as they may be tight when covering the ports and leak at one end when taking steam at the other. The water ends also require attention. If the cylinders do not fill at each stroke the pump must be run at higher speed; otherwise more steam will be required to pump the same quantity of water. For the same reason the ammonia pump should be put in first-class condition.

Every season the generator coils or pipes should be cleaned because the impurities of the aqua ammonia frequently adhere to their surface and form a nonconductor of heat. Especially is this necessary when exhaust steam is used for generating the gas from the rich liquor. With dirty coils the volume of exhaust must be increased or a higher pressure (meaning also a higher temperature) maintained. All coils or pipes should be tested to 500 lb. and on the high-pressure side especially it is common to use extra-heavy pipe.

Atmospheric condensers are generally located on the roof, and if so, should be protected from the sun's rays. The protection in the shape of a roof has a twofold effect in that it shades the coils and creates some draft at all times; hence less cooling water, less pumpage and less steam required. Double-tube condensers are usually installed in the building, but should be so located as to have a current of air passing over and through them, for the same reasons as above. It is important to have the liquified gas pipe from the condenser to the expansion valve run through a cool part of the building in order to prevent an increase of temperature.

If the freezing tank and coils have not been cleaned for several seasons, they should receive attention. To prevent corrosion this work should be done quickly so the expansion coils will not be exposed to the atmosphere any longer than actually necessary. Each coil should be blown out separately and tested to not less than 300 lb. See that the pipes in every round of coil are straight, thus avoiding the possibility of a trap which would seriously decrease its efficiency.

If it has been difficult to maintain a comparatively even temperature it may be necessary to resort to brine circulation. Either of the two methods outlined below may be used. Set a brine pump close to the freezing tank where it will be protected from excessive heat. The suction pipe should run along the bottom of the tank three-quarters of the way across the bath, with the perforations in the pipe turned upward. The discharge pipe at the other end of the tank should also be perforated and be several inches below the surface, with the holes turned down, and the pipe leading from the suction to the discharge should be inside the tank if possible or at least be protected from the surrounding atmosphere. The pump should have sufficient capacity to change the liquid about once every six hours. There is, however, no hard and fast rule to go by. The brine is simply changed enough to insure a uniform temperature of the proper degree throughout the bath. Any additional circulation means a loss in steam.

The other method is to partition the tank longitudinally into two distinct divisions, leaving ample space opposite the agitator for the brine to flow from one division to the other. The agitator consists of, say, a 16-in. pipe about 4 ft. long fitted with a four-bladed propeller, the whole set at an angle and so arranged that the brine will be drawn from the bottom of one division and discharged upward into the other division. The methods of driving the propeller are many, but in this case two-to-one bevel gears were used and driven with a ¾-in. rope from a sheave attached to the ammonia-pump flywheel. The propeller shaft runs in a babbitt metal step box, secured to a cross bar fastened to the 16-in. pipe, for an entire season without attention, the exposed bearings, of course, needing lubrication. For a 30-ton machine the speed would be about 40 r.p.m. and the power required practically nil.

Use nothing but new pipe and fittings on the ammonia system because the threads will then be sharp. Do not use any pasty lubricant to make up joints such as red lead, glycerin, litharge, etc., because particles will lodge between the threads, making it impossible to screw the fittings iron to iron. These particles will harden and be tight but it will be only a question of time before the ammonia will penetrate and cause a leak, and most likely this leak will occur when you can ill afford to remove it. The writer's experience has been to first secure pipe

and nipples which have sharp threads and no internal, ragged edges, use a little graphite and good oil and screw up good and hard iron to iron. A lasting job and a joint that can be readily unscrewed will be the result.

Freezing Can Ice in Twelve Hours*

By W. Everett Parsons

Ordinarily the time for freezing a 300-lb. block of ice in a standard can varies from 42 to 60 hr., depending on the temperature of the water filled into the can, temperature of the brine, rapidity of the brine circulation, etc. These conditions remaining constant, the freezing time is about proportionate to the square of the thickness frozen. Advantage of this is taken in the quick-freezing process owned by the United Ice Improvement Co., of New York City. The accompanying diagrams show that an ordinary standard 300-lb. can may be used. Three-inch pipes, sealed at their top ends, project upward from the bottom of the can. The lower ends of these pipes are open and are screwed into a plate fastened to the under side of the can so that a tight joint can be made and the brine used for cooling cannot leak into the can. Under the can is a brine-circulating header with three small pipes for each can projecting upward, and so arranged and spaced that when the can is in place for freezing the small pipes extend upward inside of the 1-in. pipes. The cold brine is circulated through the header and passes up through the small pipes, then down again inside the 1-in. pipes to the bottom of the can and up the sides of the can to the top. Instead of the arrangement shown in Fig. 1, the small pipes can be attached to the can when it has to be lifted from the tank in the usual way. Fig. 2 shows a cross-section of a 300-lb. block of ice frozen in 12 hr. by the new process, with brine entering the pipes in the can at about 12 deg. F.

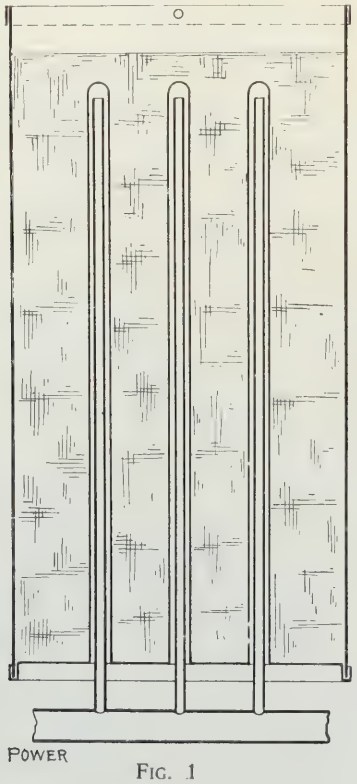


Fig. 1

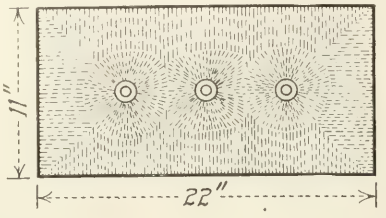


Fig. 2

The proportion of this cross-section of ice frozen from the sides of the can is about 63.6 per cent., and from the three pipes about 36.4 per cent.; whereas the periphery of the sides is 66 in. and of the three pipes 12.4 in. In other words, the ratio of the ice frozen from the pipe surface to that from the outside surface of the can is 1 to 1¾, while the ratio of the surface of the three pipes to that of the sides of the can is 1 to 5¼, showing that 5¼ times as much surface in the outside of the can freezes only 1¾ times as much ice. The rapid flow of the brine in the pipes and the slightly lower temperature of the brine account for the better showing made by the pipes. When the block of ice is released from the can there are three holes about $1\frac{5}{16}$ in. diameter, which extend from the bottom to within 3 in. of the top. When the block is stood upright in the ice storehouse, the holes are sealed by the floor and there is no melting in the holes. If the ice storage is kept below the freezing point, and it is desired to eliminate the holes, they may be filled with cold water, which will freeze solid without any further attention.

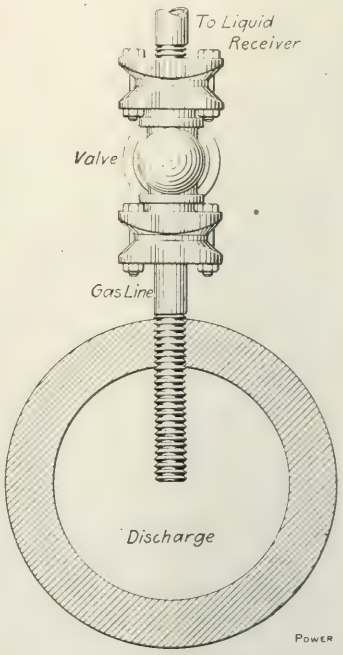
*Abstract of paper read before the Eastern Ice Association, New York City.

CORRESPONDENCE

Gas Relief from Liquid Receiver

From observation and experience I have noticed that little attention has been, or is being, paid by refrigerating engineers to gas relief from the liquid receiver. Most gas-relief pipes are tapped into the bottom or sides of the gas-discharge line at or near the header on the condenser. This arrangement does not properly relieve the gas generated in the receiver, and as a consequence it passes out with the liquid to the expansion valves, causing them to sputter and retard the steady flow of liquid ammonia.

To remedy this trouble I trapped the gas-relief line into the top and through to the center of the discharge line at the condenser gas header in the manner shown in the sketch. I made a ½-in. nipple with a running thread long enough to go down to about the center of the discharge line so that the flow of gas in this line rushing past the opening in the nipple on its way to the condenser would draw the gas out of the relief line from the receiver instead of allowing it to back down to the receiver as it generally does where the line is tapped into the side or bottom of the discharge line and merely enters the wall of the pipe.



Gas Drawn Into Discharge Pipe

From the above changes, I have obtained excellent results and was well repaid for the extra work of changing the piping.

C. E. Anderson.

Chicago, Ill.

A French metallurgist has manufactured an alloy, with the hardness of steel and with great tensile strength, by the addition to copper of chromium, aluminum, nickel and zinc at certain temperatures, which are maintained for specified lengths of time.

An interesting pamphlet has been published by the Illuminating Engineering Society under the title of "Light: Its Use and Misuse." The illustrations are well chosen and present concrete ideas on this important subject. Copies can be obtained from the secretary of the society, at 29 West Thirty-ninth St., New York City.

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FRED R. LOW, Editor.

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POWER

Increased Plant Efficiency

That further gains in the economy of the power plant will be brought about in the boiler room rather than in the engine equipment or auxiliaries has been repeatedly stated in recent years. This is undoubtedly true, although refinement of operation and improvements in steam-using apparatus will continue to be made from time to time.

Since the first steam engine, the aim of the inventor and manufacturer has been to increase its economy and to reduce the cost of operation and upkeep. This apparently has been carried about as far as it is possible to go, insofar as generating units and their auxiliaries are concerned.

It is interesting to glance back over the changes that have been made in power-plant apparatus. The old slide-valve engine was replaced by the Corliss, both simple and compound. This was the first great step in cutting down the steam consumption. Other designs of four-valve engines have followed with equally satisfactory results.

Condensers have added greatly to the economy of steam engines and are the dominating factor in the successful and economical turbine operation. When used with reciprocating units the steam consumption is reduced from ten to fifteen pounds of steam per horsepower-hour.

Coupling the reciprocating engine exhaust to a low-pressure turbine has further reduced the steam consumption per kilowatt-hour. This represents the most economical operation of power-plant generating units.

But what of the boiler room, and where is the economy to be obtained in this section of the power plant? Boilers today are as well made as high-steam pressures demand, but this does not add materially to the economy of the boiler plant.

Evidently the furnace design and the method of burning the fuel are the two avenues through which engineers may expect to increase plant economy. The furnace is a problem in itself. It can be built without any forethought as to whether it and the grates used are suitable for the kind of coal to be consumed or not. Or it can be designed with the aim of obtaining the highest rate of combustion of the fuel before the gases strike the cooling surface of the boiler. The dutch-oven, if properly designed, has this advantage. It has, however, the disadvantage of occupying additional floor space over the ordinary furnace.

Combustion of the fuel is, to a great extent, under the control of the fireman. If he knows what results are desirable, and will work to that end, fairly economical operation will obtain. If he does not understand combustion nor know what is required to secure the best furnace performance, indifferent results may be expected.

Increased boiler-plant economy is dependent upon a properly designed furnace, and skillful handling of the fires. Suitable apparatus to enable the fireman to determine when his boilers are or are not efficient is obviously needful. These comprise CO_2 instruments, water weighers, accurate coal scales and such registering instruments as will enable a check being kept upon the everyday performance of the plant.

Boiler Inspection and Engineers' License Laws

The time is at hand when the legislative mills of the several states will commence their annual or biennial grind, and committees from engineers' associations will sit up nights and wrestle with unfamiliar legal phraseology in attempts to evolve license and inspection bills, which will find favor in the eyes of the law makers. The bills will be introduced and referred to committees; the associations will tax themselves to send representatives to hearings. They will present better reasons for such laws than can be presented for seventy-five per cent. of those which pass, and then the legislature will adjourn and the com-

mittee will try to explain to its parent body why it failed.

The reason for most of the failures which mark the struggles of more than a quarter of a century for such legislation is the power of the representative of the interests. The average user of boilers and the employer of engineers looks upon government inspection as an unwarranted interference in his business, and upon a proposition to examine and license engineers, as a conspiracy to make practical engineering a close corporation and artificially restrict the supply of labor in that field. As a result of this conviction and of their influence in the legislatures the United States occupies the proud distinction of killing by boiler "accidents" more people and destroying more property than any other country in the world in proportion to the number of boilers in use.

There should be no question of the duty, to say nothing of the right, of the state to protect its citizens against dangerous boilers or boilers in the charge of incompetent men. Such a provision is not a restriction of personal liberty, or industrial activity, but an assertion of the people, the men and women who walk over and work within range of these magazines of energy, of their right to protection from their abuse.

The adoption and enforcement of such laws in Massachusetts and Ohio has weeded out many cases of incipient explosions and put responsibility where it belongs without working any hardship to the employer who wants things straight and right; and a canvass of the steam-plant owners in those two states would disclose that the law has brought none of the dreaded consequences and has won their favor.

It is very desirable that, as this legislation is effected in other states, as it undoubtedly will be, the laws should be substantially uniform. This is especially true in the requirements as to boiler construction. The Massachusetts and Ohio laws provide for boards of boiler rules. These boards prescribe the boiler which may be run in their respective states, and if similar boards were appointed in many of the states and their specifications were widely different, a hardship would be imposed upon boiler manufacturers and upon engineers who need to be informed upon the requirements of all the states.

To committees appointed to procure the passage of such laws we commend that already in force in the two states mentioned. Unimportant modifications would adapt it to the constitution and the requirements of any state; and if the boards of boiler rules appointed under such laws adopt the rules, which are the results of experience in the states which have practiced them, there will be established a condition of uniformity devoutly to be wished. Meetings of representatives of the boards of the several states with regard to extensions and revisions of the rules would lead to the preservation of unanimity and the bringing to bear upon the question the united engineering talent of the country. The committee on Standard Boiler Specifications of the American Society of Mechanical Engineers may evolve a specification which will be so universally satisfactory that it can be adopted by all, and a boiler built to it will be acceptable in any state in the Union.

"Direct" or "Continuous" Current

At present there is a lively discussion in electrical circles as to the advisability of substituting the term "continuous" for "direct" current. In its strictest sense, direct current implies a current of unidirection in contrast to an alternating current in which the direction is continually reversed, whereas continuous current is not only in one direction, but is non-pulsating. This, of course, could not be applied to a current obtained through a mercury rectifier, which, although direct, is pulsating in character. Thus it is apparent that "direct" is the broader term and can be applied without error to either case whereas it would be incorrect to call a rectified current "continuous."

Several electrical papers and societies have recently gathered the opinions of leading electrical engineers relative to the question. These show that while the term "continuous" does not lack supporters, the majority seem to prefer "direct" current. Moreover, this view has the backing of the Standardization Code of the American Institute of Electrical Engineers, which states that "a direct current is *any* unidirectional current."

The English language as compared with several foreign languages is deficient in technical terminology, and steps to remedy this would receive the approval of most engineers. However, any attempt along these lines which would tend to confuse rather than to clarify the meaning should be discountenanced. The term "direct" current has been sanctioned by long use and it is not incorrect. The only argument that can be put forth against it is that it is not specific enough. However, as rectified current forms only a very small part of the total direct current in use, when a definite designation is required, the terms "rectified" or "pulsating" may be employed.

Decreased Enrollment in Technical Schools

Statistics show that for the past two years the technical schools throughout the country, in general, have reached their maximum in the number of students enrolled and in a few cases are even showing a slight falling off.

At first glance this might seem to indicate that the demand for technical graduates is decreasing. This assumption has proven erroneous, however, by the records of the employment bureaus in the schools affected, which show that the demand greatly exceeds the supply. A significant fact in this connection is that the greatest demand comes from the large companies.

Looking further for the true cause of this condition, we find it to be due largely to two facts—a higher standard of entrance, and the stimulus which the agricultural courses have received lately. The enrollment in the latter is now increasing by leaps and bounds just as that in the engineering courses did a few years back. This may be attributed largely to the agricultural schools being, for the most part, recipients of state aid and having less exacting entrance requirements.

The temporary halt in the number of students in the engineering courses is not to be deplored, however, for the smaller classes can be handled more satisfactorily and given closer personal attention, improving the quality of the product of the engineering schools. Moreover, from a purely economic point of view, the greater the ratio of demand to supply the more the technical graduate will command.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Engine Carries Load after Repair to Shaft Governor

The following case illustrates the value of persistency in overcoming power-plant troubles. Early last fall I was employed as assistant engineer in a refrigeration plant to help out during the fall rush.

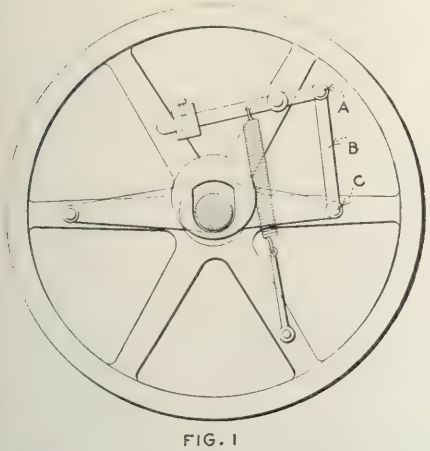


FIG. 1

FIG. 2

SHAFT GOVERNOR AND FAULTY LINKS THAT CAUSED POOR ENGINE REGULATION

This plant was large and lighted by current supplied by a central station, while a good generating outfit stood idle in the engine room.

The reason for not using the set was that the engineers found it impossible to obtain satisfactory regulation of the engine. In justice to them, I must say it was the worst engine in this respect that I ever handled. It would cause a fluctuation of voltage so great as to nearly put the lights out, while the next instant it seemed as though the lamps would burn out from brilliancy.

The chief had worked on it this day with no good results, so I requested permission to overhaul it, which was granted. Referring to Fig. 1, I first removed the nut from the pin *A*, but the pin was rusted in so tightly that it required several hard blows with a heavy copper hammer to force it out. After disconnecting the link *B* at *A*, the pin *C* was removed with difficulty.

The trouble lay in this link, which was so much out of shape as to bind the governor, preventing its proper operation until the terrific speed caused it to let go suddenly.

Fig. 2 shows the link before and after being straightened. As shown, more offset was given to the ends which, after the governor was reassembled, allowed of good regulation. When the chief returned, the central-station switch was open, the engine carrying the load. He smiled and rewarded me by increasing my wages; then I smiled.

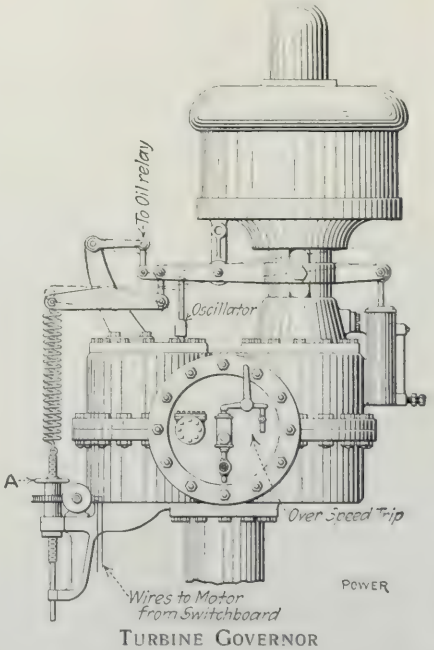
O. L. SHERMAN.

Duluth, Minn.

Why Lightly Loaded Turbine Raced

Our plant is a central-station turbine installation. We had orders from the head office one morning to synchronize with another station 20 miles distant, and to run together until further notice; a 33,000-volt line connected the two stations.

In order to steal some of the load, so as to make a good showing, the chief screwed up on the governor spring adjusting wheel *A* which is used for synchronizing, and raised the speed from 1500 to 1540 r.p.m. Our load is very light from midnight to about 6 a.m., as the other station is cut out about 11:30 p.m. Sometimes the turbine runs along at night for an hour with hardly any load.



TURBINE GOVERNOR

Recently the night engineer reported in the log book that the governor was hunting and varied the speed from 1520 to 1600 r.p.m., so the chief watched the turbine the following day, but could not see anything wrong with the speed or governing, and if it did speed up a little he recorded it as being due to the other station motoring our machine, and did not take any notice of the night engineer's statement. This condition prevailed for a week, until the engineers changed shifts.

The engineer that went on duty at midnight was surprised at the way the governor hunted, because the turbine ran steady in the daytime. The inlet pressure varied from 24 in. vacuum to 100 lb. After studying conditions for a while he lowered the speed to 1500 r.p.m., and the governor ran very steady on the light load.

The illustration shows the governor levers. An oscillating motion is given to the primary and secondary valves, to keep them from sticking, and does not interfere with the speed of the machine. When the sketch was shown and explained to the chief he could not but be convinced that the turbine raced when carrying light loads, and he gave orders to keep the speed at 1500 r.p.m. The trouble was this: At light loads the turbine had to speed up in order to have the governor elongate the already strained spring, as shown by the dotted lines, and this caused the governor to hunt. As the governor came down the spring was less effective, which accounted for the governor running steady while the turbine was loaded.

THOMAS SHEEHAN.

North Adams, Mass.

Wrecked a Coal Hoisting Engine

At a power plant, situated at tidewater, coal is received in barges and hoisted with a steam engine placed about 200 ft. from the boiler house. During recent cold days the steam pipe and cylinder drips froze, owing to the drip valves being left closed. The main steam pipe is shut off in the boiler house and it has a good pitch toward the engine. It is the custom to open the drip on the steam pipe, also the throttle valve and cylinder drips.

As freezing weather had not arrived when the last load of coal was unloaded, these drips were neglected and the condensation from the steam left in the pipe went to the lowest point and froze, but not enough to burst the cylinder or do other harm.

To avoid further trouble, the chief engineer decided to personally look after the opening of the drips. When the hoisting crew went away after finishing the last hoist the chief found the pipe drip open, and, to be sure that no condensation would gather in the pipe, he opened the throttle valve and the cylinder drips. The next morning a new cargo of coal arrived and the hoisting engineer, not knowing what the chief had done, opened the stop valve at the boiler house. When he got back to the hoisting house he found the vertical engine was over on its side, the main bearing caps were off and the shaft was out on the floor; the connecting-rod was broken and the cylinder head was cracked. In fact, the engine was a wreck.

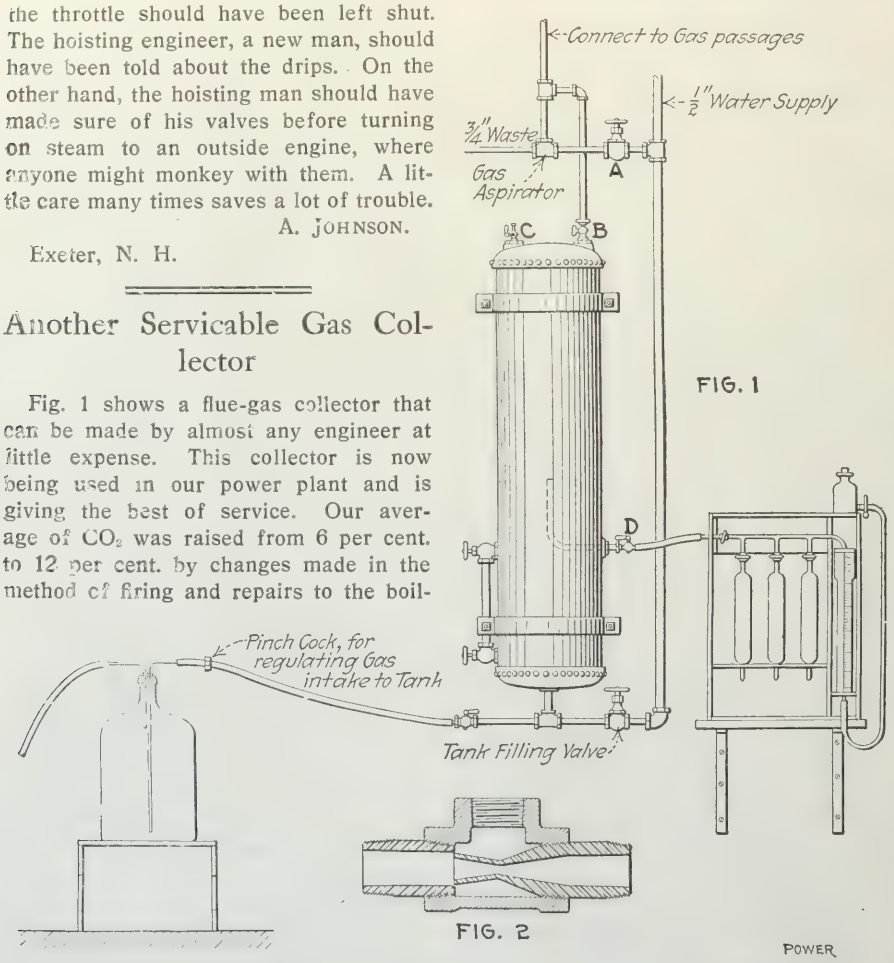
The chief was most to blame, for the main steam-pipe drip should have been piped separate from the engine drips and the throttle should have been left shut. The hoisting engineer, a new man, should have been told about the drips. On the other hand, the hoisting man should have made sure of his valves before turning on steam to an outside engine, where anyone might monkey with them. A little care many times saves a lot of trouble.

A. JOHNSON.

Exeter, N. H.

Another Servicable Gas Collector

Fig. 1 shows a flue-gas collector that can be made by almost any engineer at little expense. This collector is now being used in our power plant and is giving the best of service. Our average of CO_2 was raised from 6 per cent. to 12 per cent. by changes made in the method of firing and repairs to the boilers suggested by using the collector and Orsat. Mechanical stokers are used here.



SHOWING CONNECTIONS FOR PERMANENT FLUE-GAS COLLECTOR

The tank is a 52-gal. house boiler. The aspirator, Fig. 2, is made chiefly from a ½-in. standard tee, and a small brass nozzle. The pipe in the uptake is of ¾ in. diameter, perforated its full length with ⅛-in. holes. To this is attached the pipe leading to the aspirator, which is ⅜ in. The water-supply pipe is of ½ in. diameter.

Before filling the tank with water it is necessary to place therein about 1 gal. of machine oil, which forms a seal and prevents the water from absorbing any of the elements of the sample. Next the valves *A*, *B*, *D* and *E* are closed and the air vent *C* and filling valve are opened. When the oil purges from the vent *C*, the latter and the tank-filling valve are closed. Then *A* is opened and the water allowed to run through the aspirator, a section of which is shown in Fig. 2. When it is certain that gas is being drawn from the uptake, the valves *E* and *B* are opened, and the flow of water from the tank is regulated until the flow is correct to leave the tank empty in the desired time.

We allow 4 hr. as the time for taking a sample. The gas from the flues will take the place of the water and when the water in the tank is low enough it can be seen in the gage-glass, and the valves *B* and *E* should then be closed. The pipe in the center of the tank to which the valve *D* is connected has a hole in it whereby the water can drain out so that it will not run into the Orsat apparatus.

By carefully filling the tank the oil seal can be made to last indefinitely.

ROBERT H. SCHAFER.

Trenton, N. J.

Condenser Problem for Discussion

An old cotton mill well known to me derived its motive power from a beam engine equipped with a jet condenser. The top of the condenser was 17 ft. above the pool from which the condensing water was drawn. This pool also served as a hotwell. The water was usually very hot as much of it was circulated in the pool, which was really the dead end of a canal.

The injection water entered the condenser by virtue of the vacuum in that vessel. The water constantly flowed into the condenser and was as steadily lifted out by the air pump and delivered again at a point 17 ft. above the canal, into which it descended by a pipe and entered the canal horizontally with such force as to show a strong current almost entirely across the canal. So strong was the current that it often led to discussion as to the advisability of placing a small water turbine at the outlet of this air-pump discharge pipe to use this 17-ft. fall.

Now what bothers me at present is where the energy to lift that water into the condenser came from. Was the work done in any way by the engine, or was it a work product of the steam that did not cause more fuel consumption than if the water had been drawn into the condenser from a tank only a foot below the intake? In both cases the air pump would do equal work, just lifting the water from the condenser base and putting it outside.

But the 17-ft. lift was done somehow. Of course, the real acting agent was the external atmospheric pressure which forced the water into the injection pipe; but it could not have done this if the atmospheric pressure had not been removed from the condenser.

Can it be shown that the effect of the vacuum in reducing the back pressure on the engine piston was any the less because the water flowed into the condenser by virtue of the vacuum within it and was not pumped in or allowed to run in by gravity? It is, of course, not to be forgotten that even if pumped in or forced in by gravity the external atmospheric pressure would assist the flow of water. But, in the instance in question, work was done against the opposing force of gravity and sometimes in summer gravity got the "upper hand," stopped the injection water supply and also the engine and filled the place with steam.

If it could be shown that the vacuum was or should be a trifle less as a result of the water climbing up against the action of gravity it would aid one to understand this problem. I feel that, though it could not be experimentally determined, the energy was supplied by the steam

and that more steam was used than would have been necessary if the water did not have to be lifted. Otherwise one seems to be hovering about some form of perpetual motion and this, we know, is out of the question.

Can anyone give a clear explanation of this problem?

W. H. Booth.

London, S. E., England.

Air Receiver Exploded

The shell of a 24x72-in. air receiver, carrying 90-lb. pressure, split from end to end in a straight line along the bottom where the metal had wasted away to $\frac{1}{16}$ in. in thickness, at the end where the rupture started. This wasting seems to have been caused by water carried in by the air in the form of moisture. The interior of the shell was coated with a black ooze. Is this a common occurrence?

George Russell.

Spring City, Tenn.

Using Inferior Packing

Possibly my experience will interest those who are compelled, by force of circumstances, to use inferior rod packings.

I order all of my supplies through a purchasing agent and usually get what I order, but the last supply of packing received was not the kind I had ordered. It was of such an inferior quality that the maker must have been ashamed to put his name on it, as there was no name nor identifying trade mark to be found.

I packed the main engine piston with it and the trouble that followed almost distracted me. If I pulled the gland up only moderately tight steam would blow through, almost as if there was no packing in the stuffing-box, and if I tightened up on it, it would be smoking hot in a few minutes. Finally I took it all out and put a new ring of it in the bottom of the stuffing-box. Then I took rope asbestos well soaked in cylinder oil and graphite and wound this around the rod, filling the stuffing-box almost full, then I put in another ring of the packing on top of the asbestos rope. I pulled the gland up fairly tight, then slacked it off and screwed the nuts up finger tight and my packing troubles were over. I have now packed the engine three times in this way and one packing lasts about two months. There is not the slightest leakage of steam, and the friction on the rod is negligible. I also find it satisfactory for packing the air piston rods of the air compressors. I concluded it was cheaper to use the packing in this way instead of throwing it all in the scrap pile.

W. G. Camp.

Rock Crusher, Texas.

Dirty Boiler Indicated by Steam Chart

The two cards reproduced herewith, taken from a recording steam gage, show the importance of keeping the tubes of boilers free from soot. These charts were taken during two days' run and with loads practically the same. The fireman was in the habit of not cleaning the tubes except once a month, and the day came when he "punched" them clean. Nothing was said about the steam until the end of the run at 6 p.m., when his attention was called by the chief to the straight steam line on Fig. 2 in the engine room. Much to the fireman's surprise it was at once decided to investigate the daily record of the number of wheelbarrow loads of fuel used on the day that Fig. 2 was taken, which showed that 56 loads were burned, while the day following, when Fig. 1 was taken, 39 loads only were required.

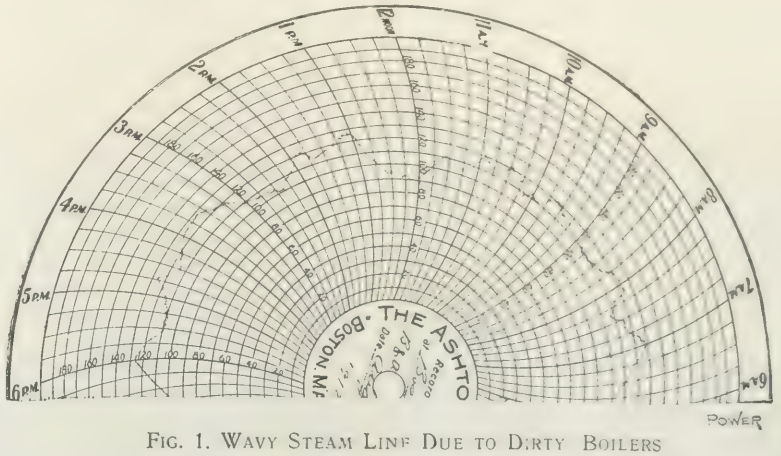


Fig. 1. Wavy Steam Line Due to Dirty Boilers

The damper closes at 126 lb. gage and Fig. 2 shows that the damper was open almost constantly, it being nearly impossible to keep up the pressure. Fig. 1 shows the pressure was maintained uniformly with greatly reduced labor.

The interesting feature of it all is that the fireman did not have the recording gage in mind at any time through the day until he was called to the engine room at night. Therefore, it shows a clear disinterestedness as regards a straight steam line.

It will be noticed on the charts that the run was from 7 a.m. to 12 a.m. and from 1 p.m. to 6 p.m., after which no special pressure is necessary through the night.

A. C. Waldron.

Revere, Mass.

Cleaning Boiler Fire Boxes

A 50-hp. boiler of the firebox, return-flue type, became incrusted in the water legs with scale and bagged between the staybolts and finally cracked in two places.

The boiler inspector ordered the sides of the firebox bricked up and the water fed from the top at the rear. The four hand plates of the water legs were set in the wrapper sheet so far toward the center of the boiler that it was impossible to use a cleaning tool inside the water legs.

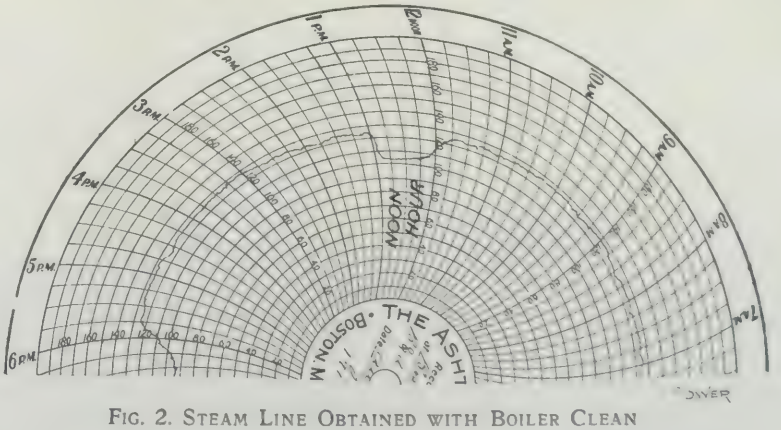


Fig. 2. Steam Line Obtained with Boiler Clean

We drilled and tapped the outside sheet for 1¼-in. pipe plugs, and were then able to keep the fire sheets clean by using a small rod scraper. We have had no more trouble with sheets burning since these changes were made, as nearly all the scale has been removed.

Paul Stuart.

Nebraska City, Neb.

Questions Before the House

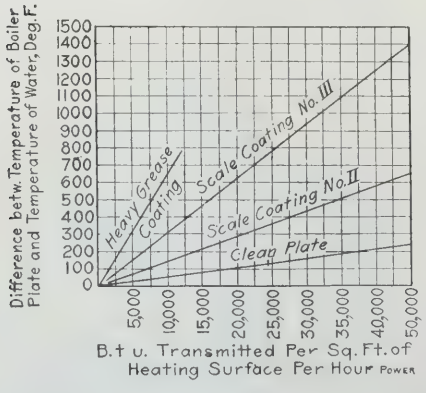
Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Overloaded Boilers

In your editorial, Oct. 29, you very properly point out the dangers incidental to ignorant handling of boilers. The source of danger to which you specifically call attention is not, however, the chief danger, nor the principal cause of boiler accidents. The impression received from reading your editorial is that overloading, that is, forcing boilers to carry peak loads, is a serious menace.

Before accepting this view, however, would it not be well to analyze the processes which go on when a boiler is overloaded?

Overloading, as I understand it, means evaporating more water per hour from each square foot of boiler surface than the boiler is rated to evaporate, involving, of course, the burning of more coal upon the same grates. It will be conceded that the evaporation of more water, involving the transmission of more heat, implies a higher temperature of the boiler plate, but how much higher? This has been studied by several European investigators, and the accompanying chart, prepared by Dr. E. Reutlinger, of Cologne, shows that in the absence of scale or other coating, the rise in temperature of the boiler plates or tubes is very small. For instance, when 40,000 B.t.u. are being transmitted through each square foot per hour, the rise in temperature of the clean plates is only a little over 200 deg. This corresponds approxi-



mately to 40 lb. of water per hour; that is, to 1⅓ hp. per sq.ft., or ¾ of a sq.ft. per horsepower, as against the ordinary builders' rating of 10 sq.ft. per horsepower. As the table shows, however, with scale present the temperature would rise 600 to 1200 deg. or more, which would certainly result in injury to the plate or tubes, and, as Dr. Reutlinger says, "In the interest of safety, the use of softened feed water or frequent boiler cleaning is to be emphatically advised."

We come to the same conclusion, that boilers are ordinarily injured by accumulations of scale, sludge or other coatings, rather than by the mere fact of overdriving, if we examine the statistics collected by boiler-inspection and insurance companies. Of 207,465 boilers examined in 1905 by the Hartford Steam Boiler Inspection & Insurance Co., 117,142, or about 56 per cent., were defective in ways which could be traced to impure feed water, and of 212,211 boilers examined by the same company in the year following, 119,096 were defective from causes due to impure feed water, which is likewise about 56 per cent. of the number examined.

The following table, abstracted from *The Locomotive*, the official organ of the Hartford company, reveals the same fact:

Whole number of defects:	1905	1906	1907	1908	1909	1910
Deposit of sediment	17,757	18,624	18,917	18,829	20,644	19,471
Incrustation and scale	38,965	38,753	38,427	37,924	41,541	43,663
Internal grooving	2,783	2,767	3,010	2,649	3,188	2,830
Internal corrosion	13,109	13,161	12,862	13,053	14,086	13,781
Burned plates	5,181	5,332	4,878	4,605	4,887	5,174
Leakage around tubes	11,768	11,036	11,357	10,929	13,423	13,015
Defective tubes	4,503	7,266	8,266	8,026		9,691
Leakage at joints	6,088	6,005	5,557	4,845		5,956
	100,154	102,950	103,214	100,910		113,581
Number of dangerous defects:						
Deposit of sediment	1,192	1,144	1,315	1,242	1,235	1,367
Incrustation and scale	1,098	1,310	1,333	1,193	1,251	1,468
Internal grooving	235	262	258	249	259	229
Internal corrosion	556	516	528	555	576	611
Burned plates	520	558	499	440	368	478
Leakage around tubes	1,381	1,381	1,599	2,103	2,039	1,789
Defective tubes	1,341	1,912	3,054	2,136		2,508
Leakage at joints	401	364	430	392		353
	6,724	7,447	9,016	8,310		8,883
Total number examined internally	155,024	157,462	159,283	151,359	136,682	138,900

Similar figures from the Fidelity & Casualty Co. are as follows:

	1905	1906
Internal corrosion and pitting	1,722	1,372
Deposit of sediment	747	914
Scale and incrustation	2,815	2,167
Burned plates	136	194
Blistered plates	161	208
Bagged plates	972	707
Leaky tube ends	731	682
Leaky seams	680	647
	7,694	6,951
Total number examined internally	34,046	35,428

From the Maryland Casualty company's report, take the table on the next page.

Finally, is there any reason why we should expect a boiler to be injured particularly by overdriving? Unless the amount of air per pound of fuel, or the grade of the fuel has been changed, the temperature of the gases of combustion will not be altered, but the gases will be supplied to the boiler in much greater quantities, and the hot gases will penetrate farther into the nest of tubes. The boiler, it is true, must disengage steam more rapidly, but this is not so apt to injure the boiler as it is to injure the engine or turbine to which the steam is supplied. The only danger to which overdriving exposes a clean boiler is failure to circulate the water rapidly enough to keep the heating surfaces wet at all times. This is the fault of the boiler designer, but the fact that boilers are being regularly operated at overloads of 200 and 300 per cent. shows that it is not beyond his ability to design boilers to meet these conditions. With free open passages without constricted throats, and with tubes of reasonable length, there is no difficulty in keeping the tube surface wet. This applies to fire-tube boilers as well

An examination of this table shows that a very large proportion of all the defects, that is, deposits of sediment, incrustation and scale, and internal corrosion, are certainly to be attributed to the character of the feed water, while the remaining defects, that is, internal grooving, burned plates, leakage around tubes, defective tubes, and leakage of joints, are in most cases undoubtedly due to the same causes, that is, the plates would not have been burned had not they been insulated from the water by a coating of scale or sludge, and the tubes would not have been injured or have become loose from their fastenings, had they not been overheated, due to the same insulating coating of scale.

	Total defects			Dangerous defects		
	1905	1906	191-	1905	1906	1910
Deposit of sediment	991	835	1,765	155	207	293
Incrustation and scale	2,532	2,535	4,701	358	445	405
Internal corrosion	1,371	1,280	1,877	104	340	189
Pitting	1,495	1,432	1,928	83	217	183
Leaky tube ends	1,778	1,933	2,872	454	516	936
Bulges, sags or bags	857	1,180	2,339	208	281	469
	9,024	9,195	15,482	1,362	2,006	2,475
Total internal inspections	18,395	19,321	26,779			

as to water-tube boilers, as witness the high rate at which locomotive boilers are regularly operated.

Speaking of locomotive boilers, we again here find confirmation of my contention that the injury to the boilers is generally due to scale and not to overloading. In 1902, the Chicago & Northwestern Ry. installed water-softening equipment at 17 stations. The results upon engine performance have been summarized as follows:

	1902	1903
Ton mileage, in million ton-miles	2,934	3,154
Increase in ton-mileage		220
Coal burned per 100 ton-miles lb.	28.7	27.5
Savings coal burned per 100 ton-miles lb.		1.2
Saving in coal (at $3 per ton for 3000 million ton-miles)		$30,000
Average assignment of engines (83 per cent. of which were in constant service both years)	159	154
Saving in assignment of engines		5
Saving in engines (10 per cent. interest and depreciation) on cost of 5 engines at $10,000 each		$5,000
Number of boiler makers employed	36	23
Number of helpers employed	42	30
Total number of laborers on boilers	78	58
Saving in laborers		20 or 26 per cent.
Saving in wages of boilermakers and helpers		$7,700
Saving in material for boiler repairs	Not estimated, but known to be large	
Boiler failures from leaky flues	544	99
Boiler failures from leaky fire-boxes	33	20
Boiler failures from leaky arch tubes	6	1
	583	120
Reduction in number of boiler failures (79 per cent.)		463
Estimated yearly saving to road, all causes		$75,000

A great diminution in the number of boiler failures was obtained by softening the water, but probably without changing the average rate of steaming in any way, or if anything increasing it, due to the greater steam-making capacity of clean boilers.

So far I have considered only the injury to boiler metal due to overheating, but boilers are also injured by corrosion, which can generally be prevented by the same treatment that is employed to prevent scale.

To sum up, then, may we not justly say that boiler disasters are due, not so much to overloading, in the way of hard driving, as to overloading with ignorance and neglect on the part of owners, designers and operators? When these facts are clearly understood in their true significance by all parties concerned, owners will insist that the designers of boiler plants provide for properly treating the feed water, when making the original installation, and that operating men see that such apparatus is properly used to insure the intended results; and on the other hand, intelligent operating men will refuse to work for owners who will not provide equipment for this purpose. It should not be difficult to secure the desired ends, once the situation is clearly understood, since the reduced cost of boiler repairs and replacement, and the fuel saved, will amply repay the cost of treating the water.

WILLIAM B. CAMPBELL.
Philadelphia, Penn.

[If Mr. Campbell will again read the editorial referred to he will find that it aimed to give the impression that overloading boilers is exceedingly dangerous *when they are not cared for properly.* Boilers will carry heavy overloads if kept clean and free from structural defects. This is so generally known, in fact, it is so generally practiced, as to have needed no comment in the editorial.

What we did condemn was the practice of heavily overloading boilers as a substitute for inadequate boiler capacity and at the same time grossly neglecting them because the exigencies of the service will not permit taking them out of service long enough to properly clean and examine them.

The editorial said in part: "But these boilers, the best, cannot long perform their function properly if overloaded, forced to the limit and left to care for themselves until so hurriedly examined that serious defects are still undiscovered, when they are again put in service."—EDITOR.]

Graphite in Boilers

Discussion of the use of graphite in boilers was invited by the editorial in the Sept. 17 issue. To that which appeared in the Nov. 19 issue, I would add the following from my experience:

Ideal boiler water is hard to get. If the scale-forming substances are absent the corrosive element is there to eat away the shell and tubes, which, as far as ultimate safety is concerned, is worse than scale. In some localities the water is so good that boilers can be operated an entire year without the necessity of cleaning. Such water is found in many places where the supply is a natural surface water in which the percentage of impurities is so low that no precipitation will adhere to the boiler interior and is just sufficiently saturated to prevent the oxidization which a pure rain water would cause.

Two batteries of nine and ten boilers each in different institutions in the vicinity of Trenton, N. J., fed with artesian well water, contained considerable scale. The analyses of the water showed 10 and 15 grains of incrustating matter per gallon. In one battery compound treatments were tried with varying results. It proved best to have the water analyzed by a disinterested chemist and the compound prescribed and bought in the open market. The analysis disclosed the presence of lime and magnesia carbonates and sulphates in smaller quantities. The scale was considerable in amount, but not very hard. We were advised to use 5 lb. of soda ash and ½ lb. of caustic soda or 6 lb. of caustic soda ash for each 150-hp. boiler. This latter substance costs about 3½c. per lb.

These boilers were fed from a hotwell which received hot water from the apparatus used in the factory. The temperature of the feed water from this source ranged from 100 to 150 deg. F. Some of the boilers were fed through a feed-water heater of the closed type, but the rest had no heater of any kind. For the first the compound was dissolved and fed through a bypass in the feed line. In the boilers having no heater it was the custom to dissolve the compound and throw it into the hotwell once a day, at any convenient time. In neither instance were the results very satisfactory. This compound, as a reagent for neutralizing the scale-forming matter, was the best we had used, yet like the others it made little, if any, progress in the dissolution of the old scale.

On taking charge of these boilers, I found them with scale from ⅛ to ½ in. thick on the shell and tubes. At the heads it was still thicker and thin pieces had collected between the tubes and become cemented into a compact mass extending from 2 to 4 in. from the heads. Tubes were continually leaking and many had to be removed because of being burned.

Several of the boilers had been bagged. In many places the vertical spaces between the tubes were closed. More than half of the tubes in the feed-water heater were closed from end to end. Under these conditions steam could not be kept up and frequently one of the main engines had to be shut down. In cleaning out the hotwell it was found half full of a dirty white sludge, the nature of which was at first not known.

Investigation showed that the compounds were causing the precipitation. Had the compound been used more intelligently by feeding continuously, instead of being thrown in all at one time, better results would have been obtained. The parties selling the compound were

responsible for this method of feeding and there are some who advise it still.

Improvements in the manner of feeding the compound were made. Each boiler was opened once in four weeks, but with continued effort it took three years to get the run of the tubes bare underneath, although the scale on top of them was anywhere from ⅛ to ¼ in. thick. At the heads very little improvement was made, notwithstanding that efforts were made to cut it away with bars and chisels. As the only time available for cleaning was between Saturday night and Sunday night, in which the boilers had to be cooled, cleaned and the fire relighted, they were always too hot for one to stay in them very long.

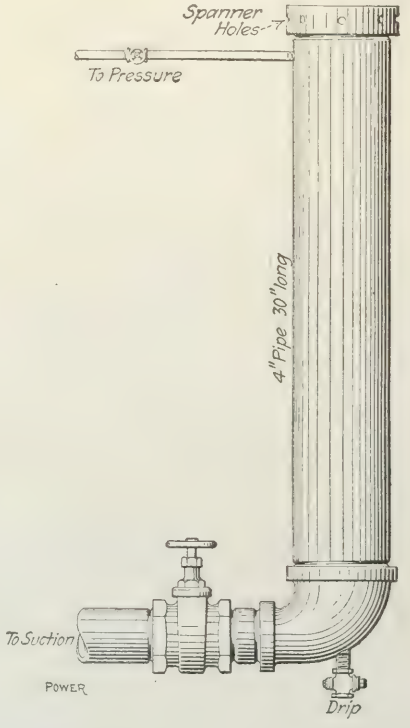
Not being able to convince the management of the value of heater purifiers, water softeners or cleaners for removing scale, all the information possible concerning compounds was sought. Almost everything was advised from sugar to coal oil, among them graphite. As a last resort, before settling down to the inevitable, a trial keg of graphite was ordered. It was difficult to see how such a neutral, insoluble element could have any effect on scale, but it was recommended by a friend whose experience gave his advice much weight. Its use was begun according to directions and results were anxiously awaited.

The first sign was the sound of scale being driven through the blowoff pipe. On opening the boilers for the first time after beginning the treatment a wheelbarrow full of scale was removed from each. Patches of scale here and there were already breaking from the shell, something that rarely occurred before. Apprehension was felt lest the lumps of scale should collect and cause the fire sheets to bulge, but the blowoff was used four times a day and it was found that the scale no longer cemented together or collected on the boiler. This treatment was continued and in about eight months the boilers were pronounced clean by the inspector.

In assuming charge of another plant of ten boilers, fed by deep-well water, the conditions were bad. The scale was hard and the boilers looked as though they were lined with porcelain. The same treatment was begun with the same result—a slow but steady disintegration of the scale. The water in the boilers became saturated and was emptied out of them at least once a month. While no case of priming was observed it is only reasonable to assume that with 20 or 30 lb. of graphite suspended in the water, ebullition could not be as free as with clean water. For this reason and to remove any scale that might have fallen on the fire sheets frequent cleaning was carried on until the boilers were clean; then the amount of graphite fed to the boilers was reduced.

There seems to be but one kind of graphite from which results can be had. It is of an amorphous noncrystalline character capable of being reduced to an impalpable powder, of such fineness that it will float in the air like dust and penetrate the smallest interstices in the scale deposit.

No known chemical action occurs in the use of this material for the removal of scale; its action seems to be so purely mechanical. In the scale deposit minute cracks are made by the expansion and contraction of the boiler and into these openings the graphite finds its way, coating the boiler surface and preventing further adhesion until the cracks unite when the pieces fall away. Obviously, a boiler that is run continuously and



GRAPHITE FEEDER

whose temperature is almost constant will not respond to the use of graphite as well as a boiler that is cooled down and washed out every two or three weeks. When the boiler is cooling, contraction breaks the scale and loosens it in spots, and when the water holding the graphite in suspension is fed into the boiler it readily finds its way into these interstices between and under the scale which sooner or later falls away.

No matter how carefully and thoroughly a boiler is washed out, considerable quantities of scale will fall and it will be necessary to open the blowoff valve frequently during the first 24 hours after raising steam to prevent small particles from choking the blowoff pipe. When the boiler is opened and the water let out a black, pasty looking substance will be found. This is the precipitated scale-forming matter mixed with graphite and it should be washed out before it dries. The boiler should be blown down regularly, the frequency depending upon the character of its service.

Graphite should be fed directly into the boiler and not thrown into tanks, hotwells or cisterns, because it will settle if the water is not in motion.

For feeding graphite the device shown herewith will be found to be all that is required. It is made of a piece of 4-in. pipe (which is large enough for 4 lb. of graphite), with a cap fitted so as to be water-tight on top, the bottom is connected to the suction pipe and is fitted with a gate valve or cock. The small valve at the bottom is for a drip and the ¼-in. pipe at the top should be connected to the pressure side of the pump. When the 4-in. pipe is being charged with graphite it should be half full of water. The graphite should then be thrown in the chamber and more water poured on top until the pipe is full. After the tank is closed the lower valve is opened a little and the upper one just cracked. If the boilers are in a battery, care should be taken that each boiler receives its share of graphite.

Steam from these boilers has been admitted to vessels containing white fabric and also used for cooking, the steam coming in direct contact with the food yet no complaint has been made because of the graphite being carried over with the steam.

Graphite is not a substance that can be used carelessly or with indifference. Unless proper care is taken to keep the blowoff pipe clear and means taken to remove the loosened scale its use is dangerous. Engineers who are not willing to use necessary precautions are advised not to use graphite, because it will bring scale down and sometimes bring it down fast.

WILLIAM HIRST.

Trenton, N. J.

By the blowing out of a nonreturn automatic stop valve, connected between the boiler and the main steam header, the assistant engineer of the Los Angeles, Calif., city lighting plant was so badly injured that he died two hours later.

The engineer had opened the valve from the header to connect No. 1 boiler with the other boilers. According to a report received, he discovered that the valve body was leaking and, calling to the chief engineer, announcing the fact, started for the ladder to descend to the floor, when a piece from the side of the valve, about 3 in. in diameter, blew out, striking him in the chest, throwing him from the top of the boiler to the boiler-room floor, where he struck on his head, fracturing his skull and breaking three ribs and his collar bone.

We are informed by a correspondent that the fractured piece showed no indication of being an old break. It was a trifle less than ¾ in. thick, and the steam pressure was between 175 and 200 lb.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Material for Grinding Valves

What material is good for grinding brass valves?

C. F.

Powdered glass mixed with a light oil to form a paste is one of the best known materials for grinding valves or other brass work.

Appearance of Different Temperatures of Fire

What temperatures of fire on a boiler grate correspond to different appearances of color?

J. B.

According to M. Pouillet, temperatures of fire corresponding to different appearances are as follows:

Red, when just visible	977 deg. F.
Dull red	1290 deg. F.
Dull cherry red	1470 deg. F.
Full cherry red	1650 deg. F.
Clear cherry red	1830 deg. F.
Deep orange	2010 deg. F.
Clear orange	2190 deg. F.
Dull white	2370 deg. F.
Bright white	2550 deg. F.
Dazzling white	2730 deg. F.

Interference of Steam Heating Returns

In a gravity return steam-heating system, how may the returns of coils and radiators which are on the same floor level be connected so that without use of check valves, one coil or radiator will not fill up with water from another?

F. B.

A coil or radiator can usually be protected against flooding from other heating surfaces by having its return dropped low enough before being connected to a return main or to other returns. Good circulation can usually be secured by thus independently dropping returns of the same floor a distance of three or more feet.

Efficiency of Alternator

What is the percentage of efficiency of a three-phase alternator which requires 335 net horsepower for developing 63 amperes in each wire at 2200 volts, the power factor being 95 per cent.?

J. J. B.

The output of the alternator is given in kilowatts by the formula

$$kw. = \frac{\sqrt{3} \times volts \times amperes \times power\ factor}{1000}$$

in which the voltage is that across any two wires and the amperage the amount in each wire. Substituting the values given the formula becomes,

$$kw. = \frac{1.732 \times 2200 \times 63 \times 0.95}{1000}$$

$$= 228.05\ kw.$$

As 1 hp. is equivalent to 0.746 kw., 335 hp. would be equal to 335 × 0.746 = 249.91 kw.; therefore the efficiency of the alternator is

$$\frac{(228.05 \times 100)}{249.91} = 91\tfrac{1}{4}\ per\ cent.$$

Reversing Turbine Driven Shafts

Is there any practical method of reversing a turbine engine, and what is the means employed for reversing turbine driven steamship shafts?

H. P.

To reverse a steam turbine would require the direction of the nozzles and blading to be reversed, and although possible, in the present stage of turbine development, seems hardly practicable. The method employed in marine practice for reversing the motion of steam turbine-driven shafts is to provide an auxiliary turbine on the same shaft as the ahead turbine, with the blading of the reverse-motion turbine set appropriately for astern motion, steam being admitted to one turbine at a time according to the direction of motion desired, the turbine which is out of use being carried idle.

Vacuum Required for Steam Heating

What vacuum should be maintained in the return line from the heating system of several buildings?

E. T. B.

The requirements vary with different plants and conditions, but generally no more vacuum should be carried at the pump than is necessary to insure the desired activity of circulation in the system. A very small drop in pressure is usually sufficient for good operation of coils and radiators, but in most plants the sizes of return pipes are generally so small that to obtain a perceptable drop of pressure at the heating surfaces 5- to 10-in. vacuum has to be maintained at the discharge end of the returns. Satisfactory results are usually obtainable with 10 or less number of inches of vacuum and in most plants where engine exhaust is used for supplying steam to a vacuum system, no advantage is derivable from carrying more vacuum for the purpose of reducing back pressure on the engine.

Size of Feed Pump

What would be the size of cylinder required for a single double-acting feed pump with 5-in. stroke; having piston rod 1 in. in diameter making 60 single strokes per minute, working with 10 per cent. slip and having capacity to supply by continuous feeding a boiler evaporation of 4000 lb. of water per hour?

W. J.

Evaporation of 4000 lb. of water per hour would be at the rate of $\frac{4000}{60} =$ 66.66 lb. per min. Taken at 62 deg. F., 1 cu.in. of water weighs 0.03609 lb. and

$$\frac{66.66}{0.03609} \quad 1847.04\ cu.in.$$

of feed water would be required per minute. Allowing 10 per cent. slippage, the pump would have to make an actual piston displacement of

$$\frac{1847.04 \times 100}{90} = 2052.26\ cu.in.$$

per minute.

The displacement on one side of the pump piston would be reduced during 30 strokes per minute by the volume displaced per stroke by the piston rod. The diameter of the rod being 1 in., its area is 0.7854 sq.in. and for 5-in. stroke and 30 strokes per minute the total reduction of piston displacement, due to the rod, would be 0.7854 × 5 × 30 = 117.81 cu.in. Adding this to the net piston displacement required gives 2052.26 + 117.81 = 2170.07 cu.in. total volumn of cylinder to be displaced per minute. There being 60 strokes per minute and 5 in. length of stroke the lineal displacement per minute would be 60 × 5 = 300 in., and, therefore, the cross-sectional area of the cylinder should be $\frac{2170.07}{300} = 7.2335$ sq.in. area, which corresponds to

$$\sqrt{\frac{7.2335}{0.7854}} = 3.03\ in.\ diameter$$

Therefore under the conditions a 3x5-in. pump would be required.

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Ratio and Proportion

If of two pocket knives one cost a dollar more than the other, we should consider that there was a big difference in price, but if of two boilers one cost a dollar more than the other, we should consider that there was very little difference in price. The dollar has the same value in both cases, and the dollar saved by not paying twice as much as a knife is worth would not buy more tobacco than a dollar, which is such a trifle of the whole cost of a boiler.

The difference in these two cases is that the dollar bears a widely different *ratio* to the total cost. In the case of the knife it might equal or even exceed the total cost of the tool, but in the case of a boiler of usual size it would be but a very small fraction of the total cost.

From the above comparison we now know that

Ratio is the relation that exists between two quantities.

There are two kinds of ratio—the *arithmetical* that considers numbers in their differences as, $8 - 6 = 2$, and *geometrical* that is the quotient of one number or quantity divided by the other, as $\frac{6}{2} = 3$.

A series of numbers are said to progress in an arithmetical ratio when the differences between any two numbers is the same, as

1, 2, 3, 4, 5, 6; or 1, 3, 5, 7, 9;

while a series of numbers progresses in a geometrical ratio when the quotient of any two numbers in the series divided by the preceding number is the same, as 2, 4, 8, 16, 32, 64; or 2, 6, 18, 54, 162.

When the word *ratio* is used alone, a *geometrical* ratio is always understood, that is, the part that one number is of another. Thus the ratio of 4 to 8 is 2, of 10 to 5 is ½, etc. This can be expressed in the form of a fraction as:

$$\frac{8}{4} = 2; \quad \frac{5}{10} = \frac{1}{2};$$

or, by placing the two dots between them, thus, 4:8 10:5.

In either case they are read "the ratio of 4 to 8" and "the ratio of 10 to 5. The dots are commonly used.

The *Terms* of a ratio are the two numbers compared.

The *Antecedent* is the first term.

The *Consequent* is the second term.

A *Direct Ratio* arises from dividing the consequent by the antecedent. Thus the direct ratio of 5 to 15 or $5:15 = \frac{15}{5} = 3$.

An *Inverse Ratio is obtained* by dividing the antecedent by the consequent. Thus the inverse ratio of 5 to 15 or 5: $15 = \frac{5}{15} = \frac{1}{3}$.

All ratios are understood to be direct unless otherwise specified.

Sometimes an inverse ratio is called a *reciprocal ratio.* The *reciprocal* of a number is 1 divided by that number as, $1 \div 8 = \frac{1}{8}$. Here the reciprocal of 8 is ⅛.

A *Simple Ratio* consists of a single couplet, as 3: 12.

A *Compound Ratio* is the product of two or more simple ratios. Thus the compound ratio formed from the simple ratios 3: 6 and 8: 2 is

$$\frac{6 \times 2}{3 \times 8}$$

$$3 \times 8 : 6 \times 2 = \frac{6}{3} \times \frac{2}{8} = \frac{12}{24} = \frac{1}{2}$$

Here we have two ratios, that between 3 and 6 and that between 8 and 2. Writing them in the fractional form, and multiplying, we have the fractional representation of a new ratio, namely, that of the new compound denominator to the new compound numerator, which is found by multiplying and reducing, to be ½ or 2: 1.

To find the ratio between two numbers divide the consequent by the antecedent.

$$Ratio = \frac{consequent}{antecedent}$$

Example: In a boiler having a grate 5 ft. square and 790 sq.ft. of heating surface, what is the ratio of grate to heating surface?

The grate surface, $5 \times 5 = 25$ sq.ft., is the antecedent and the heating surface, 790 sq.ft., is the consequent, then the

$$Ratio = \frac{790}{25} = 31.6 \; ans.$$

Rule: *To find the antecedent divide the consequent by the ratio.*

$$Antecedent = \frac{consequent}{ratio}$$

Example: What must be the length of an engine crank in order that the ratio of the crank to the connecting-rod shall be 7, when the rod is 12 ft. long?

Here the ratio is 7, the consequent 12 and the antecedent or crank $\frac{12}{7} = 1\frac{5}{7}$ *ft.*

To find the consequent, multiply the antecedent by the ratio.

$$Consequent = Antecedent \times Ratio$$

Example: The ratio of the diameter of a circle to its circumference is 3.1416. What is the circumference of a circle having a diameter of 4 in.?

In this example 4 is the antecedent and 3.1416 is the ratio, and the circumference or consequent is

$$4 \times 3.1416 = 12.5664 \; in.$$

Always express a ratio in the most convenient form, i.e., if we have the ratio 48: 24 we should reduce it to 2: 1. This does not change the value, since 24 goes into 48 two times and 48 is therefore twice as great as 24.

Proportion

We know things by their differences. No comparison of two numbers can be fully explained except by instituting another comparison. The mind does not grasp the relation of 4 to 8 from being told that their ratio is 2 until it is understood that in saying that the ratio of 4 to 8 is 2, we mean that 4 bears the same relation to 8 that *one* or unity bears to 2. The ratio is, therefore, the *consequent* of another pair of terms, having the same relation to each other as the first pair, and the antecedent of which is unity, as

$$4:8 = 1:2$$

or 4: 8: : 1: 2, which is read 4 is to 8 as 1 is to 2. Or better, the ratio of 4 to 8 equals the ratio of 1 to 2.

Numbers which have to each other such an equality of ratio are said to be in the same proportion. From this statement it is obvious that 4 bears the same proportion to 8 that 3 does to 6, the ratio being 2 in each case. This would be expressed

$$4:8 = 3:6$$

and this arrangement of figures is called a proportion and the purpose of proportion is to enable one to find a fourth or unknown quantity when three other quantities are given or known. The unknown quantity is always designated by the letter X, as

$$4:8 = 3:X$$

From the foregoing we see that a *proportion is a statement of equality between two ratios.*

The Extremes of a proportion are the first and fourth, or the outside terms.

The Means of a proportion are the second and third, or the inside terms

$$4:8::3:6$$

In the above proportion 4 and 6 are the extremes and 8 and 3 are the means.

A proportion is made up of ratios and each ratio of any proportion is called a *couplet.* In the above proportion 8: 4 is a couplet and 3: 6 is also a couplet.

Three numbers may be in proportion when the first is to the second as the second is to the third, as in

3: 9: 27

where the ratio of each couplet is 3. In this case the second term is said to be the *mean proportional between* the other two. It is simply a common proportion in which the two terms of the means are alike, and could have been written

$$3:9::9:27$$

The chief thing to remember about proportion is that

The product of the means of any proportion is equal to the product of the extremes.

Thus in the above proportion.

$$9 \times 9 = 81 \text{ and } 3 \times 27 = 81$$

From this it is readily seen that if any three terms of a proportion are given the fourth may be easily found by observing the following:

RULE: *Divide the product of the extremes by one of the means and the quotient will be the other mean.* Or,

Divide the product of the means by one of the extremes and the quotient will be the other extreme.

Simple Proportion is an equality of two simple ratios and consists of two couplets and four terms, any three of which being given, the fourth may be found as above.

EXAMPLE: A 60-in. pulley on a line shaft is belted to a 32-in. pulley on a countershaft. The pulley on the line shaft is to be changed to 45 in., but the speed must not be changed. What will be the diameter of the new pulley on the countershaft?

We realize immediately that to maintain the same speed the ratio between the pulleys after the change must be the same as before the change. Let x represent the size of the new driven pulley, then we get the proportion

$$60:32::45:x$$

Here an extreme is missing and we can find it by dividing the product of the means by the given extreme

$$\frac{32 \times 45}{60} = 34 \text{ in.}$$

Whenever possible, apply cancellation in working out problems in proportion.

Inverse Proportion is the comparison between a direct and an inverse proportion.

The safety valve affords a good illustration of this, as the pressure necessary to lift a lever safety valve is inversely proportional to the distance of its application from the fulcrum, for as the distance from the fulcrum is decreased the necessary pressure is *increased.* If the pressure necessary to lift one valve with the stem 3 in. from the fulcrum was 60 lb., the pressure necessary to raise a valve of the same kind and size but with the stem 2 in. from the fulcrum would be to 60, not in the ratio of the old distance to the new, $3:2 = \frac{3}{2}$, but in the inverse ratio or $2:3 = 1\frac{1}{2}$, i.e., the new pressure would be to the old pressure as the old distance is to the new distance.

If we call the new pressure x we may express this as:

$$3:2::x:60$$

Now multiplying the extremes and dividing by the given mean, we have

$$\frac{3 \times 60}{2} = 90 \text{ lb.}$$

In all calculations in proportion one must first make sure of what it is that must be found. This particularly applies in calculations involving inverse proportion, as sometimes it is not readily known whether the proportion is direct or inverse.

One can make sure as to whether the proportion is direct or inverse by being sure of what is to be found, and then state the proportion as direct. Then if the first term is smaller than the second, the third term should be smaller than the fourth, or if the first term is larger than the second the third should be larger than the fourth.

Compound Proportion is simply a statement which contains more than two couplets. In compound proportion as in simple proportion there is one third term which is of the same denomination or kind as the fourth or the required term, but there may be two or more first and second terms. After setting down the third term, take each pair of terms of the same kind separately and arrange them as first and second term, making the greater term the second term if this pair considered alone should require the answer to be greater. Then multiply all the first terms together and do the same with the second terms.

Multiply the product of all the second terms by the third term and divide the last product by the product of all the first terms.

EXAMPLE: If two men remove 3 cu.yd. of earth in one day, working 10 hr. per day, how many men working 8 hr. per day will remove 20 cu.yd. in 3 days?

Remember we wish to find the number of men.

Here we have 3 days: 1 day, 3 cu.yd.: 20 cu.yd. and 8 hr.: 10 hr. Multiplying all the first terms we have

$$3 \times 3 \times 8 = 72$$

And multiplying all the second terms we have

$$1 \times 20 \times 10 = 200$$

Now we have the proportion

$$72:200 = 2:x$$

The third term 2 men was given and the fourth term x or number of men required is

$$\frac{200 \times 2}{72} = 5\frac{5}{9}, \text{ say } 6 \text{ men.} \quad Ans.$$

THE UNIT METHOD

Problems in proportion can easily be worked out by what *is* called the *unit method*, an example of which is given below.

EXAMPLE: If a concrete block for an engine foundation is 15 ft. long, 5 ft. wide and 5 ft. thick, and weighs say 52,500 lb., what is the weight of a concrete block 18 ft. long, 6 ft. wide and 8 ft. thick.

Now, if a block 15 ft. long, 5 ft. wide and 5 ft. thick weighs 52,500 lb., a block 1 ft. long, 5 ft. wide and 5 ft. thick will weigh $\frac{52,500}{15}$ lb., and a block 1 ft. long, 1 ft. wide and 5 ft. thick will weigh $\frac{52,500}{15 \times 5}$ lb., and a block 1 ft. long, 1 ft. wide and 1 ft. thick weighs $\frac{52,500}{15 \times 5 \times 5}$ lb. This fraction is the *unit* we have been looking for.

Now, by a reverse process of reasoning we find that a block 18 ft. long, 6 ft. wide and 8 ft. thick will weigh, this *unit*, multiplied by each of the dimensions of the new block, or

$$\left(\frac{52,500}{15 \times 5 \times 5}\right) \times 18 \times 6 \times 8 = 120,960 \text{ lb.}$$

What we really did in this problem was to find the weight per unit of volume of the old block and then multiplied this unit by the volume of the new block to find the weight of the new block.

EXAMPLES FOR PRACTICE

(1) A boiler has a grate 4½ by 5 ft. and has 745 sq.ft. of heating surface. What is the ratio of grate to heating surface?

(2) The ratio of the width to the height of a chimney is 10. How wide is the base if the height is 175 ft.?

(3) If 14 tons of coal cost $60, what will 20 tons cost at the same rate per ton?

(4) On an engine having a 48-in. stroke, a lever is used the length of which from its point of suspension to the point of its attachment to the crosshead is 72 in. How far from the point of suspension must the pin for the indicator cord be placed to give a diagram 4 in. long?

(5) A pump discharging 20 gal. per min. fills a tank in 24 hr. How long will it take to fill the tank if the pump discharges 42 gal. per min.?

(6) A column of water 18 ft. high exerts a pressure of 41.4 lb. per sq.in. What pressure will be exerted by a column 72 ft. high?

(7) At what distance from the fulcrum must a weight of 32 lb. be placed to balance a weight of 12 lb. 16 in. from the fulcrum?

(8) If with a safety valve the valve of which is 4 in. from the fulcrum, 60 lb. pressure is required to raise the valve, what pressure would be necessary to raise the valve if it was 2 in. from the fulcrum?

(9) If 6000 gal. of water flow through a pipe in 1 day of 8 hr., how many gal-

lons will flow through 4 such pipes in 4½ days of 9¾ hr. each?

Work out the last problem by the unit method.

ANSWERS TO LAST WEEK'S PROBLEMS

(1) 12,609 B.t.u.
(2) 60,480 lb.
(3) 25 per cent. loss.
(4) 30.68 tons received.
(5) $882 actual price.
(6) 72 lb.
(7) Yes; $\frac{9}{10}$ of 1 per cent.

Refrigerating Engineers' Annual Convention

On Dec. 2 and 3, the American Society of Refrigerating Engineers held its eighth annual convention at its headquarters in the Engineering Societies Building, New York City. A number of excellent papers were presented, and the meeting was well attended.

Routine business and the reports of the election tellers occupied the morning session on Monday. Officers elected to fill the vacancies occurring this year were: President, Peter Neff, Canton, Ohio; vice-president, Theodore Kelischer, Philadelphia, Penn.; treasurer, Frank A. Home, New York City; directors, Carl Behm, New York City; Ezra Frick, Waynesboro, Penn.; Thomas Shipley, York, Penn.

President Shipley opened the afternoon session with an address on the "State of the Art." As a man "on the firing line," he spoke of the work of the manufacturing corps. The service was active and it required the concentrated effort of the artillery, infantry and officers to progress. The advance made in raw-water plants and evaporative systems was briefly mentioned and reference made to the Bureau of Standards appropriation for determining refrigerating constants.

He favored the standardization of ammonia fittings, and while urging the highest economy practical in refrigerating systems, he questioned the advisability of installing complicated machinery and apparatus to save a little fuel without giving due consideration to the life and initial cost of the plant. The coming congress of refrigeration shared his remarks and he appealed to the men in the field to show the visiting engineers America's best in refrigerating work.

"Liqgas; a Possible New State of Matter," by Gardner T. Voorhees, was the first paper read. The author has observed that CO_2 in passing from a liquid to a gaseous state had the appearance just above the critical point of a clear, colorless syrupy gas, resembling hot air with distinct lines of movement. He believed this to be a new state of matter probably common to all substances at or near the critical point. While in this condition the matter was neither liquid nor gas, so the author proposed the name "liqgas." In the discussion it was pointed out that there was nothing new in the phenomena observed and that to call it a new state of matter was unnecessary.

At the sixth annual meeting of the society, R. H. Tait read a paper on "A Gas Producer Operated Refrigerating Plant," mentioning a number of troubles which had developed. In the second paper E. W. Gallenkamp, Jr., related briefly the changes that were made to overcome these troubles and also submitted some interesting results obtained from tests of the plant.

In the discussion, producer-gas engines were not favored for the refrigerating plant, particularly those drawing gas from bituminous producers. Internal-combustion engines using oil or natural gas had operated continuously with satisfaction and might be used. If the compressor was designed for the speed of the engine direct connection would be feasible, otherwise belted units must be installed, or if a more flexible arrangement is desired a generator and motor might be placed to advantage between the engine and compressor.

An animated discussion on the use of ice in refrigerator cars in winter to prevent freezing of perishable products in transit and on the loss of refrigeration at doorways of refrigerated rooms closed the session.

In the evening, Louis Block read his paper on "Avoidable Accidents in Refrigerating Plants." He showed how an explosion might readily occur and how it could be avoided when testing the piping with high air pressure before charging the system with ammonia. Leaks in the engine room might occur with impunity unless the vapor and hydrogen from the ammonia come in contact with the open flame of an arc or gas light. Incandescent lamps would prevent such an accident. He told how breaking the compressor, breaking the crosshead of the steam engine or freezing the connecting-rod brasses to the crankpin may be prevented, and in conclusion gave a few "dont's" well worth heeding.

"Ammonia Compressor Safety Devices" was the title of the paper read by Peter Neff, the newly elected president. Protection against breakage of parts and damage from explosions, unusual pressures, pressure of liquid in the compressor, etc., were sought and obtained in numerous ways outlined by the author.

N. H. Hiller's paper on "The Manufacture of Distilled Water Can Ice" gives the particulars of elaborate tests of an exhaust-steam evaporator ice-making plant at Muncie, Ind., burning different kinds of coal. With one sample a result as high as 11.25 lb. of ice to 1 lb. of coal was obtained. The method of testing and the deductions made from the results are of interest. Following the paper discussion arose as to the ease of cleaning boiler and evaporator tubes with but little advantage on either side.

Other papers were "Atmospheric Dehumidifying," by J. I. Lyle, and "Corrosion in Refrigerating Systems," by Morgan B. Smith. In the latter paper conditions conducive to corrosion were outlined and results given of tests to show the behavior of numerous metals in brine.

The afternoon session was devoted to a discussion of the properties of anhydrous ammonia. Lionel S. Marks and F. W. Loomis presented a paper analyzing the results of the most important experimental work published so far, and discussing critically the tables prepared by various authorities. Wobsa's table on the properties of ammonia was considered the best data available.

A second paper by William E. Mosher, presented through the courtesy of the American Society of Mechanical Engineers, went still further. Not only were available data collected and correlated, but formulas to express the various properties were proposed. The constants in the various formulas were determined by aid of experimental data. No working tables were given, but such papers have been prepared for both the saturated and the superheated vapor. These with a Mollier diagram for ammonia will appear shortly in a bulletin of the engineering experiment station of the University of Illinois.

In a topical discussion on "Things New in Refrigerating Practice," Louis Block and Thomas Shipley told about their improved types of condenser. In the Block condenser the gas is introduced at the bottom and the liquid discharged from the top. When the valve admitting the gas is properly set, each lineal foot in this condenser does three to three and one-half times as much work as in the ordinary condenser. Mr. Shipley improved the efficiency of his condenser by installing a device similar to an injector through which ammonia vapor and liquid pass to the coils.

As a matter of final business it was decided to appoint a committee of three to confer with a similar committee of the American Association of Refrigeration on the advisability of standardizing flanged fittings. The selection was left to the president.

A banquet in the evening at Louis Martin's closed a very successful meeting.

Up to the present time, the total horsepower of marine turbines of the Parsons type, completed and under construction, and of licenses of the Parsons Foreign Patents Co., Ltd., is about 8,500,000 hp., an increase during the year of about 2,100,000 hp. Nearly 7,200,000 will propel warships and over 1,300,000 will be employed by yachts and the mercantile marine.—*Engineering.*

The American Society of Mechanical Engineers

The thirty-third annual meeting of the American Society of Mechanical Engineers, was held in New York during the week ending Dec. 7. The professional work was profuse and comprehensive and the social features, while somewhat of a departure, were in keeping with the traditions of the society and satisfying to and enjoyed by those in attendance.

The volume of the work of the meeting was greatly increased by the activities of the special committees, which, with hearings and discussions of partial and provisional reports, as well as in their finished contributions to the program, had the greatest effect that these embryo sections have yet exerted upon the work of the society.

The address of the retiring president, Alexander C. Humphreys, "The Present Opportunities and Consequent Responsibilities of the Engineer" dealt largely with the relations of the engineer to the public. On account of the abandonment of the usual reception at the Hotel Astor on Thursday evening, the reception following the delivery of the address on Tuesday evening took on increased importance.

Wednesday forenoon was devoted to a business session in which the reports of the council, and various committees were considered. Amendments to the constitution, introducing a new grade of membership, the "Associate," intermediate between the junior and the full member, transferring the function of deciding upon the admission of a candidate from the membership to the council and making the Committee on Constitution and By-Laws a standing committee were assigned for letter ballot.

The report of the tellers of election announced the selection of the following officers: President, W. F. M. Goss; vice-presidents, James Hartness, I. E. Moultrop, H. G. Stott; managers, W. B. Jackson, H. M. Leland, Alfred Noble; treasurer, William H. Wiley.

The new president, William Freeman Myrick Goss, was born at Barnstable, Mass., in 1859, studied at the Massachusetts Institute of Technology 1877-9, took the degree of master of science at Wabash in 1888 and doctor of engineering at the University of Illinois in 1904. He organized the department of practical mechanics at Purdue University in 1879, and was professor of experimental engineering, dean of the school of engineering and director of the experimental laboratory of that institution from 1890 to 1907, giving special attention to the testing of locomotives. Since that time he has been dean of the College of Engineering of the University of Illinois. He was a member of the Jury of Awards in the Chicago Exposition of 1893, expert in charge of the mechanical exhibit, Land Grant Colleges and chairman of the advisory committee of the Pennsylvania Ry. Co., charged with testing locomotives at the St. Louis Exposition of 1904. He is a fellow of the American Association for the Advancement of Science, member of the International Society for Testing Materials and of the Society for the Promotion of Engineering Education and ex-president of the Western Railway Club.

The report of the council showed a gratifying increase in the membership of the society.

Following the business session, a Power Plant Session had been arranged for the main auditorium, while the Gas Power Section held an interesting and well attended session which will be covered in our Gas Power Department in the issue for Dec. 24, and in still another hall a Textile Session under the auspices of the subcommittee on Textiles, was held, at which, among other papers, was one presented by John A. Stevens, of Lowell, on "The Power Plants of Textile Mills."

W. F. M. Goss

The following officers were elected by the Gas Power Section for the ensuing year: Chairman, F. R. Hutton, one year; Executive Committee, W. B. Owens, five years; W. A. Blauvelt, three years; H. J. Freyn and C. H. Benjamin, one year.

At the Power Plant session the first paper presented was "Dimensions of Boiler Chimneys for Crude Oil," by C. R. Weymouth. Chimneys suitable for coal fuel are larger than needed for oil fuel, and tables of dimensions of the former are of no use where oil is to be burned. A single table cannot be compiled, but the paper gives the elementary principles for determining the correct dimensions under any given set of conditions. The dimensions given are for sea level and at various altitudes with correcting factors for fuel economizers.

This paper was discussed by William Kent and E. H. Peabody, who referred to the importance of having every burner properly adjusted and the furnace of right design for oil burning.

"Tests of a 1000-hp., 24 Tubes High B. & W. Boiler," by B. N. Bump, was the next paper, and described tests on an inclined-header Babcock & Wilcox boiler with superheater and mechanical stoker. The peculiar feature of the boiler was the unusual height, the sections being 24 tubes high. The best combined efficiency was obtained at about 56 per cent. of rating and decreased slowly with an increase in rating.

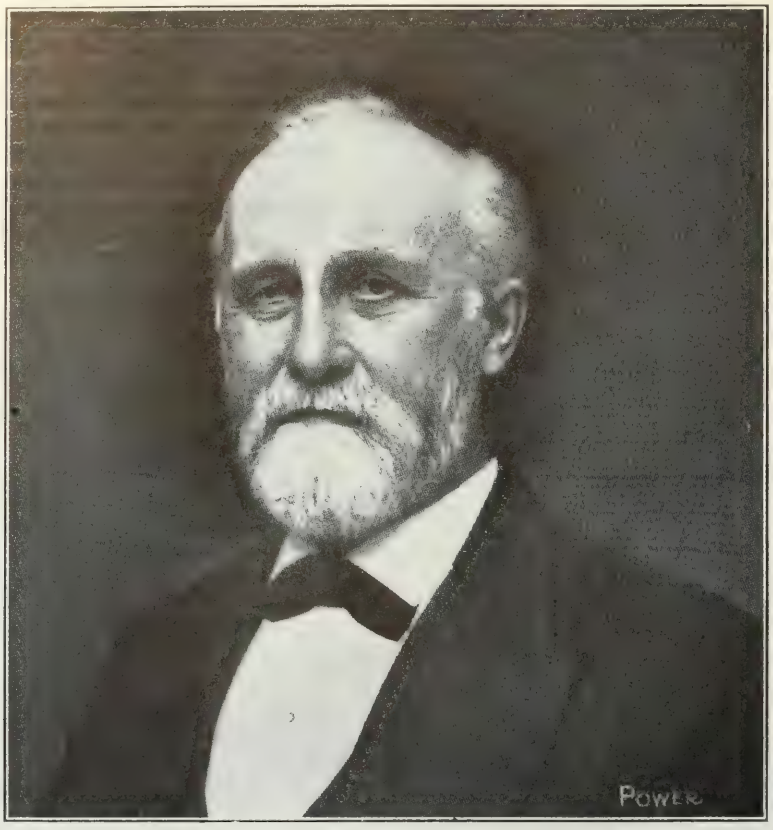
Prof. W. D. Ennis remarked the extremely low evaporation rate given—below 2 lb.—and the unusual ratio of heating to grate surface and believed that the large heating surface should have more completely absorbed the furnace heat. William Kent called attention to the diagram in the paper confirming what has been generally found, that there is a slight decrease in efficiency with increased rate of driving, and that the decrease will be less the more nearly the proper theoretical conditions are obtained for the ratio of air to combustible. He thought the boiler would have given better efficiency if the air supply had been exactly right, which would have been shown by an analysis for the oxygen in the gases. J. N. Browne commented on the system of baffles used, employing asbestos board and regretted that furnace temperatures were not given as they would have thrown more light on the results. D. S. Jacobus added to the discussion some remarks on the common mistake of not getting the true amount of CO. He was satisfied that the analyses given in the paper were correct. He spoke also of the difficulties in getting above 10, 11 or 12 per cent. of CO_2, and the indefiniteness of most temperatures as obtained in a furnace as they vary so much with where or how they are taken.

The session adjourned for luncheon recess to reconvene in the afternoon, when the discussion of Mr. Bump's paper was resumed. C. D. Young spoke of superheater experiments in locomotive practice, noting some confirmation of their conclusions in Mr. Bump's paper and asking what the experience was relative to the effectiveness of heat transfer to the water-heating surface as compared with that to the superheating surface. The speaker's results had indicated that the latter is only about half as effective as the former.

"Air in Surface Condensation," by George A. Orrok, the next paper, is an addition to the same author's paper on

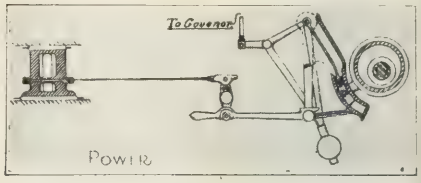
the "Transmission of Heat in Surface Condensers" previously presented before the society. It gives the results of his investigations on the amount of air present in feed water and the presence and effect of air in surface condensers and described an apparatus for measuring the air leakage. This paper was discussed November issue of the *Journal* and would require a complete session for its comprehensive discussion.

It was regretable that this, perhaps the most important feature of the program from the point of view of the power-plant engineer, was crowded to the end of a busy session taken up at a time when the limited number present were satiated with a day of attendance and under conditions, which tended to discourage rather than to bring out discussion. It is impossible to reproduce this report in POWER. It will be the basis of American practice in general testing work just as the former reports of the Committee on Boiler Testing has been for that particular division of the subject, and ought to have the exhaustive consideration of power-plant engineers before it is accepted as expressing the concrete views of American engineers. We shall treat it at some length in a later issue.



JOHN E. SWEET

On Wednesday evening a testimonial dinner was given to Prof. John E. Sweet, honorary member and past president of the society, in celebration of his eightieth birthday. Dr. A. C. Humphreys, Capt. Robert W. Hunt, Prof. Albert W. Smith and Dr. John A. Brashear were the speakers. The picture reproduced here-



VALVE-GEAR DIAGRAM

with of his original Straight-Line engine, as used in the mechanical laboratory of Sibley College, where he taught, adorned the menu. The designs, drawings, patterns and a good part of the machine work on this engine were made by Professor Sweet between Thanksgiving Day, 1871, and Apr. 1, 1872. This photograph was donated to the society by Mrs. Davis, widow of R. H. Davis, who did the rest of the machine work.

On Thursday forenoon the session in the main auditorium was devoted to the discussion of papers contributed by the subcommittee on Machine Shop Practice. Railroad and Cement sections held simultaneous meetings.

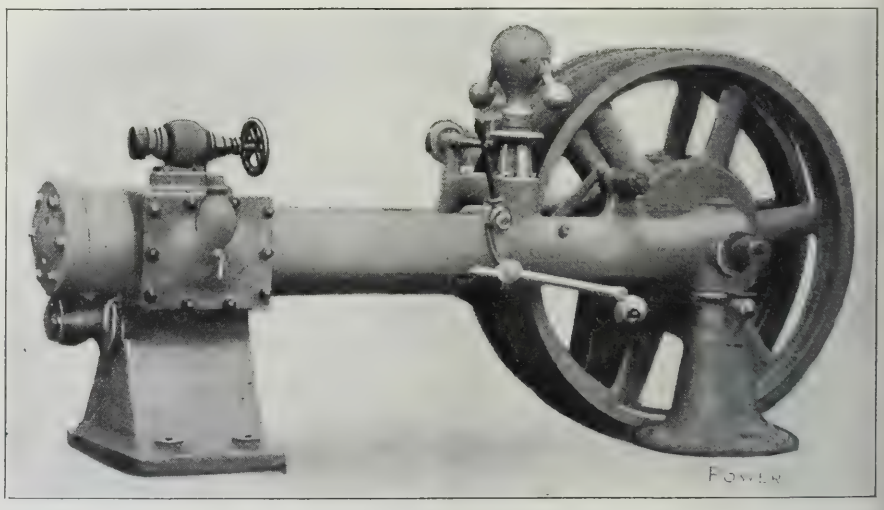
On Thursday afternoon the session in the main hall was devoted to papers dealing with the flow and measurement of fluids.

by George J. Foran and E. W. Christie.

The next two were papers also presented before the American Society of Refrigerating Engineers and are alluded to in a report of that society's meeting on another page of this issue. They were "Properties of Saturated and Superheated Ammonia," by William E. Mosher, and "Physical Properties of Anhydrous Ammonia," by L. S. Marks and F. W. Loomis.

This session was concluded with two papers that passed without discussion. "Experiments with North Dakota Lignite in a Steam Power Plant and a Gas Producer," by Calvin H. Crouch, and the "Baltimore Sewerage Pump Valve," by A. F. Nagle.

Wednesday afternoon had been assigned to the reception and discussion of reports of technical committees, including Power Tests, Hoisting and Conveying, Standard Cross Section Symbols, Involute Gears, Standardization of Catalogs, Flanges, Pipe Threads and others. The most extensive and important of these was a preliminary report from a subcommittee of the Committee on Tests. This report occupied 190 pages of the



FIRST STRAIGHT-LINE ENGINE

The ladies served tea on Wednesday and Thursday afternoons in the rooms of the society.

On Thursday evening the society joined with the civil, mining and electrical engineers in the ceremonies attending the award of the John Fritz medal to Robert Woolson Hunt, past president of the society, "for his contribution to the early development of the Bessemer process." Following the presentation, a joint reception to Mr. Hunt and to the members and ladies of the four societies was held.

Friday was devoted to Industrial Management under the auspices of the subcommittee on Administration. This subject, always provocative of discussion, occupied well attended sessions both morning and afternoon.

During the meeting a number of special committees were busy, two of which, those upon Safety Valves and Standard Boiler Specifications, promise to develop material of special interest to POWER readers. The papers with the discussion evolved which are of special interest in our field have been or will be presented in other issues.

As professional sessions were in progress throughout the meeting—morning and afternoon—the excursion program had to be made up without regard to them and in consequence most of the trips were ill attended.

Wednesday afternoon there were two excursions; one to the Grand Central Terminal, where the new construction there under way was inspected, and another to the Waterside station of the New York Edison Co.

Thursday morning the Ellis Island Immigrant Station was visited and the Aquarium. In the afternoon one party inspected the present state of work on the Fourth Ave. Subway in Brooklyn and another visited the Keller Mechanical Engraving Co., where opportunity was given to see the processes of making dies for silverware and the special machines for kindred engraving work. Still another party inspected the Bush Terminal in Brooklyn with its elaborate system for shipping and receiving freight and handling it to and from the various factories and warehouses at the terminal.

A trip to the Navy Yard, Friday morning, always interesting, was especially so this time, as the party was able to go all over the "Wyoming," the largest battleship in commission and then in drydock, and the "New York," recently launched and now being equipped, which will be the largest battleship. At the same time an excursion was made to the works of the August Griffoul Brothers Co., in Newark, N. J., which makes very fine castings of gold, silver, bronze, iron, etc., including commercial and art work, such as bas reliefs and statuary.

Friday afternoon, the Pennsylvania R.R. entertained a party at its terminal, showing the train- and mail-handling systems, power plant, and including a trip in a multiple-unit train through the tunnel to the Sunnyside yards on Long Island.

A number of additional excursions were also provided that individuals might take at their own election. These included power plants, large and small, industrial plants of various kinds, the museum of safety devices and a collection of ancient and modern firearms.

National Gas Engine Association Convention

The fifth annual convention of the association was held at the Claypool Hotel, Indianapolis, Ind., Dec. 3, 4 and 5, the meeting being the most important held since the organization of the society.

The feature of the first day was the opening address by the president, O. C. Parker, in which he laid before the members for discussion two principal questions: The matter of a uniform rating formula and the desirability of a salaried secretary for the association who would devote his whole time to promoting the interests of the gas-engine industry.

The opening paper of the second day was the "Present Status of the Gas Turbine," by Prof. C. D. Wagner, of Rose Polytechnic Institute. Prof. Wagner first reviewed the work of Armengaud, who built a 300-hp. turbine of the constant-pressure type in 1906. As this machine used six times the fuel required by the ordinary gas engine its commercial impossibility was at once recognized. He then gave in some detail the work of Hans Holzwarth who has recently realized 20 per cent. thermal efficiency in a machine of the explosive type developing 450 hp. (Described in POWER, Feb. 6, 1912.)

In a paper on "Uniform Systems of Cost Keeping," C. B. Segner called attion to the fact that it was not unusual for the price of one engine to be three times as large as that of another make of the same size. He attributed this great discrepancy largely to different methods used in estimating the cost of production, many concerns failing through marketing their product below the total cost. So strongly impressed was the association with the importance of this subject that it was determined to make a consideration of cost accounting a special feature of the next convention.

In a paper on "Practical Results from Gas Power," Charles Kratsch compared the cost of power as developed by electricity, steam, gasoline and illuminating gas and showed that there should be a large and profitable field for power production from illuminating gas in large cities.

In presenting a paper on "The Manufacturer and the College Man," Prof. George W. Munro, of Purdue University, called attention to the way the technical graduates are being absorbed by the large corporations and pointed out some of the problems presented to the smaller manufacturers in securing and utilizing the college man.

Following Professor Munro, Mr. McCulloch, of the National Implement & Vehicle Association, spoke on the benefits to be derived from permanent organization and hearty coöperation in a craft and instanced the success of concerted protest to the Interstate Commerce Commission against freight rate advances.

The program for the day was closed with an address by F. L. Walsh, of the Independent Petroleum Marketers' Association, on the desirability of establishing recognized standards in gas-engine oil, which he thought should be sold on specification rather than by trade name.

In the absence of A. E. Potter, his paper on "Gas Engine Economy" was read by R. A. Oglesby. The paper was largely devoted to an analysis of the results of carbureter tests published by Professor Munro in the March *Journal* of the American Society of Mechanical Engineers, the final conclusion being that engine efficiency depends primarily on correct mixture (a matter emphasized by E. H. Stickels in the afternoon) and to a lesser degree on a correct ignition system and the selection of lubricants.

"Application of Producer Gas to Brass Melting," a paper by B. M. Herr, described the application of producer gas to several heating processes and gave in detail the results of tests on producer gas and other fuels in brass foundry work.

One of the very interesting papers of the convention was on "Gas Engine Paints," by G. W. C. Sedgewick, who explained the difficulties of securing acceptable finishing material; a filler that would not crack or peel off under the repeated expansion and contraction, due to the high temperatures; a loader to properly take the enamel and finally an enamel which would withstand the action of heat, water and oil.

During the convention the association decided to employ a salaried secretary who should devote his entire time to the interests of the association and elected H. R. Brate, of Lakemont, N. Y., to this important position. To provide for the increased expense the annual dues were raised from $5 to $10 per year and it is expected that under the new arrangement the membership will soon reach five hundred. The officers elected for the ensuing year are, president, O. C. Parker, LaCrosse, Wis.; vice-president, R. A. Oglesby, Indianapolis, Ind.; treasurer, O. M. Knoblock, South Bend., Ind.; executive committee, O. C. Parker, B. M. Walker and R. A. Oglesby.

After selecting Springfield, O., for the next meeting, the association adjourned.

Cheap Electricity in London

Electric power at one-eighth of a penny per unit may have seemed to the audience of Prof. Fleming at his opening University College lecture a figure not likely soon to be attained in this country. Yet, for all that is boasted of the future of "water-power countries," it is difficult to suppose that the lowest records of price in Great Britain will be easily surpassed anywhere.

Recent negotiations on three sites in the north of England have related to contracts for 1500 kw. at 0.15d. ($0.003), 4000 kw. at 0.123d. ($0.00246), and 10,000 kw. at 0.11d. ($0.0022) per unit. The last named is for an equalizing load—that is to say, the supplies have the option of switching off when their plant is occupied in the "peak" hours. The other two examples represent electro-metallurgical works taking a continuous supply for 24 hours a day.

Waste heat plays, of course, an important part in the settlement of such rates, and it must not be assumed that the day is fast approaching for the universal distribution of power at such prices for ordinary purposes from ordinary sources.—*London Times.*

Tests of Wood, Paper and Steel Pulleys

Inadvertently the accompanying table was omitted from the article on the above subject, which appeared on page 848 of the last issue.

This table gives the data and results of the tests described in the article as made by E. D. Biggs and the author, H. A. Woodworth, at Purdue University, in 1909, to determine the breaking strength of pulleys of various kinds wtth a view to disclosing their principal points of weakness.

C. L. Huston Lectures on Boiler Steel

Charles L. Huston, vice-president of the Lukens Iron & Steel Co., gave an instructive talk on the manufacture of boiler steel before the American Institute of Boiler Inspectors at the latter's quarters, 29 West Thirty-ninth St., New York City, Friday evening, Nov. 29. The lecture was well illustrated by lantern slides.

After his talk on the processes of boiler steel manufacture, Mr. Huston showed several slides illustrating sections of ingots. The ingots were selected at random and planed down and analysis made so as to show the segregation of the carbon, sulphur, phosphorus and manganese.

It was interesting to note the distribution of holes at the edges of the ingots. These holes are caused by the action of the gases during the cooling of the ingot. Of the ingots illustrated none showed piping at the center. The speaker explained that when the ingot was rolled into plate form the honeycombed edges were cut off to insure a perfect plate.

Sometimes the holes are present in the center of the plate and are not thoroughly rolled out after the heated ingot has passed through the rolls. Should this condition be very pronounced, the speaker said, the plate would show pitting after coming from the rolls, i.e., the surface of the plate would contain pits similar to those caused by sulphurous or acid boiler waters. The speaker said that these pits caused in the manufacture of the plate might accelerate the pitting action of bad boiler waters.

Mr. Huston said he did not recommend boiler-pressure tests based on a high tensile strength, and further claimed that he regarded a boiler made of 54,000-lb. tensile strength material to be safer in practice than one of 60,000- to 64,000-lb. tensile strength. Good boiler plate he considers is that which shows a tensile strength of somewhere between 50,000 and 60,000 lb.

To get the plate to show a high tensile strength, it is necessary to make it hard, and striving after very high tensile strength usually results in a hardness of the plate that borders on brittleness, and such plate is obviously poor as boiler material.

Asked as to the advantage of firebox steel over flange steel for boiler shells as recommended by the Massachusetts Board of Boiler Rules, the speaker said that firebox steel was from selected stock whereas flange steel was from standard stock. Mr. Huston thought that the use of firebox steel for boiler shells was very good as firebox steel is not supposed to blister and should it blister, the manufacturer must make good by furnishing a new plate. It was conceded that a boiler made of firebox steel was well worth the extra cost of the better quality steel.

William Kent related some very interesting experiences of the early days of boiler-plate manufacture.

PERSONAL

H. E. Risher is in charge of the newly established Chicago offices in the First National Bank Building, of the Evens & Howard Firebrick Co., of St. Louis.

Elmer E. Carr has accepted a position with the Evens & Howard Fire Brick Co., St. Louis, Mo., which company will exploit the Carr patent ventilated furnace arch.

L. H. Mesker has resigned as manager of the St. Louis branch of Manning, Maxwell & Moore, Inc., to be connected with the Ferro Machine & Foundry Co., Cleveland, Ohio, after Jan. 1, in its sales department.

DATA AND RESULTS OF TESTS ON TWENTY-FOUR INCH PULLEYS

1	2	3					4		5	6		7	8	9
		Rim					Arms			Bursting speed		Centrifugal tension V^2		
No. of Test	Kind of Material in pulleys	Style	Dia. in inches	Bdth. inches	Depth inches	Area sq.in.	No.	Area sq.in.	Weight in pounds	Rev. per min.	Peripheral speed = V ft.per sec	const.	Character of fracture	Remarks
1	Wood. . .	Solid	24	6.25	1.62	10.15	2	14.4	29.37	2720	284.7	810.54	Completely broken	
2	Wood. . .	Solid	24	6.25	1.62	10.15	2	14.4	29.37	2550	266.9	712.35	Whole rim shattered	Arms remained.
3	Wood. . .	Two sections	24	6.5	1.78	11.6	2	8.25	29.67	2210	231.8	534.99	Whole rim shattered	Arms remained.
4	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	29.67	2110	220.8	487.53	Half of rim shattered	Arms and half of rim left on shaft.
5	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2390	251.0	630.01	Whole rim shattered	Arms remained on shaft.
6	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2430	254.3	646.68	Half of rim shattered	Arms and half of rim left on shaft.
7	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2360	247	610.09	Half of rim shattered	Arms and half of rim left on shaft.
8	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2420	253.3	641.61	Whole rim shattered	Arms remained, slightly shattered.
9	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2570	258.5	668.22	Whole rim shattered	Arms remained.
10	Wood.	Two sections	24	6.5	1.78	11.6	2	8.25	28.81	2535	244.4	597.31	Half of rim shattered	Arms and half of rim left on shaft.
11	Cast iron. .	Solid	24	6.0	0.406	2.44	6	0.99	70.44	3720	389.4	15,163.2	Not fractured	Not sufficient power.
12	Cast iron. .	Solid	24	6.0	0.406	2.44	6	0.99	70.44	3380	353.8	12,517.4	Not fractured	Not sufficient power.
13	Paper.	Solid	24	6.0	1.75	10.5	Solid web 1.5 thick		77.37	2820	295.2	871.43	Portion of rim thrown off	Rim fastened to web by wooden pins.
14	Paper.	Solid	24	6.0	1.75	10.5	Solid web 1.5 thick		77.37	2930	306.7	940.65	Portion of rim thrown off	Rim fastened to web by wooden pins.
15	Steel.	Two sections	24	6.75	0.0625	0.422	6	0.656	41.75	2240	234.5	5,499.0	Flanges and bolts bent	Flanges riveted on to rim.
16	Steel.	Two sections	24	6.75	0.0625	0.422	6	0.656	41.75	2240	234.5	5,499.0	Flanges torn off	Some rivets sheared.

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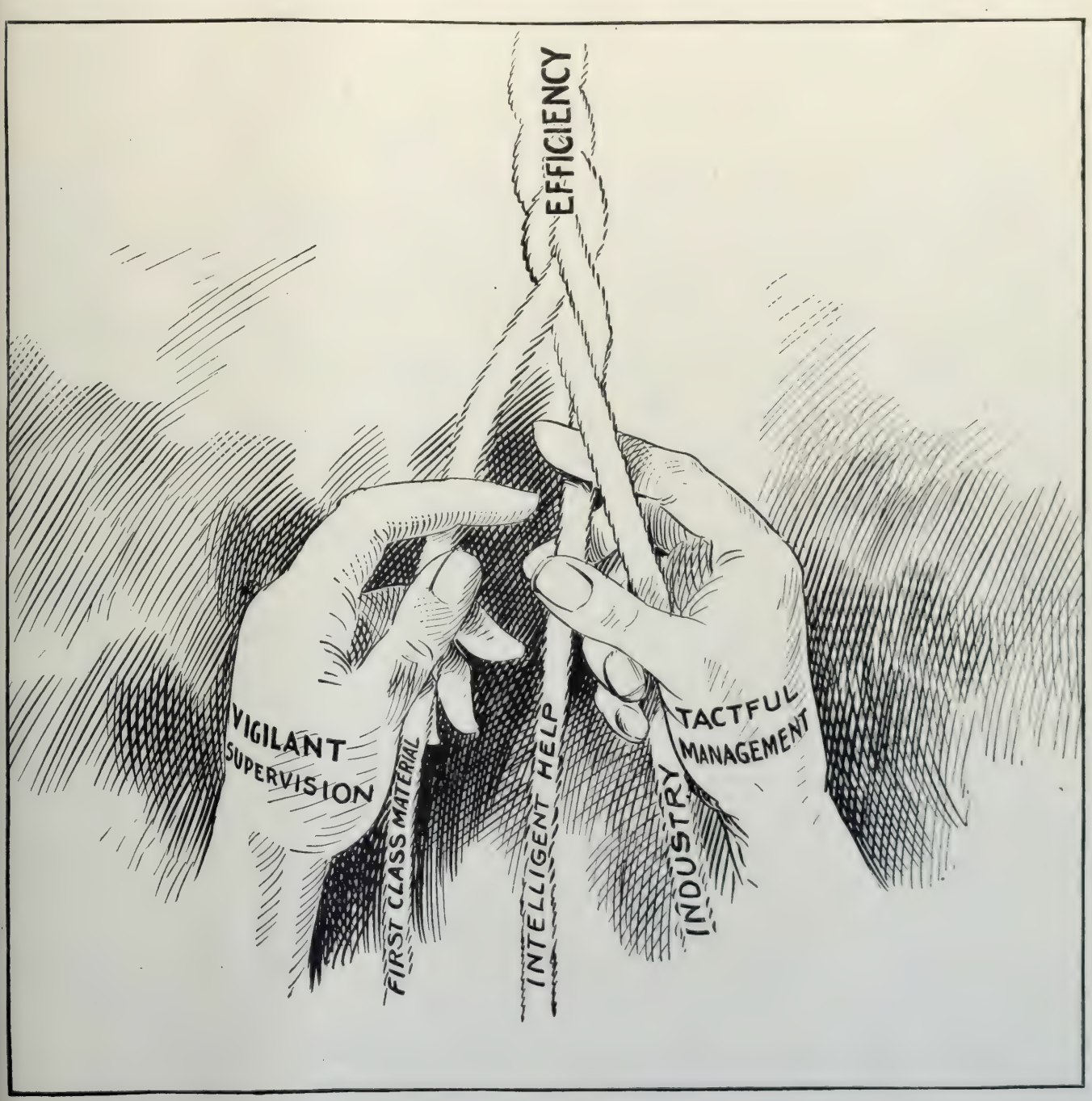


CAUSE AND EFFECT

Flat Surfaces Supported by Stay Bolts

By A. J. Toppin

A set of tables and curves showing the safe working pressures on flat stayed surfaces with different pitches and diameters of stay-bolts. They are computed for both the Massachusetts and the United States Government rules.

To save laborious calculations on the part of the boiler inspector and the boiler maker the writer has computed a set of tables giving the safe working pressure on the flat stayed surfaces of a boiler. By a flat surface is meant that which has a curvature above a certain radius; in Massachusetts, Ohio and the city of Detroit this radius is 19 in. and the United States Government specifies it as 21 in. A true flat surface is one which has an infinite radius.

The staying of flat surfaces resolves into two separate problems: the allowable pressure for a given thickness of plate and pitch, and the size of stay-bolt necessary to support the load. For illustration, consider the parallel case of a truss bridge spanning a river. No matter how strong the piers may be, if the members are not strong enough for the span and load, the structure will collapse, the piers remaining intact. On the other hand, although the structure may be strong enough for the given span and load, if the piers are not properly built, the truss will drop into the river.

Reverting to flat surfaces, the methods of failure are illustrated in Figs. 1, 2 and 3. A plate which is too thin would bulge as in Fig. 1; whereas a stay-bolt can fail in either of two ways, by breaking across its weakest section—the root of the threads (Fig. 2)—or by stripping its threads and riveted head (Fig. 3).

First Case

There seems to be a difference of opinion relative to the allowable pressure for a given thickness of plate and pitch of rivets. There are formulas by different authorities which give notably different results, as will be pointed out later when discussing the curves plotted. Massachusetts, Ohio and the city of Detroit employ the formula:

$$P = \frac{66\,(t+1)^2}{S^2 - 6}$$

in which,

P = Safe working pressure in pounds per square inch;
t = Thickness of plate in sixteenths of an inch;
66 = A constant determined by experiment;
S = Maximum pitch in inches.

The United States Government rule is:

$$P = \frac{k \times t^2}{S^2}$$

in which P, t and S are the same symbols as in the previous formula and k is a constant depending on the method of staying. For screwed stays riveted over and plates not exceeding $\frac{7}{16}$ in. thick, k = 112. For the same conditions and plates over $\frac{7}{16}$ in. thick, k = 120. There is also a formula by D. K. Clark:

$$P = \frac{k \times t}{l}$$

in which

P = Pressure in pounds per square inch, which will strain the plate to its elastic limit.
k = A constant, 5000 for iron and 5700 for steel;
t = Thickness of plate in inches;
l = Maximum distance in the clear.

By "distance in the clear" is meant from

Fig. 1. Failure by Bulging

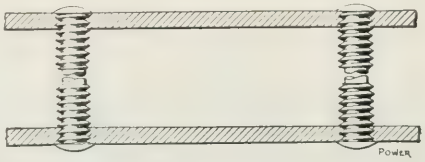


Fig. 2. Breaking of Stay-bolt at Root of Thread

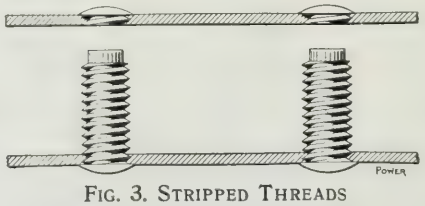


Fig. 3. Stripped Threads

root of thread to root of thread. If d is the diameter at the root of the thread, then l = Pitch — d (see Fig. 5).

The latter formula does not appear to be employed as universally as the other two but curves are plotted from it for comparison. It will be noted that this does not give the safe working pressure direct, but must be divided by a factor. As the elastic limit is generally about one-half the breaking load, the factor should be one-half the factor of safety based on the ultimate tensile strength. In other words, if a factor of safety of 6 is ordinarily employed the pressure found by the formula should be divided by 3 to get the safe working pressure. This factor was used in plotting the curves. In Clark's formula the factor of safety is chosen, while with the other two it is already contained within them and cannot be varied. Also in Clark's formula distinction to a small extent is made as regards the material, as he allows a constant of 5000 for iron and 5700 for steel plates.

Most authorities consider flat surfaces the same as beams fixed at each end and uniformly loaded, and claim that the formula for flat surfaces should follow the general formula for flexure of beams, constants being determined by experiment. Others claim that the formula should not follow these lines inasmuch as the fibers are all in tension and not half in tension and half in compression as in beams. In the writer's opinion the Massachusetts and United States Government formula and several others are based on the former reasoning and Clark's formula on the latter.

Second Case

The second problem is much simpler than the first: To determine the area of bolt necessary to sustain the load, let

P = Pressure in pounds per square inch;
$S_{max.}$ = Maximum pitch in inches;
$S_{min.}$ = Minimum pitch in inches;
A = Net area under pressure, in square inches;
D = Outside diameter of stay-bolt in inches;
d = Diameter at root of thread of stay-bolt.

The net area supported by one stay-bolt would be the product of the two pitches minus the area of stay-bolt at the root of the thread, which expressed in symbols would be

$$A = (S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)$$

The load sustained by the stay-bolt would be this area times the pressure or

$$P \times \left[(S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)\right]$$

Let T = the allowable tensile strength of the stay-bolt in pounds per square inch. Then the strength of the stay-bolt would be $\left(T \times \frac{\pi d^2}{4}\right)$. As this must balance the load

$$T \times \frac{\pi d^2}{4} = P\left[(S_{max.} \times S_{min.}) - \left(\frac{\pi d^2}{4}\right)\right]$$

Transposing,

$$P = \frac{\frac{T\pi d^2}{4}}{S_{max.} \times S_{min.} - \frac{\pi d^2}{4}} = \frac{T\pi d^2}{4\,S_{max.} \times S_{min.} - \pi d^2}$$

or

$$d = 2\sqrt{\frac{PS_{max.} \times S_{min.}}{\pi\,(T+P)}}$$

Stay-bolts 1¼ in. in diameter and under, generally have 12 threads per inch and the outside diameter

$$D = d + 0.1443$$

or

$$d = D - 0.1443$$

Substituting in the foregoing

$$D - 0.1443 = 2\sqrt{\frac{P S_{max.} \times S_{min.}}{\pi (T + P)}}$$

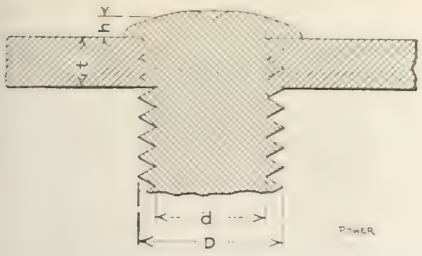


FIG. 4. REPRESENTING SHEARING RESISTANCE OF HEAD

or

$$D = 2\sqrt{\frac{P S_{max.} \times S_{min.}}{\pi (T + P)}} + 0.1443$$

The value of T varies according to different authorities. Massachusetts allows only 6500 lb. per square inch on mild steel or wrought-iron stay-bolts up to and including 1¼ in. diameter, whereas the United States Government allows 6000.

Failure by stripping the threads is remote and is therefore not calculated but a formula can be deduced for determining the height of the head of the stay-bolt. The threads would strip if the plate into which the bolt is screwed was too thin to afford the proper number of threads to have a hold, unless it is sufficiently headed over to make up the deficiency. The shearing surface furnished by the head would be $3.1416 \times d \times h$, as shown in Fig. 4. The area to resist shearing the

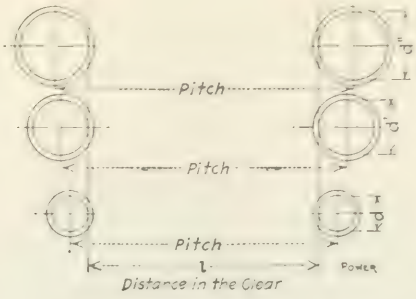


FIG. 5. DISTANCE IN THE CLEAR

threads would be $3.1416 \times d \times t$. The total area to resist stripping would be:

$$(3.1416 \times d \times h) + (3.1416 \times d \times t) = 3.1416\, d\, (h + t)$$

Multiplying this by the shearing value per square inch of the metal in the bolt would give the stress necessary to strip the threads. Calling this value *S.S.*, the stress would be $3.1416\, d\, (h + t)\, S.S.$ This must be at least equal to the stress necessary to break the bolt at the root of the thread, which would be $\left(\frac{3.1416\, d^2}{4}\right) \times T.S.$ in which *T.S.* is the ultimate tensile strength per square inch of the metal in the bolt. Then

$$\frac{3.1416\, d^2}{4} \times T.S. = 3.1416\, d\, (h + t)\, S.S.$$

or

$$\frac{d \times T.S.}{4} = (h + t)\, S.S. = h \times S.S. + t \times S.S.$$

and

$$h = \frac{d\, T.S.}{4\, S.S.} - t$$

The value of *S.S.* varies from 35,000 to 55,000 lb. per square inch for wrought iron and from 45,000 to 75,000 for steel. This formula shows that with constant, thickness of plate, the height of head should be increased as the diameter of the bolt is increased. Keeping the diameter constant the height of the head should be increased as the thickness of plate is decreased. If the shearing value of the metal of the plate was less than that of the bolt and there was no head to the bolt the threads of the plate would strip.

STAY-BOLTS WITH TELLTALE HOLES

The United States rules call for a ⅛-in. telltale hole drilled in the ends of each stay-bolt. This, of course, reduces its strength in direct proportion to the area of the bolt. The area of a ⅛-in. circle is 0.01227 sq.in., and the area of a ¾-in. bolt at the root of the thread is 0.288 sq.in. The percentage of reduction of area would be

$$\frac{0.01227}{0.288} \times 100 = 4.26 \text{ per cent.}$$

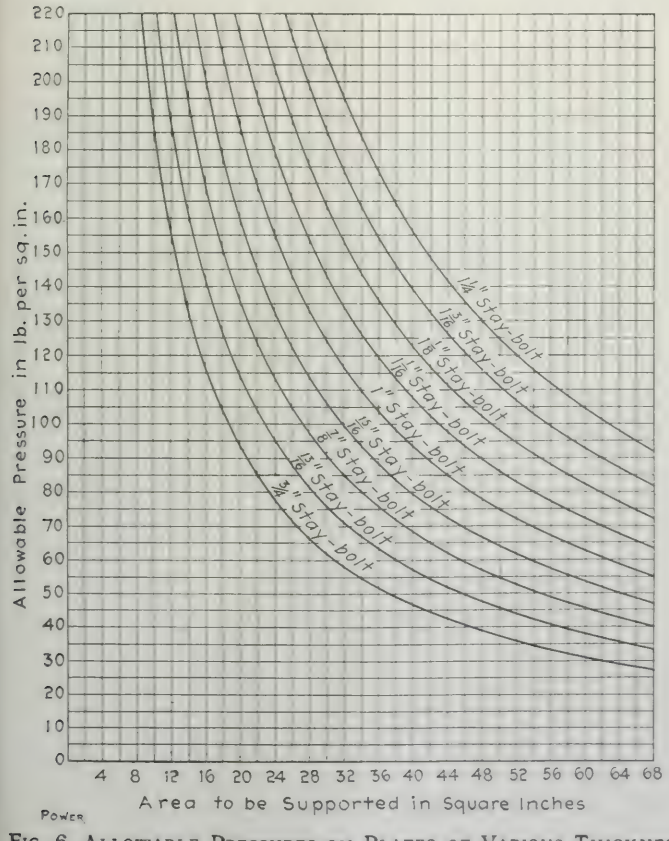


FIG. 6. ALLOWABLE PRESSURES ON PLATES OF VARIOUS THICKNESS AND PITCHES—FACTOR OF SAFETY OF 6 USED

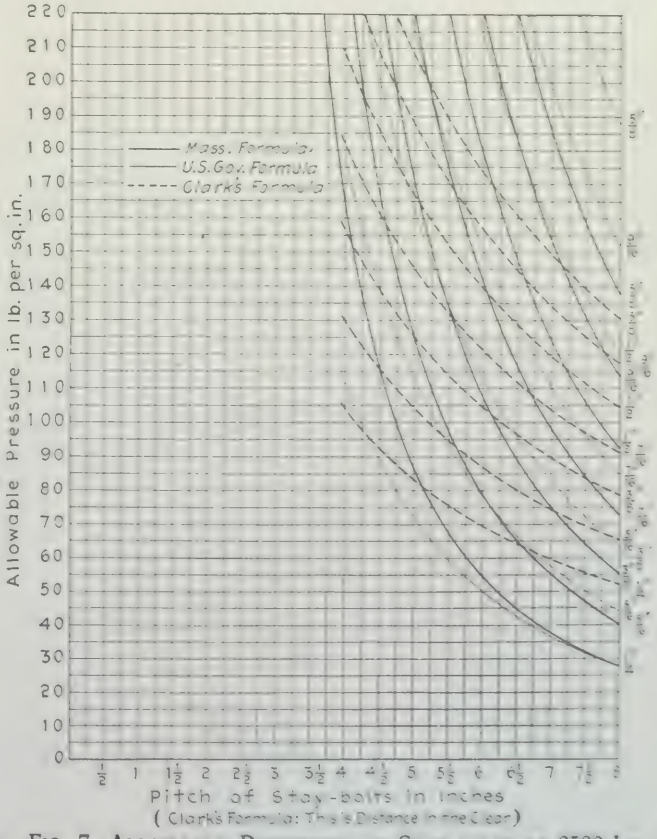

Allowable Pressure in lb. per sq. in.
Mass. Formula
U.S. Gov. Formula
Clark's Formula
Pitch of Stay-bolts in Inches
(Clark's Formula: Thick Distance in the Clear)

FIG. 7. ALLOWABLE PRESSURES ON STAY-BOLTS AT 6500 LB. STRESS

| Thickness of Plate in Inches | Allowable Pressures on Stay-bolted Flat Plate — Varying Pitch and Thickness — Maximum Pitch in Inches | | |

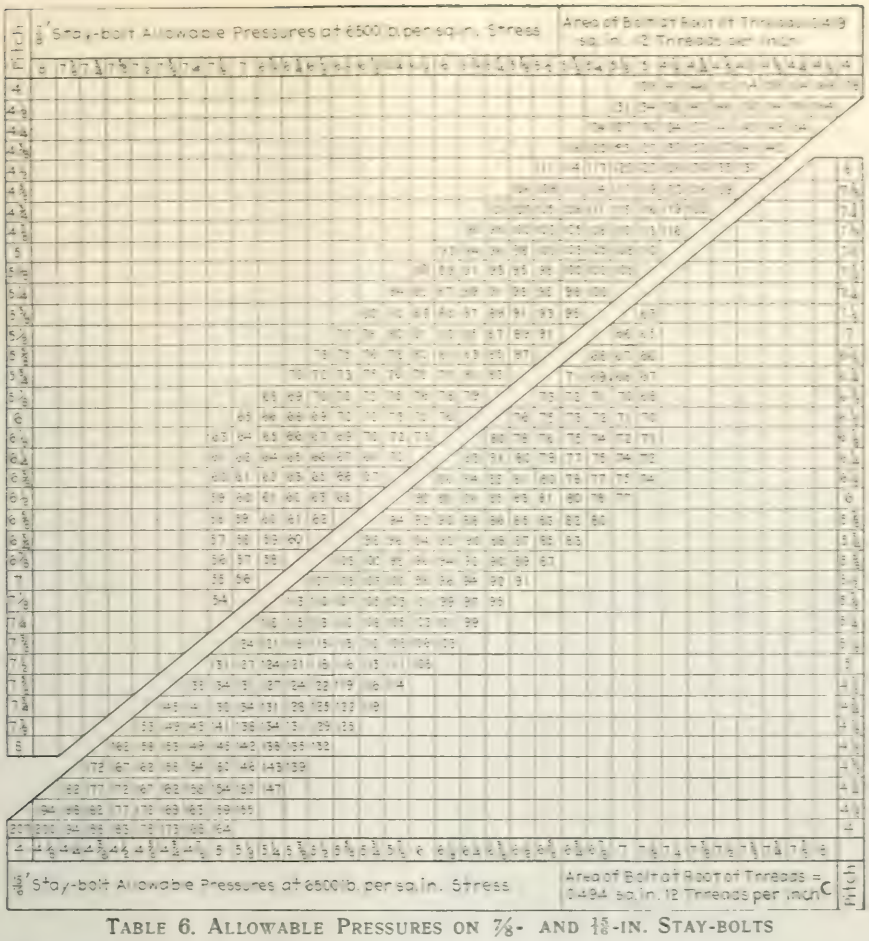

⅞" Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress
Area of Bolt at Root of Threads = 0.494 sq. in. 12 Threads per Inch
Pitch

TABLE 6. ALLOWABLE PRESSURES ON ⅞- AND $1\frac{1}{16}$-IN. STAY-BOLTS

Next consulting the stay-bolt tables it will be found that using equal horizontal and vertical pitches (5⅜x5⅜ in.) a $1\frac{1}{16}$-in. stay-bolt would be necessary. This stay-bolt is slightly stronger than necessary, being good for 113 lb. If, however, the minimum pitch is made less, say 5⅛ in., a ⅞-in. stay-bolt (Table 6) will answer the purpose, being good for just 100 lb. Or, with both pitches equal and below 5⅜ in., a ⅞-in. stay-bolt pitched 5¼x5¼ in. will be good for 100 lb. pressure.

Consider another problem in which the pressure is 180 lb. gage and the plate ½ in.; find the pitch and diameter of the stay-bolt.

The greatest pitch permissible for this thickness of plate and pressure (see Table 1) will be 5⅞ in. If the other pitch were made the same, a 1¼-in. stay-bolt would be good for 187 lb. This is more than necessary; hence using a pitch of 5⅜x5⅞ in., a $1\frac{1}{16}$-in. stay-bolt will answer, being good for 181 lb. If it is desired to use equal pitches, yet keep below 5⅞ in., a $1\frac{1}{16}$-in. stay-bolt pitched 5⅝x5⅝ in. will be good for 180 lb. Of course, this increases the value of the plate as the maximum pitch is 5⅝ in. and is good for 208 lb.

The tables are much simpler from the inspector's point of view. For example, let the pitches be 6x5⅝ in., the stay-bolt 1⅛ in. and the plate ⅜ in. thick. The maximum pitch is 6 in., and this is good for 107 lb. for a ⅜-in. plate (Table 1). A 1⅛-in. stay-bolt pitched 6x5⅝ in. at 6500 lb. stress will be found good for 148 lb. pressure. Therefore, the plate is the weaker and only 107 lb. should be allowed.

Again, let the plate be $\frac{7}{16}$ in. thick, the stay-bolt 1 in. and the pitch be 5x4¾ in., the boiler being under United States Government jurisdiction. The maximum pitch is 5 in. and is good for 215 lb. for a $\frac{7}{16}$-in. plate (Table 2). A 1-in. stay-bolt pitched 5x4¾ in. is good for 161 lb. at 6500 lb. (Table 7), but as the boiler is under government jurisdiction the bolts must be figured at 6000 lb. stress. The constant for this stress is 0.922. Therefore, the pressure would be

$$0.922 \times 161 = 148 \text{ lb.}$$

Hence, the stay-bolt is weaker than the plate and only 148 lb. should be allowed.

CURVES

Plotting the values given in the tables a set of curves was obtained as in Fig. 6. The curves for allowable pressure on plates according to the Massachusetts and United States Government formulas show at a glance the comparative values for a given pitch. Those based on Clark's formula do not show this as clearly as it must be remembered that what is termed pitch in the other formulas is "distance

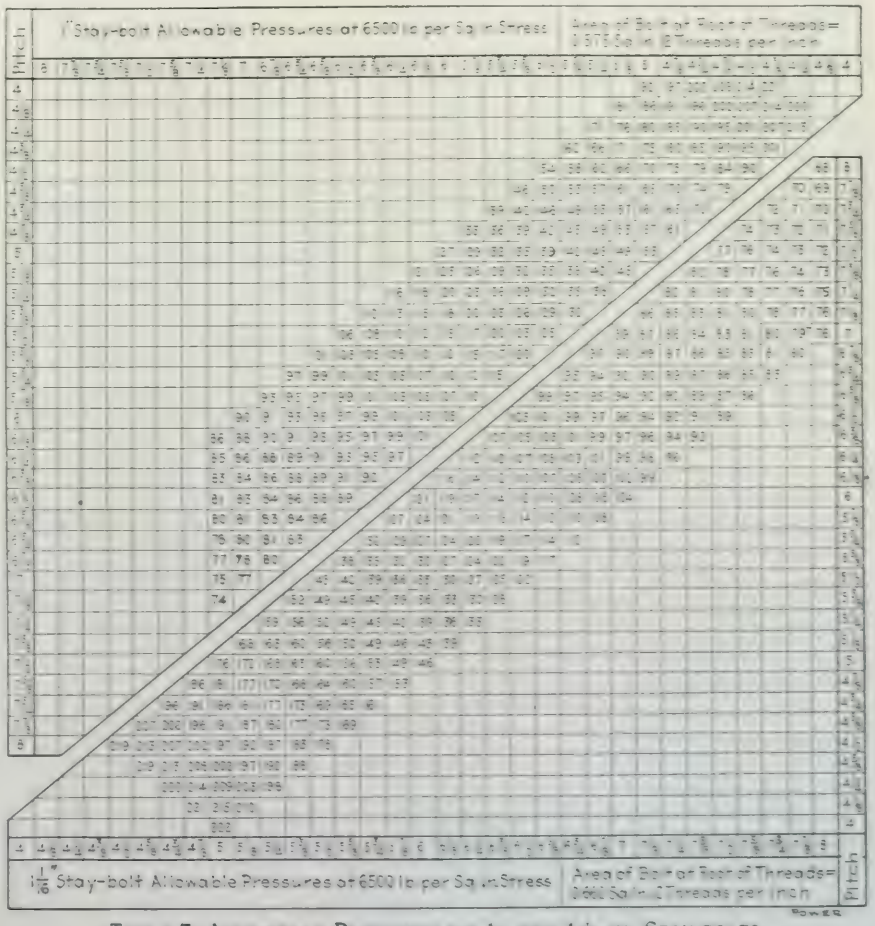

1" Stay-bolt Allowable Pressures at 6500 lb. per Sq. in. Stress
Area of Bolt at Root of Threads = 0.575 Sq. in. 8 Threads per Inch
Pitch

TABLE 7. ALLOWABLE PRESSURES ON 1- AND $1\frac{1}{16}$-IN. STAY-BOLTS

in the clear" in Clark's formula. In other words, to get the pressure from the curves first subtract from the pitch the diameter of the stay-bolt, at the root of the thread.

It will be noted that the pressures allowed for ⅝-in. plate and 4-in. pitch are very great, in fact, 798 lb. by the United States Government and 750 lb. by Massachusetts. It can be shown mathematically that with a pitch of 2.449 in. the pressure allowed by the Massachusetts formula on ⅝-in. plates is *infinite*.

It should be stated, however, that the Massachusetts rules do not allow using ¼-in. plates for stayed flat surfaces. This plate, however, has been used in the calculations for comparison.

Curves are also plotted in Fig. 7, showing the allowable pressure on different size stay-bolts, for any area to be supported, at a stress allowance of 6500 lb.

There is another way in which a stay-bolt can fail, namely, by bending. In fact, this is very common, being caused by the unequal expansion of the outer sheets and furnace sheets. However, this is another problem and one about which very little can be predetermined.

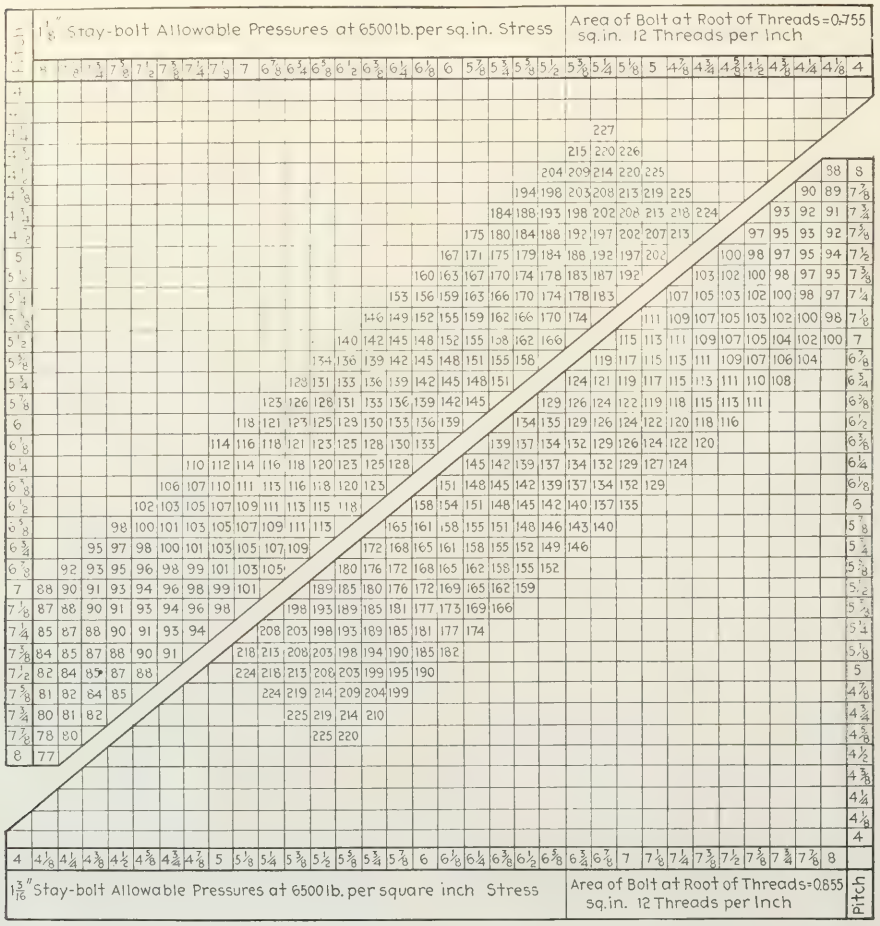

1⅛" Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress
Area of Bolt at Root of Threads=0.755 sq. in. 12 Threads per Inch
1 13/16" Stay-bolt Allowable Pressures at 6500 lb. per square inch Stress
Area of Bolt at Root of Threads=0.855 sq. in. 12 Threads per Inch
Pitch

TABLE 8. ALLOWABLE PRESSURES ON 1⅛-AND 1 13/16-IN. STAY-BOLTS

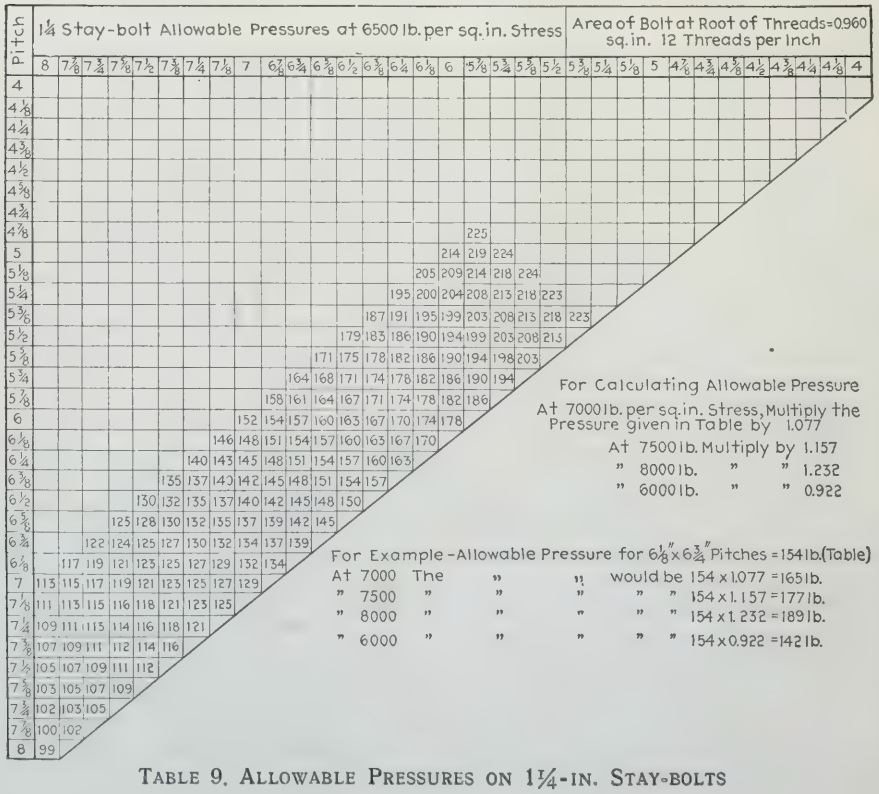

1¼ Stay-bolt Allowable Pressures at 6500 lb. per sq. in. Stress
Area of Bolt at Root of Threads=0.960 sq. in. 12 Threads per Inch
Pitch
For Calculating Allowable Pressure
At 7000 lb. per sq. in. Stress, Multiply the Pressure given in Table by 1.077
At 7500 lb. Multiply by 1.157
" 8000 lb. " " 1.232
" 6000 lb. " " 0.922
For Example—Allowable Pressure for 6⅛" x 6¾" Pitches = 154 lb. (Table)
At 7000 The " " would be 154 x 1.077 = 165 lb.
" 7500 " " " " " 154 x 1.157 = 177 lb.
" 8000 " " " " " 154 x 1.232 = 189 lb.
" 6000 " " " " " 154 x 0.922 = 142 lb.

TABLE 9. ALLOWABLE PRESSURES ON 1¼-IN. STAY-BOLTS

Motor Driven Pipe-Threading Machines

A compact motive-driven pipe-cutting and threading machine has been designed by the Curtis & Curtis Co., 85 Garden St., Bridgeport, Conn., which cuts off and threads 2½- to 12-in. pipe.

The die-cutting head is of the Forbes pattern, and is mounted on an inclosed cabinet. The machine is self-contained and can be moved from place to place; after connecting the lead wires it is ready to operate. The various speeds obtainable and the starting and stopping of the machine are controlled by a lever, but the motor runs continuously except when the machine is shut down for long periods.

Thumbscrews for adjusting the dies are done away with as the dies are now clamped in position by one movement of a lever.

The motor is inside the base, where it is protected from the drippings of oil used in the screw-cutting operations. The machine can be taken from its base and used as a hand machine.

Huge Hydroelectric Project

It is reported that financiers have asked permission of the Norwegian government to harness the Aura and Lilledal Rivers, east of Molde, in the Romsdalen district, Norway. The scheme involves the erection of a dam 140 ft. high and the transmission of water through a tunnel to Sundalen, where 200,000 hp. could be developed.

The estimated cost of the project is $10,000,000 and the scheme if carried out will result in one of the largest hydro-electric power plants in the world. —*Electrical World.*

Water-power reserves were created by the Secretary of the Interior during October in California, Idaho, Montana, Oregon and Washington, covering an area of 36,902 acres.

Some Notes on the Slide Valve

By L. Holder

Why lap and lead are used on the valve and the effect they have on the events of the stroke. Diagrams to show the relative positions of eccentric and valve and piston and crank.

The simplest form of valve with which engineers have to do is the slide valve. It has four edges, two steam and two exhaust, and may be designed with or without lap. A lapless valve is so designed that it will just cover the port openings when the valve is at mid-travel, as shown in Fig. 1.

A valve may have both outside and inside lap. Outside lap means that the lips project beyond the port openings when the valve is at mid-position, as shown in Fig. 2. When a valve has inside lap its arch is shorter than the width of the exhaust port and the bridges,

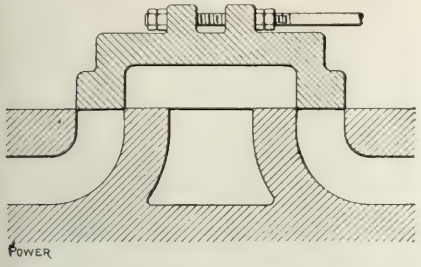


FIG. 1. LAPLESS VALVE IN CENTRAL POSITION

that is, the exhaust edges of the valve lap over the inside edges of the steam ports. This is shown in Fig. 3.

After examining these illustrations it will be evident that if there are two valve seats having equal steam ports and bridges (the thickness of metal between the steam and exhaust ports) and if one has a wider exhaust port than the other, the former will require a correspondingly wider arch, but there will be no difference in the travel, lap or point of cutoff. Supposing, however, that these same valve seats have equal steam and exhaust ports but a difference in the width of the bridge, the one having the wider bridge will require a proportionately longer arch. There will be no difference in the lap or point of cutoff, but to give free admission the wider bridge will require a longer travel. Again, if the two valve seats have the same size of exhaust ports and bridges but have different widths of steam port, the valve having the wider steam ports will require proportionately wider laps.

Most engineers know that a valve with lead opens a little before the engine is on dead center. With negative lead the valve does not uncover the port until the piston has started on its return stroke. In other words, the valve is set with late admission. Inside lap has nothing to do with steam admission nor does it affect the point of cutoff, but it does increase and hastens compression and retards the exhaust release, or, in other words, it tends to choke the escape of the exhaust steam. It allows expansion of the steam to take place during a longer period than if there were no inside lap. On the other hand, increasing the outside lap has no effect on the compression in the cylinder and it neither hastens nor retards the release of the exhaust steam, but the more outside lap a valve has the later the admission will take place.

Suppose the cutoff is retarded, due to increased travel of the valve, it may be neutralized by increasing the outside lap of the valve. Late compression can be corrected by adding more inside lap to the valve. If the exhaust takes place too late, due to excessive travel of the valve, the inside lap can be reduced or entirely removed.

When a valve has neither lap nor lead

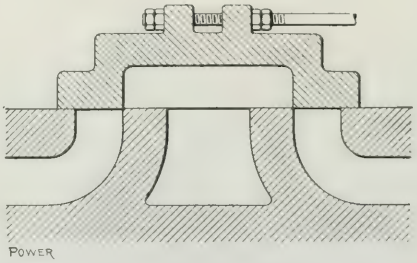


FIG. 2. VALVE WITH OUTSIDE LAP IN CENTRAL POSITION

the eccentric should be set at right angles to the crank and ahead of it. If there is lead and lap the eccentric should be set enough more than 90 deg. ahead of the crank to move the valve the length of the lap. Thus it will be seen that the position of the valve will depend upon the position of the eccentric. On some engines the eccentric is fixed, but on others its position on the shaft may be changed so as to vary the position of the valve during the stroke. This regulates the cutoff, so that just the proper amount of steam is admitted to carry the load. The governor changes the position of the eccentric and the eccentric governs the travel of the valve. Such an engine is of the variable-cutoff type. When equipped with a single valve the outside and inside lap is fixed, and the governor varies the cutoff by changing the valve travel or angular advance, and sometimes both.

To many engineers the effect of changing the movement of the valve is not understood. For instance, if the outside lap is increased the admission of steam begins later and ceases sooner, and the expansion of the steam occurs earlier and continues longer; the exhaust and compression remain the same. If the outside lap is decreased the admission of steam begins earlier and ceases later, while the expansion begins later and the period shortens, the exhaust and compression remaining the same.

If the inside lap is increased, the steam admission will not be changed, the expansion will begin as before and continue longer, the exhaust will begin later and cease earlier, while the compression will begin sooner and continue longer. If the inside lap is decreased the steam admission remains unchanged, the expansion will begin as before but the period of expansion will be shortened, the exhaust will begin earlier and cease later, and the compression will begin later so that the period will be shortened.

This is not all, for if the valve travel is increased the admission of steam will begin sooner and cease later, the expansion will begin later and cease sooner. The exhaust will begin earlier and cease later and the compression will begin later and end sooner. If the valve travel is decreased the admission of steam begins later and ceases earlier, expansion begins earlier and ceases later. The exhaust and compression is also changed, for the exhaust begins later and ceases earlier while the compression begins earlier and ceases later.

Moving the eccentric around on the engine shaft changes the angular advance of the valve but does not vary the travel of the valve. The events of the stroke all occur earlier but the periods remain unchanged. If the angular advance is decreased the events begin later but the periods are not changed.

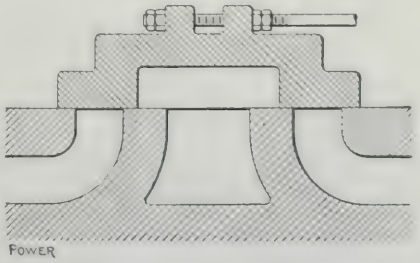


FIG. 3. VALVE WITH BOTH OUTSIDE AND INSIDE LAP

The lead varies with different engines and is given for the purpose of filling the clearance space of the cylinder with live steam before the piston begins the stroke. High-speed engines require more lead than slow-speed engines. The purpose of lap is to give the steam an oppor-

tunity to expand in the cylinder after cut-off occurs, and to keep the ports covered during the time expansion is taking place both inside and outside lap is given to the valve.

The position of the valve for corresponding positions of the eccentric can be determined by the use of a diagram such as Fig. 4. First draw a circle representing the path of the eccentric during one revolution of the engine, the

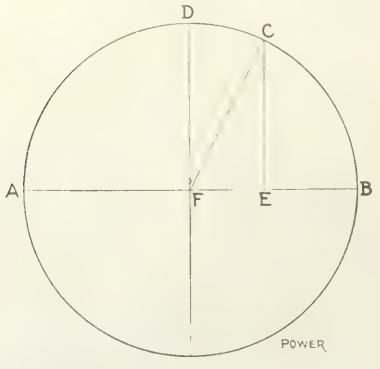


FIG. 4. RELATIVE POSITION OF ECCENTRIC AND VALVE

diameter *AB* indicating the valve travel. Next designate a point on the circumference of the circle which will represent the center of the eccentric, as at *C*. It is then an easy matter to find the distance the valve has moved from mid-position by drawing a perpendicular line from the point *C* to the line *AB*. The valve has moved the distance *FE* from mid-position.

The angularity of the eccentric is small and may be neglected, but is greater for the connecting-rod, so that the displacement of the piston relative to the crank must be found by the use of a diagram such as is shown in Fig. 5. In this instance the line *OH* represents the stroke, *J* the piston, *K* the crosshead, *KL* the connecting-rod and *LM* the crank. With the parts in the position indicated in the

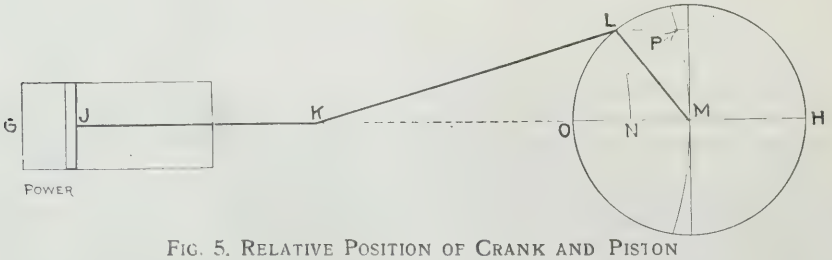


FIG. 5. RELATIVE POSITION OF CRANK AND PISTON

illustration, if the connecting-rod were of infinite length the displacement from the end of the cylinder would be equal to the distance from *O* to a line from *L* perpendicular to *OH*. To find the true displacement draw the arc of a circle *LN* through *L*, having its center at *K*. Then *NM* equals the displacement from the middle of the cylinder or *ON* equals the displacement from the end.

The same results could be obtained by drawing an arc having the same radius through *M* instead of through *L*, and by drawing the line *LP* parallel with *OH*, in which case *LP* being equal to *NM* indicates the displacement from the center of the cylinder.

Blonck Boiler Efficiency Meter

The Blonck boiler-efficiency meter illustrated herewith, consists essentially of a differential gage, showing the drop in draft through the boiler and a direct gage showing the draft in the furnace. The differential gage is a measure of the volume of gas passing through the boiler. The direct gage is a measure of the resistance of the fuel bed.

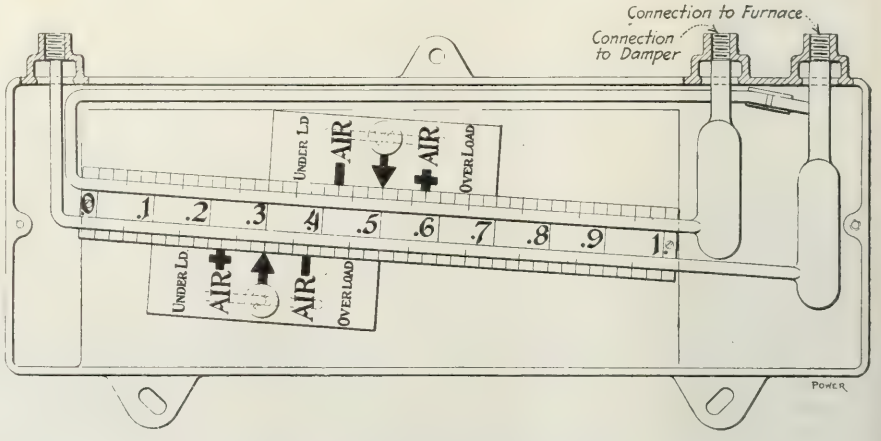
For a given boiler, a given grade of fuel and given rate of working there is a certain correct drop in draft between the furnace and the damper. Assuming a newly cleaned fire, under normal working conditions the resistance to the air passing through the fuel will be at the minimum and it will gradually increase as the thickness of the fuel bed increases, due to the accumulation of ashes on the grate, until just before the next cleaning, then it will be at the maximum. Each time fresh fuel is thrown on the fire this resistance to the passage of air increases slightly, but as this fuel ignites and burns the resistance gradually becomes less until the draft is once more the same or very nearly the same as it was before the firing.

The drop in draft through the boiler should remain constant as long as the rate of steaming remains the same and the damper should be adjusted to keep it so.

Thus, if the differential gage reads low, indicating an insufficient flow of air, the damper must be opened slightly to bring the amount up to normal. The furnace draft will also show an increase, indicating that the thickness of the fuel bed has increased.

If the differential draft is high, too much air is being admitted. The furnace draft may be low at this time, indicating that the fire is too thin. Or, the furnace draft may be normal, indicating that the damper in the stack should be throttled to reduce the effective draft in the boiler.

The important advantages of this instrument are its simplicity and its rapidity of action. The fireman can be quickly taught what the changes of draft mean and what should be done to bring conditions back to normal. Its rapidity of action makes it possible to instantly detect wasteful conditions and to correct them quickly.



BLONCK BOILER-EFFICIENCY METER

This meter is manufactured by W. A. Blonck & Co., Fisher Building, Chicago, Ill.

The resuscitation-from-shock chart recently circulated by the National Electric Light Association will be adopted by the Navy Department, and policemen and firemen in large cities are making good use of it. It has a wide use among lighting companies, one Middle Western company alone having ordered 1000 copies for distribution among its power-plant and other employees.

The effect of vanadium in plain carbon-steel castings is to increase the elastic limit about 30 per cent., giving a much higher proportion of available strength for the same ultimate strength. Moreover, it increases the power of castings of this character to withstand repeated and alternating stresses fully 50 per cent.

Rotary Jet Condenser

A new type of rotary jet condenser and high-vacuum, rotary air pump, has recently been placed on the market. It is a development of the Rees "Roturbo" type of centrifugal pump, described in POWER, Sept. 10, 1912, the feature of which was the employment of a revolving accumulator or pressure chamber. In the condenser, this pressure chamber is used to lift the condensing water to a limit of 20 or 25 ft., and give it an initial pressure after raising it.

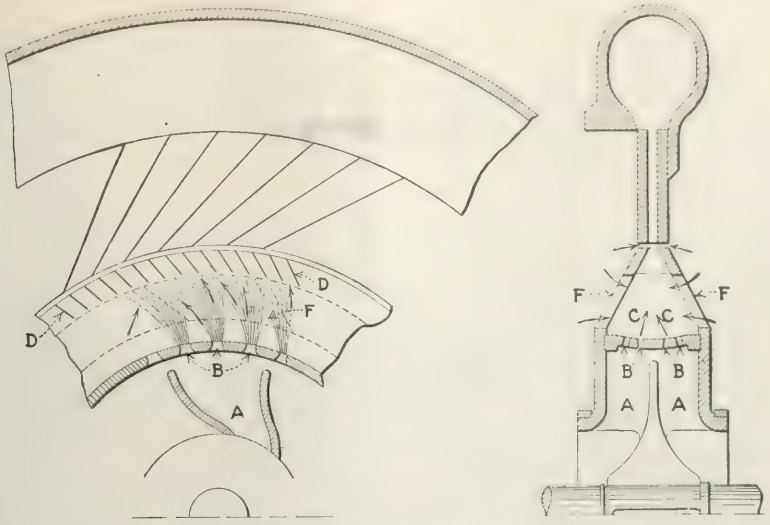


FIG. 1. PARTIAL SECTIONS OF ROTARY JET CONDENSER

Water flows up the suction pipe into this revolving pressure chamber *A*, Fig. 1, on the periphery of which nozzles *B* are arranged, not less than ⅜ or ½ in. in diameter, on the smaller, and up to ¾ or 1 in., on the larger sizes. These nozzles allow the water to escape from the pressure drum in radiating jets which are arranged to impinge in pairs, making the water jets fan-shaped and subdividing the water into a fine spray, that is projected in lines radiating from the shaft, but still rotating as a whole with the impeller, across a space *C*, into which steam exhausts, through the openings *F*. This, it is said, ensures a perfect mixture of steam and water which is ideal for the most rapid condensation.

Air entrained with the steam and the condensate are picked up by the blades of the exhausting fan *D*. Fundamentally these rim blades are the equivalent of the mixing cone, and the neck or "shock zone" of the old ejector, adapted to be rotated to give a positive exhausting effect through centrifugal action.

The neck of the stationary ejector is the seat of most of the losses due to shock, or the sudden change of velocity of the water from the high speed at which it is projected across the vacuum chamber to the much lower velocity of the mixture of air and water in the neck, where the initial compression of the air takes place, and where the volume of air is suddenly reduced. The mechanical ejection of the condensate in this condenser and the initial compression of the air by the fan blades removes the old difficulties of choking and gives high efficiency with certainty of ejection.

These exhausting fan blades are said to also form an automatic regulator, which deals with the fluctuating conditions of steam or vacuum without any speed regulation. For instance, if the vacuum is reduced below the normal, the rim blades are practically empty, and the condensate is swept out through the exhausting fan with but little assistance from the blades. If the vacuum rises beyond normal, thus exerting a greater backward pull, tending to cause choking, the rim blades fill up with water to a greater depth. Because of the extra centrifugal force created with a greater depth of the water carried around by the fan blades, a greater exhausting effect is secured when running at the same speed.

This regulating effect is absent on other devices of this kind, it is claimed, because the water has not heretofore been controlled by the rotary member at both the point of injection to, and ejection from the vacuum chamber, and at the same time spraying freely across a large intervening condensing space.

This feature, allowing for long, impinging jets, which form the finest spray and a great area for condensation, enables the apparatus to be used for the largest size condensers, without any auxiliary apparatus for extracting the air or water.

In Fig. 2 is shown an outside view of a motor-driven unit designed for condensing 30,000 lb. of steam per hour at 28 in. vacuum. The revolving impeller is fitted to the shaft as shown in Fig. 3. Fig. 4 shows the impeller with the side plate removed. A sectional view of the condenser is given in Figs. 5 and 6. When run as an air pump the same design of impeller is used and practically the only difference is that the external casing is of a different shape and has a smaller opening for the entrance of

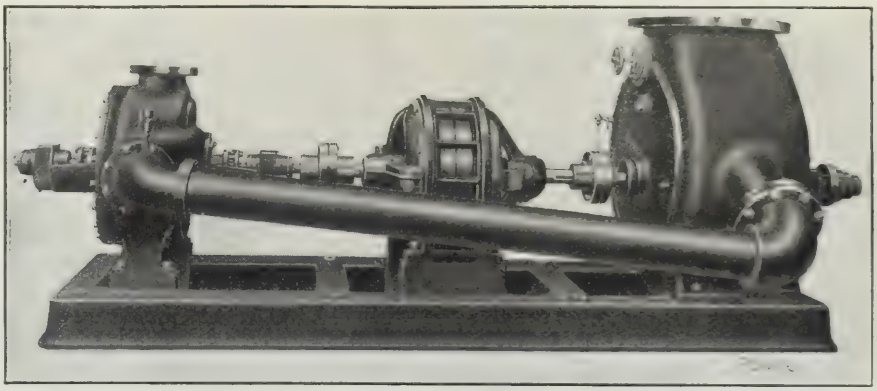


FIG. 2. MOTOR-DRIVEN ROTARY JET CONDENSER

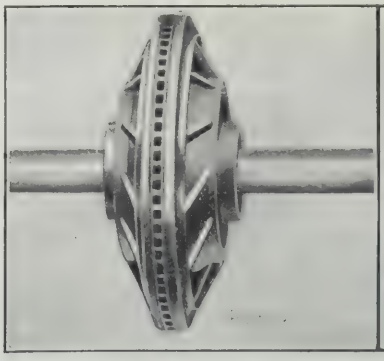


FIG. 3. IMPELLER OF ROTARY CONDENSER

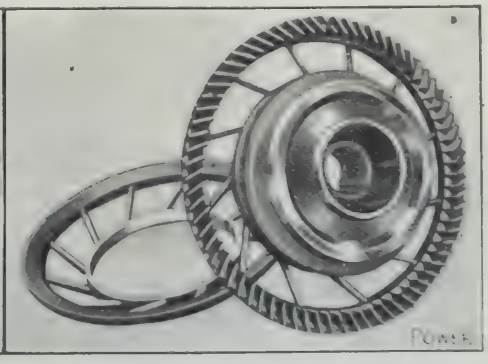


FIG. 4. IMPELLER WITH SIDE PLATES REMOVED

the air than would be required for steam.

Low first cost and maintenance, small floor space required and simplicity of construction are other advantages claimed.

Fig. 7 shows performance curves from manufacturer's tests of this machine, with various volumes of steam, etc. An allowance was made for 3 cu.ft. of free air per minute per 10,000 lb. of steam condensed per hour, for reciprocating engines and half of this for turbines.

Suppose it is desired to find the horsepower required by the condenser and the amount of cooling water at 60 deg. F. for a steam turbine using 30,000 lb. of steam per hour, and exhausting into a vacuum of 28 in.

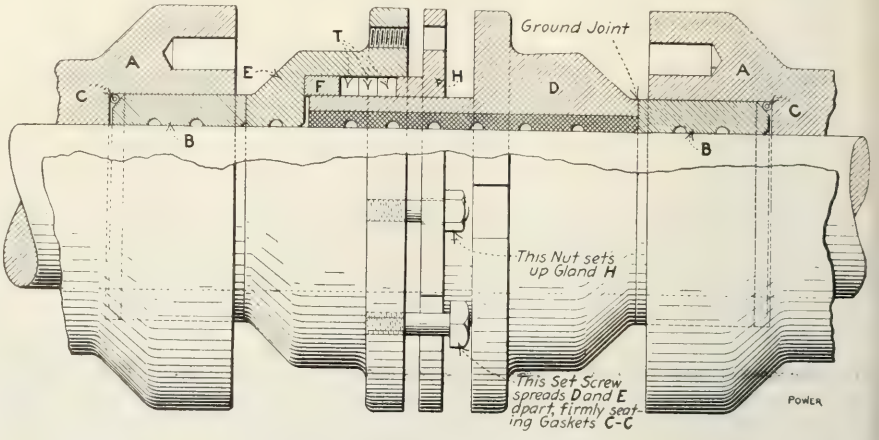
On the left side of the diagram run up the vertical line above 28 in. until the curve marked 60 deg. is reached; then follow this line horizontally to the right until it intersects the vertical line from 30,000 lb. of steam per hour. This point falls on or very near the curve for 40-hp. and 120,000 gal. of water per hour. The curves in Fig. 8 are for the air pump only, and are used somewhat similarly.

The Manistee Iron Works, Manistee, Mich., has the United States rights in this British invention.

Metal Pump Plunger Packing

This packing is a continuous connecting sleeve between two cylinders, and closely fits the pump plunger. It has water grooves to cause a reversal of flow of any fluid that may tend to pass the sleeve. The gland shown herewith is for a slip joint, and no wearing from the plungers occurs to the square hydraulic packing *T*, which is a permanent fixture, to be renewed only in case of accident. The packing is held in place by the gland *H*. *F* is a bottom ring. This slip joint fits between the parts *D* and *E*. The two parts may be forced apart after being placed between the cylinders to seat the gaskets *C* at the bottom of the stuffing-boxes *A*, which receive the bushings. On most outside-packed pumps, the sleeve may be made from 12 to 18 in. long. When wear occurs *D* is relined.

This packing is made by the Metallic Packing & Mfg. Co., Elyria, Ohio.



METAL PUMP-PLUNGER PACKING

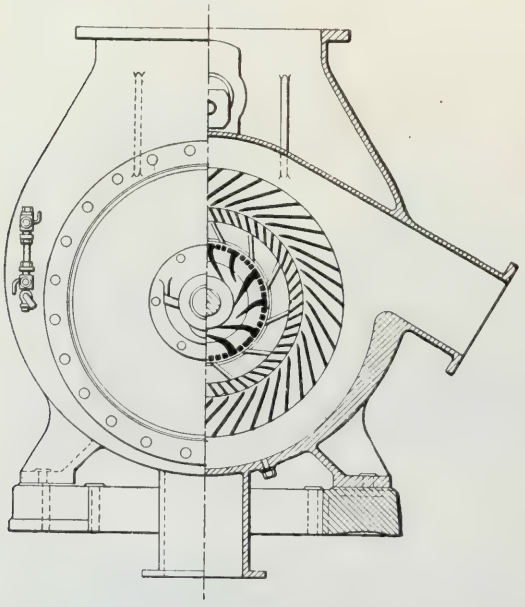


FIG. 5. SEMI-SECTIONAL END VIEW OF ROTARY JET CONDENSER

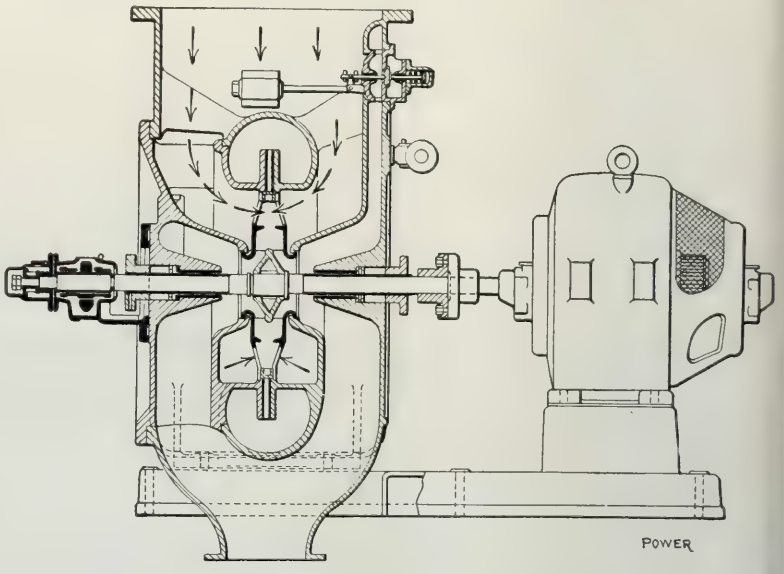


FIG. 6. SECTION THROUGH ROTARY JET CONDENSER

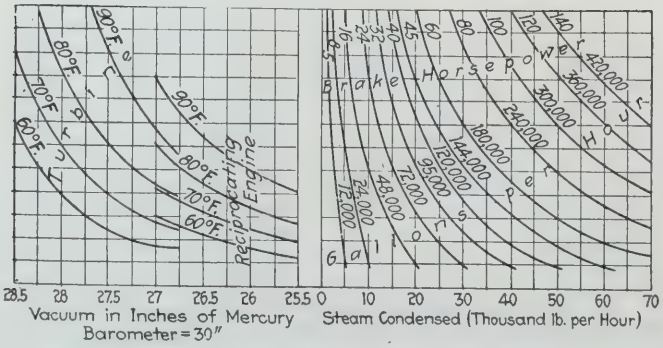


FIG. 7. CURVES SHOWING VARYING VOLUME OF STEAM AND TEMPERATURE OF COOLING WATER

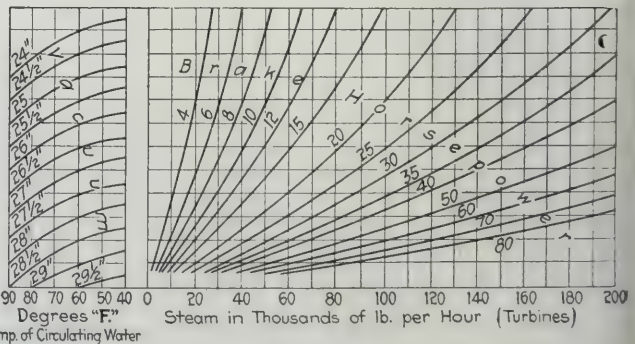


FIG. 8. PERFORMANCE CURVES OF ROTARY AIR PUMP

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Primer of Electricity

By Cecil P. Poole

The Series-wound Motor

This type of machine is used only for work requiring a heavy torque at starting and where large speed changes are not objectionable, as in the cases of railways, cranes, hoists and similar classes of service. Series-wound motors are supplied from constant-potential circuits almost exclusively and, as a rule, the entire armature current passes through the field windings. Any change in load, therefore, causes a change in speed much greater than that of a shunt-wound motor. When the current increases, it not only produces an increase in the voltage drop through the resistance of the windings but strengthens the field magnet and thereby increases the counter electromotive force of the armature for a given speed.

Because of the changes of field strength with changes in load, the speed of a series-wound motor is very unstable. Such a machine cannot be left in circuit without a load because the armature would race to such an excessive speed as to destroy itself by centrifugal force.

The series-wound motor possesses the great advantage, however, of enormous starting torque—hence its use for railway cars and hoists. The reason for this is that torque is directly proportional to both the field-magnet flux and the armature current, and, as already explained, the armature current is also the field current; therefore, a heavy armature current produces a heavy field-magnet flux. The field-magnet flux does not vary in direct proportion to variations in the motor current, because the field-magnet cores and armature teeth are magnetized considerably above the bend of the "saturation" curve.* Doubling the current, therefore, does not double the magnetic flux.

Fig. 116, which is the excitation curve of a series-wound motor, illustrates the effect of current changes upon the field strength of the machine. With 10 amp. in the field winding the total flux in the air gaps between the pole faces and the armature core was 2,000,000 maxwells; with an increase to 20 amp. the flux increased to only 3,800,000 maxwells, and four times this exciting current did not quite double the flux.

The test data from this machine are presented in Table 7, and a careful study of these figures should be made in order to get a grasp of the characteristic behavior of series-wound motors. The numbers in the extreme right-hand column

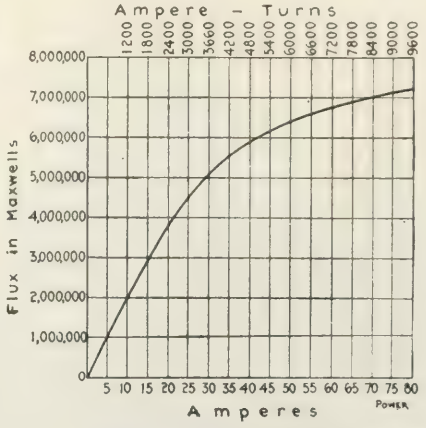


Fig. 116. Excitation Curve of a Series-wound Motor

are total horsepower developed by the armature; that is, the external power plus the power absorbed in revolving the armature and supplying core losses. This corresponds precisely with the indicated horsepower of a steam engine, and the power lost in the windings and connections corresponds very closely with the loss due to condensation in an engine.

In fact, a series-wound motor operating on a constant-potential circuit has almost all of the general characteristics of a steam engine supplied at constant pressure and running without a governor.

Referring again to Table 7, it will be found that the drop due to the electrical resistance of the machine is given as being 2 volts greater than the product of *Resistance* × *Current*. The explanation of this is that the drop at the brush faces was almost constant at 2 volts for all loads, the lowest drop being 1.9 volts and the highest 2.15 volts. (See lesson of July 26, 1910, on the resistance of brush contacts.) The table shows very clearly how rapidly the torque increases with the current and how the horsepower increases almost in proportion to the current, in spite of the great reduction in speed.

A chart made by plotting these figures in the form of curves, however, gives a clearer grasp of their significance. Fig. 117 is such a chart. The "R.p.m." curve, for instance, is much more effective than the tabulated figures in conveying to the mind the rapid decrease of speed during the first part of the range. It also illustrates the statement regarding the tendency of a series-wound motor to race at light loads; if the left-hand end of the curve were prolonged it would show a speed of about 3700 r.p.m. at the 5-amp. line of the current scale.

A series-wound motor is usually provided with a starting rheostat constructed on the same general principle as that of the shunt-wound machine. As the field winding is in series with the armature, the rheostat is connected in series with the whole machine, as indicated by Fig. 118. In some cases, the connections are arranged so that when the contact arm is at the end of its travel, giving maximum speed, the resistor that has been cut out of the main circuit is connected in parallel with the field winding. This "shunts" some of the current out of the winding and weakens the field, thereby compelling the armature to run at a higher speed in order to generate the counter electromotive force required to balance the impressed electromotive

TABLE 7. TEST DATA FROM A 20-HP. SERIES-WOUND MOTOR

250 volts. 800 armature conductors. 2 armature paths. 120 turns in each field-magnet coil. Total resistance of armature and field-magnet windings, 0.4 ohm.

Amperes	Ampere Turns	Total Flux	Total Drop	Counter E.m.f.	R.p.m.	Torque	Total Hp.
10	1200	2,000,000	6	244	1830	9.38	3.27
15	1800	2,950,000	8	242	1260	20.8	4.87
20	2400	3,800,000	10	240	947	35.7	6.43
25	3000	4,540,000	12	238	786	53.3	8.0
30	3600	5,100,000	14	236	694	71.9	9.5
35	4200	5,550,000	16	234	632	91.2	11.0
40	4800	5,900,000	18	232	590	110.8	12.4
45	5400	6,170,000	20	230	559	130.3	13.9
50	6000	6,400,000	22	228	535	150.2	15.3
55	6600	6,575,000	24	226	516	169.7	16.7
60	7200	6,740,000	26	224	498	189.8	18.0
65	7800	6,880,000	28	222	484	209.9	19.3
70	8400	7,000,000	30	220	471	230.0	20.6
75	9000	7,100,000	32	218	460	250.3	21.9
80	9600	7,200,000	34	216	450	270.3	23.2

*See lesson on Dynamo Field Excitation, Nov. 8, 1910.

force. Fig. 118 illustrates the electrical principle of such a rheostat but not the usual mechanical construction. When the arm first reaches the last contact, the resistor *Rh* is cut out of the circuit; further movement of the arm to the right will bring the auxiliary finger *F* into contact with the first button, thereby connecting the otherwise free end of the resistor to the far end of the field winding and putting the two in parallel. Besides taking some of the current out of the field winding, this reduces the electrical resistance of the circuit and therefore the drop due to resistance. The required counter electromotive force is thereby increased, entailing still more increase in speed.

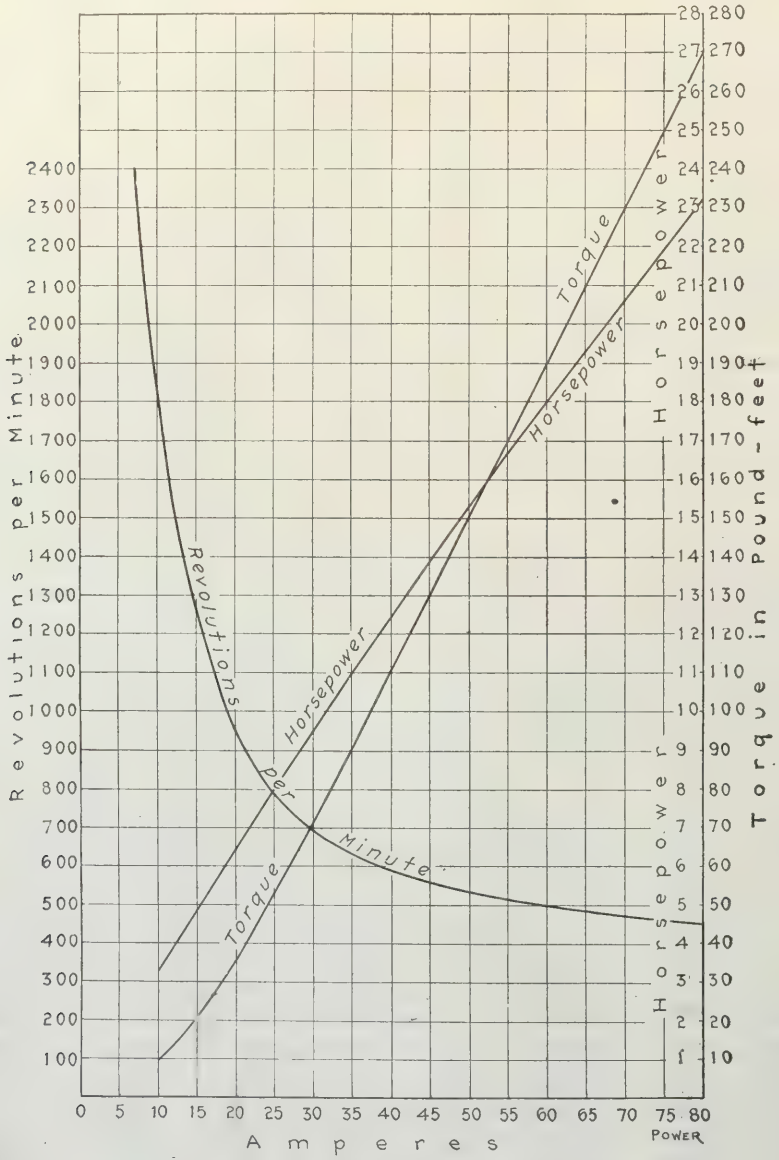


FIG. 117. CHART OF SERIES MOTOR PERFORMANCE

THE COMPOUND-WOUND MOTOR

To avoid the excessive speed changes of the series-wound motor and at the same time obtain the advantage of a powerful starting torque, motors are built with "cumulative" compound windings. (See Fig. 111.) The series winding gives the heavy torque at starting and the shunt winding provides a steady field flux which keeps the armature from running away when the load is not heavy enough to hold the speed down with a simple series winding.

For example, suppose the motor last discussed had been equipped with a shunt winding of 7000 ampere-turns per pole, which produced a total flux of 6,690,000 maxwells, and a series winding of 20 turns per pole. For convenient comparison also assume the total resistance of the armature and series field windings to be 0.4 ohm, as before. With the full load of 20 hp. the armature current of 70 amp. would produce

$$70 \times 20 = 1400 \text{ ampere-turns}$$

in the series winding and these added to the 7000 shunt ampere-turns would make the total excitation 8400 ampere-turns. This would produce the normal total flux of 7,000,000 maxwells (see Table 7) and the speed would be 471 r.p.m., as in the case of the series-wound machine. But if the load were removed entirely, leaving the series winding practically dead, the shunt winding would maintain 6,690,000 maxwells. The drop in the armature circuit would be at least 3 volts, leaving 247 volts to be opposed by counter electromotive force; the speed, therefore, would not rise above

$$\frac{6{,}000{,}000{,}000 \times 2 \times 247}{6{,}690{,}000 \times 800} = 554 \text{ r.p.m.}$$

With the series-wound machine previously described, a decrease of only 6 hp. in the load would produce the same increase in speed.

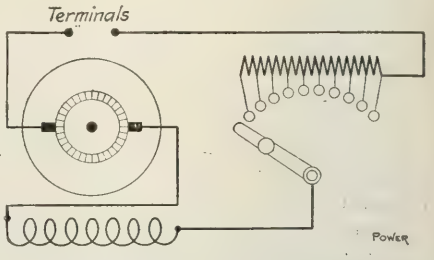


FIG. 118. SERIES-WOUND MOTOR AND STARTING RHEOSTAT

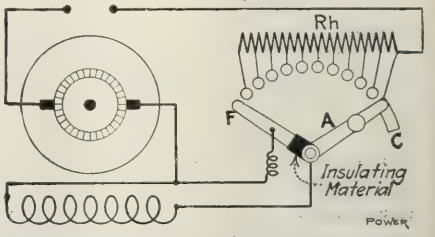


FIG. 119. SPECIAL RHEOSTAT SHUNTING FIELD WINDING FOR MAXIMUM SPEED

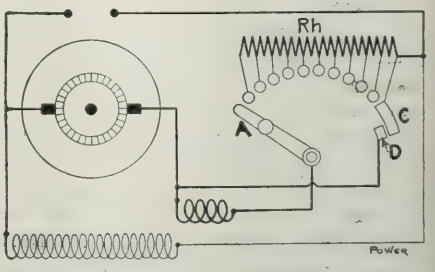


FIG. 120. COMPOUND-WOUND MOTOR AND STARTING RHEOSTAT

As the preceding example shows, the speed regulation of a compound-wound motor is poor. When a fairly close approach to steady speed is desired, therefore, together with powerful starting torque, a compound-wound motor is used and with it a starting rheostat arranged to cut out the series winding after the resistor has been cut out. A diagram of such an arrangement is shown in Fig. 120. The final contact *C* of the rheostat is a curved plate instead of a button, and an auxiliary contact *D* is located near the end of it so that the arm *A* will touch both plates at the end of its travel. As the arm is connected to one terminal of the series field winding and the contact *D*

to the other, the winding is short-circuited when the arm rests on the contact.

Compound-wound motors are not commonly built with differential field windings now. This type was used to some extent years ago when the speed regulation of the shunt-wound motor was not good. The effect of the differential winding is to weaken the field strength as the load increases, thereby causing the speed to be higher than if the series winding were not there.

For example, take the machine just considered, but assume that it is equipped with a shunt field winding containing 8400 ampere-turns per pole, producing a flux of 7,000,000 maxwells, and suppose the series winding to contain 40 turns per pole and to be connected to oppose the shunt winding. Assume the resistance of the armature circuit to be 0.4 ohm, as before. At no load the effect of the series winding would be too small to consider, and (with an armature-circuit drop of 3 volts) the speed would be

$$\frac{6,000,000,000 \times 2 \times 247}{7,000,000 \times 800} = 529\ r.p.m.$$

With a load sufficient to cause 70 amp. to pass through the armature circuit, the series winding would have an exciting force of 2800 ampere-turns, and, this being opposed to the shunt winding of 8400 ampere-turns, there would be 5600 ampere-turns per pole left active. According to the excitation curve, Fig. 116, 5600 ampere-turns will produce a total flux of 6,240,000 maxwells. The speed, therefore, with this load would be

$$\frac{6,000,000,000 \times 2 \times 220}{6,240,000 \times 800} = 529\ r.p.m.$$

the same as at no load.

The speed of a differentially-wound motor, however, is not the same at all loads, because the excitation curve is not a straight line; that is, the change in flux is not proportional to the change in ampere-turns. If the winding be designed to give the same speed at no load and at full load, the speed at all loads between will be lower. If the winding gives the same speed at half load as at no load, the speed below half load will be lower and that at all loads above one-half will be higher than the no-load speed. A complete discussion of the differentially-wound motor is not worth while, because this type has gone almost out of use.

Conduit Sizes

Recognizing the tendency in the past to use conduits of too small diameter, the National Electrical Contractors' Association has prepared data as to the proper sizes of conduit to be used in installing wires and cables. As relates to power and lighting systems this is as given in the accompanying tables.

The conduit sizes are based on the use of not more than three 90-deg. elbows in runs taking up to and including No. 10 wires; and two elbows for wires larger than No. 10; wires No. 8 and larger are stranded.

THREE-WIRE CONVERTIBLE SYSTEM

Size of wires, B. & S. Gage		Size conduit, in.	
2-wire	1-wire	Internal	External
14	10	½	1.05
12	8	¾	1.05
10	6	1	1.31
8	4	1	1.31
6	2	1¼	1.66
5	1	1¼	1.66
4	0	1½	1.9
3	00	1½	1.9
2	000	1½	1.9
1	0,000	2	2.37
0	250,000	2	2.37
00	350,000	2½	2.87
000	400,000	2½	2.87
0,000	550,000	3	3.5
250,000	600,000	3	3.5
300,000	800,000	3	3.5
400,000	1,000,000	3½	4
500,000	1,250,000	4	4
600,000	1,500,000	4	4.5
700,000	1,750,000	4½	5
800,000	2,000,000	4½	5

NUMBER OF WIRES IN SYSTEM

Size, B. & S. Gage	Capacity Amp.	One wire in a conduit Size of conduit, in.		Two wires in a conduit Size of conduit, in.		Three wires in a conduit Size of conduit, in.		Four wires in a conduit Size of conduit, in.	
		Internal	External	Internal	External	Internal	External	Internal	External
14	12	½	0.84	½	0.84	½	0.84	½	1.05
12	17	½	0.84	¾	1.05	¾	1.05	¾	1.05
10	24	½	0.84	¾	1.05	¾	1.05	1	1.31
8	33	½	0.84	1	1.31	1	1.31	1	1.31
6	46	½	0.84	1	1.31	1¼	1.66	1¼	1.66
5	54	¾	1.05	1¼	1.66	1¼	1.66	1¼	1.66
4	65	¾	1.05	1¼	1.66	1¼	1.66	1½	1.9
3	76	¾	1.05	1¼	1.66	1¼	1.66	1½	1.9
2	90	¾	1.05	1¼	1.66	1½	1.9	1½	1.9
1	107	¾	1.05	1½	1.9	1½	1.9	2	2.37
0	127	1	1.31	1½	1.9	2	2.37	2	2.37
00	150	1	1.31	2	2.37	2	2.37	2½	2.87
000	177	1	1.31	2	2.37	2	2.37	2½	2.87
0,000	210	1¼	1.66	2	2.37	2½	2.87	2½	2.87
CM.									
250,000	235	1¼	1.66	2½	2.87	2½	2.87	3	3.5
300,000	270	1¼	1.66	2½	2.87	2½	2.87	3	3.5
400,000	330	1¼	1.66	3	3.5	3	3.5	3½	4
500,000	390	1½	1.9	3	3.5	3	3.5	3½	4
600,000	450	1½	1.9	3	3.5	3½	4		
700,000	500	2	2.37	3½	4	3½	4		
800,000	550	2	2.37	3½	4	4	4.5		
900,000	600	2	2.37	3½	4	4	4.5		
1,000,000	650	2	2.37	4	4.5	4	4.5		
1,250,000	750	2½	2.87	4½	4.5	4½	5		
1,500,000	850	2½	2.87	4½	5	5	5.56		
1,750,000	950	3	3.5	5	5.56	5	5.56		
2,000,000	1050	3	3.5	5	5.56	6	6.62		
Duplex Wire.									
14	12	½	0.84	¾	1.05	1	1.31	1	1.31
12	17	½	0.84	¾	1.05	1	1.31	1¼	1.66
10	24	¾	1.05	1	1.31	1¼	1.66	1¼	1.66

This information has also been prepared in the form of a set of charts showing the conduit and conductors in full size. These have been copyrighted by the National Electrical Contractors' Association, Utica, N. Y., and are furnished by them at cost.

CORRESPONDENCE

Truing Commutators

If a commutator is kept properly trued by the use of sandpaper on a curved block or any other good smoothing device, it is hard to see why it should ever be necessary to take it to the turning lathe. However, if the grinding apparatus is not properly made and used it will grind more on one side than the other or grind in spots. In either case the trouble seems always to become worse with each grinding rather than better.

I always use two pieces of sand stone or grind rocks if I can get them. They should be chiseled out to fit, and each should reach nearly half-way around the commutator if possible. I use two pieces in order to be certain that the pressure is exactly the same on both sides and for the same reason always grind while the armature is revolving at a fair speed.

An attempt to smooth a commutator by drawing the sandpaper back and forth on it while it is not turning is quite certain to throw it out of round and the use of a short block or stone will usually grind low spots; but a stone reaching one-third or more around the commutator will touch only the high places and automatically keep it ground to a true circle.

Where the commutator is quite large I make the half circle with several stones clamped between two boards. The boards should be heavy and the bolts drawn up very tightly to prevent accident. The same stones may be used on two or more sides to fit the different circles and they may be rearranged in the clamp.

FRED BOONE.

Davidson, Okla.

A company has been organized in Rio de Janeiro, Brazil, for the purpose of utilizing the Paulo Alfonso Falls of the San Francisco River to generate electric power. At first 200,000 hp. will be generated, which quantity is to be increased at a later date to 300,000 hp.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Gas and Oil Power Development in Europe

One of the most instructive and comprehensive treatises on current European practice in gas and oil power development is contained in the report of H. I. K. Freyn, chairman of the Gas Power Section of the American Society of Mechanical Engineers. Mr. Freyn recently returned from a trip of five or six months abroad devoted almost exclusively to a study of gas and oil power conditions, and the results of his findings were given at the recent annual meeting of the society. His report is in part as follows:

The Diesel Engine

Without desiring to detract from the importance of the Diesel engine my careful investigation has shown that the warning against over-enthusiasm which I sounded about a year ago on a similar occasion was fully warranted. In this connection I refer more particularly to the large Diesel engine, above 1000 hp. per unit, about which very optimistic reports and accounts appeared from time to time. The actual achievements do not altogether coincide with these glowing accounts nor with the claims of some manufacturers.

However, it is unnecessary to dwell on the importance of the Diesel engine for ship propulsion. Efforts are being made by many manufacturers of marine Diesel engines in Europe, to design an engine which, while giving nearly the same excellent fuel economy as the stationary engine, will also meet the severe requirements of simplicity, reliability and reduced first cost. That the last condition is not yet fulfilled is shown by the fact that according to authentic information the cost of an 800-hp., four-cylinder, two-stroke-cycle marine engine recently built was found to be 50 per cent. higher than anticipated and 150 per cent. higher than that of corresponding steam-engine equipment.

I was unable to find anywhere the 10,000 and 15,000-hp. Diesel-engine ships mentioned in current literature and even Diesel-engine enthusiasts do not believe that our modern leviathans will ever be propelled by Diesel engines.

The largest single Diesel-engine unit propelling any ship afloat develops not over 2000 hp. and there still remain many obstacles to be overcome before the single-acting multicylinder, four-stroke- or two-stroke-cycle Diesel engine on board ship is supplanted by the double-acting engine. Vertical engines of the latter type are still in the experimental stage, and their future does not appear very promising on account of the serious mechanical complication of scavenging pumps and the multiplicity of scavenging valves in each cylinder head.

The lack of simplicity of these designs has prompted inventors and manufacturers to experiment with the so called "valveless" two-stroke-cycle type which presumably will become the large marine Diesel engine of the future. One valveless two-stroke-cycle experimental engine of 260 hp. is claimed to have shown an oil consumption of 0.375 lb. per i.hp.-hr., which at a mechanical efficiency of 60 per cent. would give 0.462 lb. per b.hp.-hr.; an economy as good as that of a regular two-stroke-cycle engine with scavenging valves. The mean effective pressure obtained reached 125 lb. per sq.in., which is remarkable in view of the extremely simple scavenging method employed.

Among the valveless, two-stroke-cycle engines, the Junkers oil engine has attracted much attention. The excellence of the form of combustion space is a great advantage in this engine, and gives it the distinction of being the best oil engine, at least from a thermodynamic point of view. Practical experience alone, however, will tell whether its unusual design and certain mechanical disadvantages will be offset by this advantageous feature. Only a few smaller engines, mostly for marine propulsion and of the twin- or three-cylinder vertical types have been built abroad. The tandem arrangement of cylinders which results in the same number of crank efforts realized in the ordinary single-cylinder, double-acting, two-stroke-cycle engines has so far been applied only on two 800-hp., three-cylinder vertical engines to be fitted in a vessel of one of the largest steamship companies. Aside from two experimental engines in Professor Junkers' laboratory, no horizontal engines of this type have been built and opinions abroad are much divided concerning the future of the stationary horizontal tandem Junkers engine.

The horizontal, double-acting, four-stroke-cycle Diesel engine in twin-tandem arrangement of 1600-2000 b.hp., which I had the privilege of examining at Halle, is a splendid piece of machinery and operates apparently very satisfactorily on tar oil with a small addition of ignition oil. Although it is claimed that such an engine can be sold in Germany for from $30 to $32.50 per hp. against $35 per hp. for a steam plant, it should be expected that the repair and maintenance cost will be much higher than even that of our present gas engines on account of the high pressures and temperatures and because of carbonization at partial loads which can be prevented only by scrupulous cleanliness.

The small- and medium-sized Diesel engines from 40 or 50 hp. in single-cylinder units up to about 600 hp. in four-cylinder units play an important part abroad. They have come to stay and have reached a high degree of perfection. It is amazing to note how many manufacturers of gas and steam engines abroad have taken up the manufacture of Diesel engines, because they found that the sale of suction gas-producer plants and smaller steam engines has fallen off alarmingly within the last few years.

The reason is plain: The single-acting, four-stroke-cycle, single- or multi-cylinder Diesel engine, but particularly the former, is comparatively simple in construction and operation, especially when its unequalled efficiency is considered. This engine is always ready to be put into service at a moment's notice and has no standby losses; it does not require upkeep and attendance of boilers or gas producers, and its cost compared with that of a steam or gas plant is reasonable. Moreover, it can be installed in the basements of buildings below occupied dwellings, boilers not being permitted in such places in Europe. One of the greatest advantages is that the actual fuel consumption of Diesel engines taken over long periods of operation does not materially exceed the guaranteed figures, whereas in gas producer and steam plants this excess is quite considerable.

While the Diesel engine was originally a vertical engine, the last two or three years have witnessed a remarkable development of the horizontal type. Experience shows that the fuel consumption is very little, if any, higher than that of vertical engines, but the horizontal type is preferred by the customer not only on account of its lower cost, but because of its greater simplicity and accessibility. Especially the horizontal Diesel engine

with Lietzenmayer's open fuel injection nozzle offers these advantages. The open nozzle is simpler, can be more easily taken care of by unskilled attendants, gives better results when burning heavy oils and is less sensitive than the closed injection nozzle of the original Diesel engine.

The only difficulty of which some users of small- and medium-sized Diesel engines sometimes still complain is caused by the discharge valves of the air compressors for fuel injection.

The price of oil fuel in any locality must be considered in connection with the prevailing coal price to determine whether a Diesel-engine installation is economical or wasteful. Cheap oil fuel will favor the Diesel engine only where coal is very expensive, as on the Pacific Coast, whereas high oil and low coal prices will exclude it from any part of the country where such conditions exist.

Considering interest on the investment, depreciation, cost of fuel, attendance, maintenance and repairs, cost of supplies, etc., it is evident that in Europe where fuel prices are from two to three times higher while labor is only one-half to one-third of that in this country, the item of fuel cost constitutes a large part of the total cost of power, particularly since the other items, such as interest and depreciation, attendance and repairs are actually much lower and thus form a still smaller percentage of the total cost.

In this country conditions are practically reversed. Our fuel cost is not only lower and our labor cost much higher than in Europe, but the quality of our mechanical labor is not as excellent. Therefore, the fuel cost plays a much less important part in the total cost of power than the items into which labor largely enters, whereas the lower standard of manufacture and especially of maintenance favors the use of simple, although less economical, machinery.

The formation of syndicates and other influences in Europe have caused the price of oil to go up nearly 100 per cent. within the past few years, and, although this has somewhat retarded the steady growth of the Diesel-engine industry, it has by no means paralyzed it. An adequate liquid fuel has been found, however, in tar oil, a byproduct of the distillation of coal.

France now has to rely almost exclusively on the use of this fuel for Diesel engines, because of the high import duty on natural oils. In Germany a number of collieries have combined to form a company for the utilization of tar which operates a large factory at Meiderich, where, at present, over 300,000 tons of tar a year is distilled, producing over 80,000 tons of tar oil. This is contracted for in larger quantities at a price of approximately $12.50 per ton (5c. per gal.).

The heat value of tar oil is over 17,000 B.t.u. per lb. and its consumption in the Diesel engine is, therefore, not very much higher than that of natural oil varying now from 0.49 to 0.46 lb. per b.hp.-hr., according to the size of the engine.

The operation of medium-sized Diesel engines on tar oil, especially of those above 50 hp. per cylinder and with open fuel-injection nozzles, is satisfactory. For smaller engines the addition of a so called ignition oil is necessary.

Gas Turbines

Renewed activity is noticeable abroad in the matter of designing an adequate gas turbine, and I noticed that much interest is taken in this question by the most prominent manufacturing concerns in Europe. A large experimental gas turbine in Germany is operating splendidly, mechanically speaking, although the problem of satisfactory thermal efficiency still remains to be solved.

Gas Engines

The use of the so called "industrial waste gases" has progressed considerably. The four most prominent manufacturers in Germany alone are now building large gas engines for blast furnace and steel works at the rate of 120 per year and a number of twin-tandem engines of 5000 b.hp. operating very satisfactorily on rich coke-oven gas can now be seen in German coking plants.

The splendid results obtained with gas-blowing engines in Germany have prevented the turbo-blower from getting a firm foothold in that country, although its use in France, England and Russia is steadily increasing. Within the past two years the use of coke-oven gas for producing electric power by gas engines has greatly increased and after overcoming the many earlier difficulties the coke-oven gas engine is today just as safe and reliable as the blast-furnace gas engine. These difficulties arose principally from over-rating the engines, and from sulphur in the engine gas. Since the reasons for the trouble were recognized the causes, or their consequences, are now avoided.

The most prominent gas-engine builders are now governing their engines on a simple combination quantity-quality regulation principle. The valve-gear has been much simplified by all manufacturers, resulting not only in a reduction of the manufacturing cost, but also of operation; and gas-engine prices are, generally speaking, lower than they ever were before.

The gas cylinders, after a few unsatisfactory attempts to use cast steel, are invariably made of cast iron and with the exception of a few types are cast in one piece with an integral water jacket. The gas cylinders are often provided with hard cast-iron liners. The cracking of gas cylinders which formerly occurred too frequently is now much reduced by proper design, selected material and better pouring methods. Experience has taught that the life of a gas cylinder can be materially increased if the water jackets are thoroughly cleaned of all scale and muddy deposit at regular intervals of from four to eight weeks, depending upon the character of the cooling water used.

Gas cylinders on engines driving electric generators crack more frequently than those on blowing engines, because the former carry much higher loads and are usually shut down oftener. It was found that frequent shutting down and starting of gas engines shortened the life of the cylinders materially if the engines were allowed to cool off too much during shutdowns. The average life of a gas cylinder in the large steel plants in Germany is generally reckoned to be from five to six years.

The pistons are generally ribless cast-iron castings, although for large engines cast steel is used. One concern has recently introduced for blast-furnace gas engines one piece solid noncooled, forged steel pistons.

The piston rods are now usually of openhearth steel of about 100,000 lb. tensile strength, uncambered, and their diameter is approximately 26 per cent. of the diameter of the gas cylinders. Crossheads are made of forged nickel steel.

No uniformity of design can be claimed for piston-rod packings, but they invariably consist of single- or double-sectional cast-iron packing rings pressed against the rod by garter springs and separated by solid one-piece rings forming from 12 to 14 chambers. It is claimed that the clearances of the ring sections must be reduced to a minimum. So called fire rings are sometimes used, but several builders object to these solid rings.

The exhaust valves are of forged steel or Durana metal and are now almost exclusively of the noncooled mushroom type. Some builders, however, still prefer water-cooled exhaust valves for coke-oven gas engines.

Gas Cleaning Devices

The progress of the large gas engine in Europe has been accompanied by great improvements in the method of purifying blast-furnace gas. Until recently two systems of cleaning gas for engine purposes were largely used, the hydraulic fan and the Theissen washer. Results obtained with these appliances regarding both cleansing efficiency and cost of operation are satisfactory. Nevertheless, they now are superseded in Europe by new processes, which still further reduce not only the cost of the installation, but principally the cost of purification on ac-

count of the smaller power and water consumption.

One of these new methods is a radical departure from the former wet-cleaning processes and works on the principle of filtering the gas in a perfectly dry condition. The filtering is achieved by means of filter bags, made of special fabric and suspended vertically in several air-tight compartments. The raw blast-furnace gas coming from the furnaces at high temperature is first cooled by radiation to about dew point (around 130 deg. F.) and is then superheated 20 to 30 deg. F. by using any available source of heat, such as waste heat from gas engines, exhaust steam or even the sensible heat of the raw blast-furnace gas itself. By cooling the gas a large amount of moisture is removed and the scorching or burning of the filter bags is prevented, whereas the subsequent superheating avoids condensation of the remaining vapor and clogging of the bags.

The raw gas under the action of a fan passes from the inside to the outside of the filter bags, depositing all impurities on the surface. The layer of dust naturally increases continually and would eventually render the bags impenetrable, were it not for an automatic mechanism which shakes the bags at regular intervals of about four minutes and causes the dust to drop off. Simultaneously the compartment which is to be cleaned is automatically shut off from the others, and the direction of the gas current is reversed; the purified gas surrounding the filter bags now enters the latter, opening the pores of the filter material.

Gas Producers

It seems extraordinary that in spite of the high price of coal the development of the gas producer for power purposes has not kept pace with that of the gas engine in Europe. For many years the Mond gas producer with sulphate of ammonia recovery has been used extensively in England, and recently much interest was shown in Germany in plants that offer material financial advantages over the ordinary gas-producer installations. Since much sulphate of ammonia having a ready market as an artificial fertilizer, can be recovered from coal, the cost of gas making is considerably reduced, and installations are known where the saving thus obtained reaches $2 per ton of coal, which in many cases is as much as the value of the coal itself. The gas manufactured in such plants has a heat value of about 150 B.t.u. per cu.ft., and is of very uniform quality. On account of the removal of tar and dust, it is particularly adapted for gas engines, but its low cost makes its use profitable for metallurgical purposes.

The growing demand for coke has caused a remarkable increase in coke-oven plant activity. The days of the wasteful "bee-hive" coke oven are past, and the highly economical byproduct coke-oven is gradually taking its place. The manufacture of coke entails the accumulation of large quantities of coke breeze. Successful attempts were recently made to gasify this coke dust in special gas producers. One system operates with liquid slag very much on the order of a blast furnace, whereas another adapts a well known European producer design to the special requirements of gasification of very fine fuels.

Overloading Gas Engine

In one steel plant a novel system of overloading four-stroke-cycle gas engines 25 to 35 per cent. of their normal capacity is employed. This consists in scavenging ordinary four-stroke-cycle gas engines and introducing the fresh mixture under a pressure somewhat higher than atmospheric. This is achieved by raising the pressure of the combustion air and gas in turbo-blowers to a gage pressure of a few inches of mercury and by scavenging the spent gases of the previous power stroke with a blast of this air, subsequently introducing gas and air under slight pressure into the combustion chamber. By utilizing the waste heat of the gas engines for raising steam in suitable waste-heat boilers and driving steam turbines direct connected to the turbo-blowers, the power requirements of the auxiliaries can amply be supplied without resorting to outside sources of power, nor detracting from the increased output of the gas-engine.

Discussion

Following Mr. Freyn's report, W. R. Haynie, American representative of the Carels Frères Diesel engine, offered a few remarks substantiating in general the preceding speaker's remarks. Mr. Haynie expressed the opinion, however, that, although he had been unable to find any of the large Diesel engines talked about, he nevertheless believed that since commercial engines of 1000 hp. had been developed, it would be only a matter of time before 2000- and 3000-hp. units would be built successfully. He believed the future of the Diesel engine lay largely in marine practice. He also stated that many manufacturers considered 150 hp. per cylinder to be the commercial limit for four-stroke-cycle engines, although two-stroke-cycle engines could be built much larger.

Referring to oil fuels for Diesel engines, Mr. Haynie said:

"There is a question as to whether the heavier petroleums can be used successfully in Diesel engines. Generally, however, if the oil be free from noncombustible foreign matter, it can all be readily and profitably used.

"Among the Diesel-engine builders some raise the compression pressure and proportionately the injection pressure to create a higher temperature in the cylinders when using the heavier oils. This is not in conformance with the true Diesel principles. The purpose of creating heat by compression of air in the Diesel cylinder is to raise the temperature of the air to such a point that the process of combustion will take place.

"It may be stated that there is no petroleum residue of whatever heavy specific gravity, the ignition of which is not attainable by the temperature created by 30 to 35 atmospheres of compression. The combustion of the oil after ignition takes place is altogether another operation.

"The complete combustion of fuel oil in the cylinders after ignition by the temperature of the compression, is entirely dependent upon the process of handling the fuel before entering the cylinders. The period during which the oil has to be consumed is very short; in fact, a small fraction of a second; therefore, the process of combustion must take place almost instantaneously. The rapidity with which the oil is consumed, does not depend upon the specific gravity of the oil used, or its chemical properties, but upon the size of the particles or globules into which the oil is broken before meeting the air at ignition temperature.

"When using heavy residues in Diesel engines without reducing the size of the particles or globules, only the surface of the particles are consumed in the short period allowed for combustion, with the result that the interior portion is converted into a pitchy substance, which deposits on the surface of cylinders, piston and valves, causing first the use of a greater volume of oil than is necessary, and second the clogging of the cylinder and valves. Therefore, the oil should be completely atomized so as to effect complete combustion within the short period of approximately $\frac{1}{10}$ of the stroke."

Professor Magruder followed Mr. Haynie with some interesting statistics regarding the gasoline engine. These were to the effect that Great Britain has 150,000 motor cars for a population of fifty million, whereas in this country there are approximately 980,000 motor cars for a population of nearly one hundred million. This reduces to three and one-third cars per thousand people in Great Britain as against 10 cars per thousand people in this country; that is, this country has three times as many per thousand.

Further statistics given by Professor Magruder showed that the present annual output of gasoline in this country is sufficient only to operate 5 per cent. of the gasoline engines now sold and in operation in this country if run continually at their rated load.

In regard to the Diesel locomotive Professor Magruder stated that it was prohibitive as to price, and contained other detrimental features which made it thus far commercially impracticable.

Heating and Ventilation

Considered as power-plant problems. Layout and operation of systems and apparatus

Heating and Ventilation of Mount Vernon High School

By Ira N. Evans

The following is a description of the heating, ventilating and power equipment as designed for the new high-school building of Mount Vernon, N. Y. The building is now in the course of erection from plans of Architects Goldwin, Starrett & Van Vleck and the heating and power equipment involving several novel features in this class of work are under the supervision of Douglass Sprague and Associate, consulting engineers, of New York City.

Fig. 1 shows the building, which will have three stories and basement, and in plan will be about 300x100 ft., with an ell on the rear, 50x100 ft., for the gymnasium and auditorium. There are at present about 30 rooms with the usual space allotted for administration purposes. The toilet, locker and lunch rooms are in the basement. The south end of the building will be terraced and excavation made to form a sub-basement for the boiler, engine and fan rooms, with large coal bunker capacity. This removes all machinery from the basement proper, leaving it free for school purposes. The locker and toilet rooms at the rear of the building will be roofed over for the present, but at some future time these walls may be extended, allowing additional space for six classrooms on each floor or eighteen in all. A hot-water heating system under forced circulation is to be installed and capacity is provided at present to include the eighteen rooms without disturbing the present layout in any respect. Tees are to be left on the water mains for the additional risers and direct radiation and openings in the main air duct for the rising flues. Fig. 2 shows the piping system for one-half the building as it will be when these additional rooms are built, the other half being in duplicate.

Fig. 3 is a plan of the boiler and engine rooms. Two boilers of the return-tubular type, 60 in. in diameter and 17-ft. tubes, equipped with the Sidney Smith patent setting will be installed and will aggregate about 300 hp.

Each boiler will be provided with two circulating coils either side of the fire-box so arranged as to furnish prime heating surface, or provide means of heating the boiler-feed water before it enters the shell. This is especially advantageous when cold water is used.

A 6x4x6-in. feed pump will be provided for feeding the boilers through a closed feed-water heater and two filters with bypasses and valves. An injector is also provided. As all live steam will be returned to the boilers by gravity and all exhaust-steam condensation from the heaters can be utilized immediately, only a small receiving tank will be provided. There will be an automatic governor for the pump and a cold-water connection for makeup water. Due to the high temperature, 210 deg. F., and immediate return of condensation from the heaters, the steam requirements for feed water will be slight.

Each boiler will be provided with three nozzles: one for the safety valve, a 6-in. connection for the live-steam heater, and another 6-in. nozzle for the main steam header supplying the engines.

The high-pressure piping is made up of standard pipe and extra-heavy flanged cast-iron fittings capable of withstanding an operating pressure of 125 lb. if desired, although 80 to 100 lb. will probably be carried.

Bosses on the bottom of all tees in the main header will be tapped for a 1-in. drip connection to drain the main to a high-pressure trap in the engine room. This trap will also take the drip from the steam separators on the engines and discharge into the feed-water and exhaust heaters. There will be a line from each boiler for operating the feed pump independently in case the main steam header is out of commission. All steam connections from the boilers will have two valves, as shown, with open bleeders between the valves.

In the engine room two direct-connected units will be provided for lighting and power, one of 125 kw. and the other of 60 kw. capacity. A 10-in. pipe will handle the exhaust and this will communicate through an oil separator with the feed-water and exhaust-steam heaters. The piping will be arranged so that the steam may be bypassed around either heater to the atmosphere. The drip from either heater will be passed to the return tank shown underneath.

Domestic hot-water heaters situated back of the boilers will be arranged to use live steam through a reducing valve and will be trapped back to the feed-water heater as indicated in Fig. 3. A cast-iron tank heater will be provided for use with these hot-water tanks when the boilers are out of commission.

All steam and exhaust pipes, heaters, etc., will be covered with the best non-conducting material and in the sub-basement all return and supply water pipes of the heating system. The balance of the water piping will be exposed as heating surface and thereby reduce the temperature of the circulated water.

Fig. 1. Mount Vernon High School

Tables 2 and 3 show that one boiler will take care of the building in average winter weather above 35 deg. F. There will be less than 200 hr. when two boilers will be required. It should be borne in mind that when the greatest power is required during sessions, the sun is high and the outside temperature fairly warm. As school is in session only about 25 per cent. of the season requiring heat, there is plenty of time to make any necessary repairs between sessions, especial-

ly if the plant is in competent hands. This reduces the duplication usually necessary. With direct radiation and air supply for ventilation there will be practically two systems of heating, either of which could be utilized independently of the other if occasion required.

In addition to 40 cu.ft. per min. for each pupil, or a total of 70,000 cu.ft., air supply will be provided in the corridors, offices and administration rooms. In the corridors this is an innovation of the engineers who have found it advantageous in keeping the air of the building in better condition. The air supply will be taken through a window at the rear of the building which sets high and free from surrounding structures.

The hot-water indirect stack is composed of vento radiators two sections high and five deep, aggregating 6080 sq.ft. of surface. This stack will raise the air from 0 to 115 deg. with the water averaging 200 deg. The vertical flues to the different rooms will be set in the corridor walls, and the main air ducts will be run on the sub-basement ceiling in the boiler and engine rooms and under the floor of the basement corridor as the ducts pass to the north end of the building. A No. 15 multivane fan belted to a 40-hp. motor and of capacity to deliver 70,000 cu.ft. of air per min. against ¾ oz. pressure will be installed. The air velocity will be 500 ft. per min. in the flues and one-half that entering the rooms. Wire grills will be provided throughout for both air inlets and outlets. A separate duct under the main air duct will furnish the air supply to the gymnasium and auditorium. Dampers operated by air pressure from the automatic heat control system will be provided so that these rooms may be cut out when unoccupied.

Ventilating ducts from all of the rooms will lead to the attic space and into a trunk line duct connecting to each of two 84-in. propeller fans driven by 10-hp. motors. Provision will be made to discharge the air through a large ventilator on the roof. About six air changes per hour will be made in all classrooms. The provision of a power plant makes it possible to drive these fans by motors, thus gaining in economy over the use of small fan engines which are very wasteful of steam. Motors would also prove expensive if operated on public-service current at 6c. per kw.-hr.

A complete system of direct radiation composed of cast-iron radiators and 1¼-in. pipe coils operated on hot water will be provided to maintain the building at 65 to 70 deg. when school is not in session. The completed building will contain about 14,000 sq.ft. of radiation designed to furnish the requisite heat necessary in extreme weather with a maximum average water temperature of 200 deg.

In Fig. 2 is shown one of the special coils designed to eliminate the spring pieces and provide for expansion. This arrangement will allow space for the risers to run to the floor above without passing behind or in front of the coil. The system will comprise four main circuits: one for the north end with 4-in. supply and return mains, one for the south end with 3-in. mains, a 2½-in. supply and return for the gymnasium and auditorium and 5-in. pipes for the indirect stack. All returns will be brought back to the header near the pump with a separate valve and thermometer in each case. The supply mains will be run in the basement ceiling and the returns in the sub-basement. The north and south circuits will be divided into two 2½-in. loops, with valves and drawoffs connected at either end in all cases and so arranged that any one of the sections may be drawn down quickly in case of trouble without interfering with the remainder of the system.

The basement coils and radiators will be connected with 1¼-in. pipes in such a way that the drop-in head on the main 2½-in. line at the ceiling level will be sufficient to raise the water in the radiator when cold against the force of gravity and establish circulation.

All first-floor coils will have 1-in. connections shunted off the 2½-in. main, and 1¼x1-in. and 1½x1¼-in. risers will take care of the second and third floors in the same manner.

No valves will be provided on the risers as the two stop valves above referred to and drawoffs are considered sufficient in case of repairs. In all cases the riser connections will be crossed on the 2½-in. mains in such a manner as to enlarge the area by that of the riser when the surface is turned on.

The gymnasium and auditorium will be supplied by two circuits shown in Fig. 2 and so that the radiation on either side of the room may be drawn off or controlled separately. The radiators in the auditorium will have 1-in. connections shunted off the 2-in. main on the ceiling of the gymnasium. The 5-in. main to the indirect stack will be divided into five 3-in. connections so as to maintain as rapid a velocity through the stack as possible without too great a drop in head.

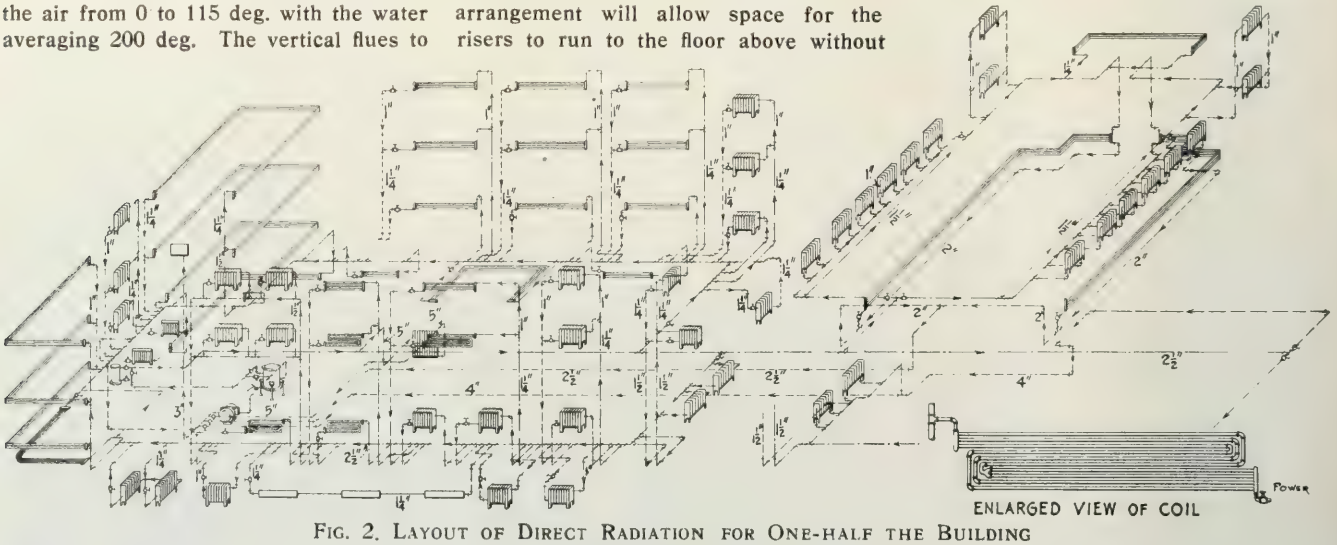


FIG. 2. LAYOUT OF DIRECT RADIATION FOR ONE-HALF THE BUILDING

In Figs. 2 and 3 are shown the connections to the heaters and pump on the hot-water system. The pump has a capacity of 1200 gal. per min. at 75-ft. head, and a 30-hp. motor and steam turbine are connected with a clutch so that the pump may be operated by either. The power may seem excessive but as all exhaust steam can be utilized, as per Tables 2 and 3, and all heat is returned in friction to the water, it is good practice in this case. The system is designed to give a maximum drop of less than 15 deg. under extreme conditions. The high velocities and capacity make it possible to use small mains. The reduction in size of the distributing mains in the north and south sections to 2½-in. minimizes the labor of cutting in tees.

The exhaust heater is 6 ft. 6 in. in diameter and 10 ft. long. The water is passed around the tubes and the steam through them on the counter-current principle. The live-steam heater is 3 ft. 6 in. in diameter and 12 ft. long, and is capable of taking care of the entire plant.

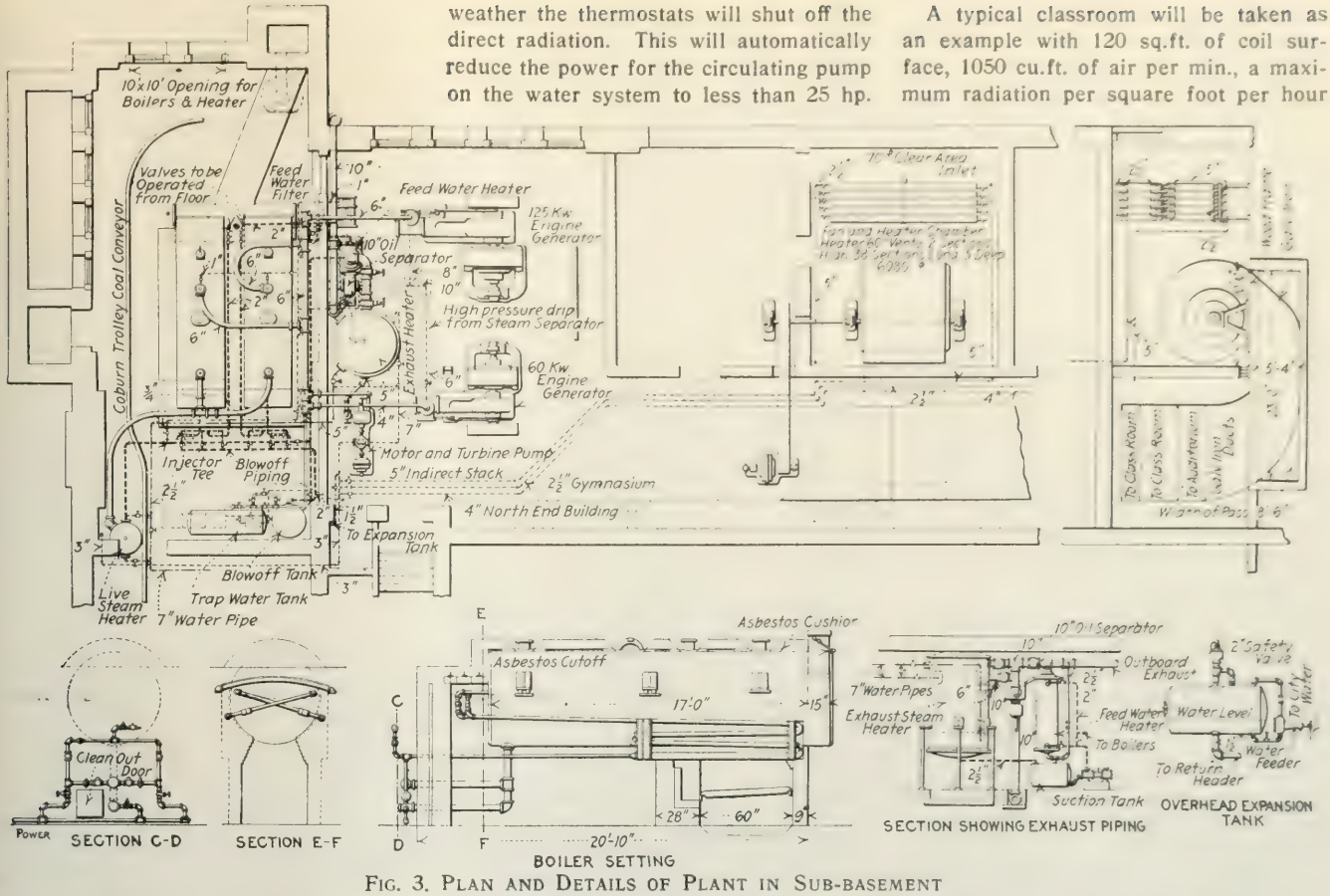


FIG. 3. PLAN AND DETAILS OF PLANT IN SUB-BASEMENT

It is provided with a gravity return to the boilers. A bleeder connection is made to the 2½-in. drip line so the condensation may be discharged to the feed-water heater and exhaust heater temporarily and then returned to the boilers by the feed pump. From the live-steam heater, the 7-in. water main will be run on the basement ceiling and reduced as the different circuits are taken off.

A complete automatic temperature-control system of the pneumatic type is provided for all main rooms on the direct radiation and to operate dampers. The indirect stack, however, is controlled by hand, as the residual heat in the water after the stack is shut off would interfere with automatic regulation. It is the intention to regulate the temperature of the entering air so that in all but extreme weather the thermostats will shut off the direct radiation. This will automatically reduce the power for the circulating pump on the water system to less than 25 hp.

Many of the rooms, such as toilets, will have direct radiation and no air supply. Consequently if the load on the direct radiation for the whole building were taken to find the necessary raise in air temperature to supply the heat radiated from the building when maintaining a temperature of 70 deg., it would bring the entering air up to 115 or 120 deg.

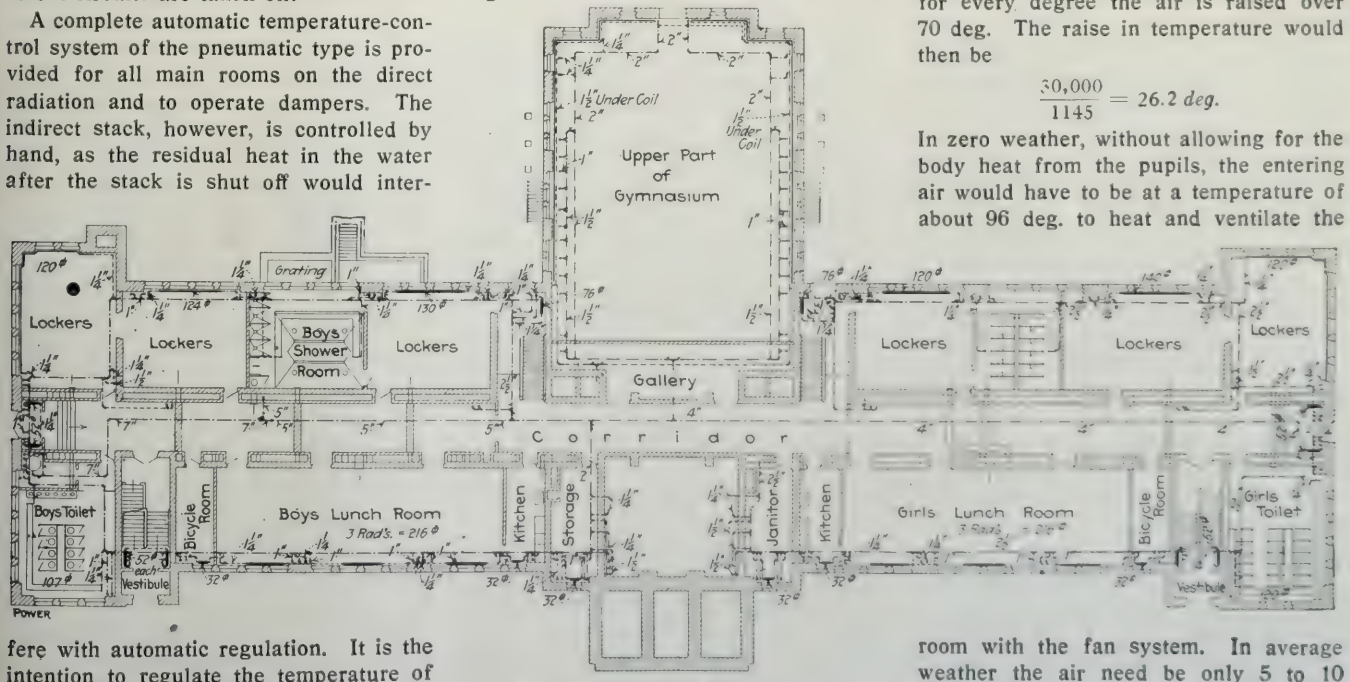


FIG. 4. PLAN OF BASEMENT

A typical classroom will be taken as an example with 120 sq.ft. of coil surface, 1050 cu.ft. of air per min., a maximum radiation per square foot per hour of 250 B.t.u., and 55 cu.ft. of air cooled or heated 1 deg. will require 1 B.t.u. Then.

$120 \times 250 = 30{,}000$ *B.t.u. per hr.*

will be transmitted by the coils, and to supply the loss of heat per hour to the outside with the air will require

$$\frac{1050}{55} \times 60 = 1145 \text{ B.t.u.}$$

for every degree the air is raised over 70 deg. The raise in temperature would then be

$$\frac{30{,}000}{1145} = 26.2 \text{ deg.}$$

In zero weather, without allowing for the body heat from the pupils, the entering air would have to be at a temperature of about 96 deg. to heat and ventilate the room with the fan system. In average weather the air need be only 5 to 10 deg. above 70 to do the heating, and any

shortage may be made up by the thermostat turning on the direct radiation automatically.

Table 1, taken from an article in the Sept. 12, 1911, issue, shows the number of hours for each 10-deg. period of outside temperature during the heating season in the New York district. The data were obtained from the weather-bureau records. The entire season comprises 5034 hr. or 7 months. In school work there would also be two months; 40 days', operation after the heating season when only ventilation would be required and the temperature outside would be from 60 to 70 deg.

This school will be in operation from 8 a.m. to 1 p.m., 6 hr. per day, or 240 hr. operation after the heating season closes. During the seven months there will be in the present case 31 Saturdays of 5 hr. In the previous article, day operation was taken at 10 hr. instead of 6 and 5 hr. on Saturday. The school under consideration will then be occupied

$$(1545 - 155) \times 0.6 = 834 \textit{ hr.}$$

and 3192 hr. of night heating will be required. This does not include the 50- to 60-deg. period during nonsessions as presumably the plant will be shut down. Thus the school is actually in session about 20 to 25 per cent. of the heating season.

Tables 2 and 3 give the heat required for each outside period, the probable water temperature, transmission, boiler power and total steam required. The loss from the building is given for the different temperature periods and in Table 3 this loss is added to the heat for ventilation to obtain the total steam required. This is figured on continuous operation and, although the system may be shut down nights or over Sundays, the coal required will be the same and possibly somewhat more. The direct radiation is here figured operative during the 50 to 60-deg. period, but in practice it may be possible to shut down at that temperature, making some reduction in coal consumption.

The main engines will operate on 32 lb. of steam per i.hp.-hr. or on about 48 lb. per kw.-hr. at the switchboard. The total power required when ventilation is needed will be about 95 hp., including attic fans, supply fan and circulating pump. At 32 lb. per hp.-hr. this would provide 3040 lb. of exhaust steam per hour, which is just about what is required for heating in the 50 to 60-deg. period. During the 60 to 70-deg. period the exhaust will pass to the atmosphere as practically no heating is required.

At night the steam-turbine pump on the heating will require 20 hp., which at 60 lb. will require a steam consumption of 1200 lb. per hr. This is lower than any period in Table 2, so nothing is lost. In case street current were used and low-pressure steam with automatic heat control, the labor would be somewhat reduced, but possibly larger boilers or grates would be necessary to burn the coal at a lower rate. This coal would cost $6 per ton of 2000 lb., and 8 lb. evaporation will be assumed. The motor pump on the water system would be eliminated, leaving 60 hp. required by the fans for 1074 hr. Allowing 10 per cent. loss between boiler and engine raises the

TABLE I. HOURS FOR EACH TEMPERATURE PERIOD

Outside Temp. Deg. F.	Sessions		Nonsessions	
	Per Cent.	Hours	Per Cent.	Hours
0–10	0.60	5	1.60	67
10–20	3.25	27	6.13	257
20–30	13.25	110	17.83	750
30–40	32.40	270	27.82	1168
40–50	22.70	190	22.62	950
50–60	27.80	232	24.00	....
	100	834	100	3192
60–70		240		
Totals...		1074		3192

horsepower to 66, which practically equals 50 kw. Then the season load amounts to

$$1074 \times 50 = 53{,}700 \textit{ kw.-hr.}$$

Table 2 gives the total steam during nonsessions as 7,068,516 lb. and for sessions Table 3 gives 4,020,240, making a total of 11,088,756 lb. per season. Current at Mount Vernon costs 6c. per kw.-hr., so the charge per season would be

$$53{,}700 \times 0.06 = \$3222$$

With 8-lb. evaporation the season tonnage of coal would be

$$\frac{11{,}088{,}756}{8 \times 2000} = 693$$

which at $6 per ton would cost $4158, making a total of

$$\$3222 + 4158 = \$7380 \textit{ per season}$$

In Table 3 it is shown that the power will furnish excess steam only in the period of 60 to 70 deg. for the 240 hr. after the heating season is over and school is still in session.

In the case of the plant as installed, assume an evaporation of 7.45 lb. and pea coal at $4 per ton instead of an evaporation of 8 lb. and $6 per ton for egg and broken coal. For the fans 50 kw. will be required for 240 hr. At 48 lb. per kw.-hr. they would require

$$50 \times 48 \times 240 = 576{,}000 \textit{ lb. steam}$$

Subtracting the 91,440 lb. steam for the 60 to 70-deg. period, Table 3, gives 484,560 lb. to be added to the totals of Tables 2 and 3. The total steam would then be 11,088,756 + 484,560 = 11,573,316 *lb.* and the coal required

$$\frac{11{,}573{,}316}{7.45 \times 2000} = 776.7 \textit{ tons}$$

The cost of operating the entire plant would be

Coal 776.7 tons at $4	$3106.80
Extra water	10.00
Oil, waste and supplies	100.00
Labor of engineer and fireman	2000.00
Int. and Dep. on $6000 for engine, generator and piping at 10 per cent	600.00
	$5816.80

This shows a balance in favor of installing the engine and operating the entire plant of

$$\$7380 - 5816.80 = \$1563.20$$

It is not at all improbable that a large portion of this labor would be obligatory in either case. When nearly all the fuel and all the boilers are provided, operating an engine for power is highly economical. The above figures show the saving. If the difference in price of the two grades of coal had been $1 instead of $2, the difference in cost of operation on the same basis would be $526.60 less, or $1037.60. Even this is over 17 per cent. of cost of engine equipment.

It might be asked why coal at different prices is used. With pea coal the fires need attention constantly and when running the boilers for heating only, a fireman would be necessary. This would increase the labor charge. The egg or broken coal can be banked higher and left for longer periods.

TABLE 2. HEATING NONSESSIONS

Outside Temp. Deg. F.	Hours Each Period	Av. Temp Water Deg. F.	Temp. Water 65 Deg.	Trans. B.t.u. per Deg. Dif.	B.t.u. for 14,000 sq.ft.	Lb. Steam per hour	Boiler Hp. (34.5 lb.)	Total Steam Lb.
0–10	67	195	130	1.8	3,276,000	3276	95	219,492
10–20	257	185	120	1.8	3,024,000	3224	88	777,168
20–30	750	170	105	1.7	2,499,000	2499	72	1,874,250
30–40	1168	155	90	1.7	2,142,000	2142	62	2,501,856
40–50	950	140	75	1.7	1,785,000	1785	52	1,695,750
50–60	1008	130	65	1.6	1,456,000	1456	42	
								7,068,516

TABLE 3. VENTILATION AND HEATING (SESSIONS).

Outside Temp. Deg. F.	Temp. Air Ent. Rm. Deg. F.	Loss Ducts Deg. F.	Raise in Temp. Air Deg. F.	Lb. Steam (76.36 per Deg.)	Direct Rad. Steam for Heating, lb.	Total Lb. Steam per hr.	Hours Each Period	Total Lb. Steam Each Period	Boiler Hp. (34.5 lb.)
0–10	70	10	75	5727	3276	9003	5	45,015	261
10–20	70	10	65	4963	3024	7987	27	215,649	231
20–30	70	10	55	4200	2499	6699	110	736,890	194
30–40	70	8	43	3283	2142	5425	270	1,464,750	157
40–50	70	5	30	2291	1785	4076	190	774,440	118
50–60	70	5	20	1527	1456	2983	232	692,056	86
60–70	65	5	5	381	....	381	240	91,440	11
							Total.....	4,020,240	

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POWER

The Steam Specialty Man

Not infrequently, when the steam-specialty man sends his card to the engineer, or comes personally into the engine room, he is cut short before he has an opportunity to explain his proposition or show his sample and leave descriptive matter. He is regarded as a pest, unless the engineer is interested in his particular line or is in the market. There are times, of course, when the engineer is particularly busy and his mind is concentrated on the work in hand and no matter how interested he may be he cannot give any time to the salesman. As a rule, however, the engineer could devote a few moments to him, and he would find it a benefit to do so in the long run.

Many of the specialty men are practical men, who have been in the operating field. Such men are qualified through past experience to explain the operation of their specialty, practically and clearly and make useful suggestions. This practical explanation, with the descriptive matter given by the manufacturers of the article, serves to keep the engineer posted in recent developments in the business. It simply means additional knowledge to the engineer, which generally is superior to information gained from a textbook because it is explicitly explained by a practical man. Often he has a model or sample showing the construction and actual operation of a piece of new apparatus. An engineer may not have, or is unfamiliar with the article a salesman carries, but there is a possibility of his changing his position and finding just such a piece of equipment in the new plant—the information then comes in handy.

Besides this, the salesman is continually visiting plants and has the opportunity of seeing things and learning new kinks. He sees repair jobs being done, and the different ways of doing them. He notices some unusual wrinkle worked by an engineer on an emergency job and learns of new methods used in routine operation. As it is his business to interest the engineer, naturally he usually talks shop. He will mention he saw such and such a scheme carried out today and explains how it was worked. In this way the engineer obtains some more information he is not apt to find in books, and which may prove valuable at some future time.

On the whole, devoting a few moments to the specialty man is a decided benefit. He will not go away disgruntled because he did not make a sale at the time, for he left the engineer interested and gained a prospect who will remember him when later in the market, or who will present his proposition, if a good one, to the employer when the right time arrives.

The least the engineer can do is to be civil to the specialty man, and if too busy to see him at the time, invite him to call again. It pays in the long run. One way in which the specialty man sometimes proves of direct personal service to the engineer is in acquainting him with a new or better position, being very likely to hear of such opportunities.

Give Firemen a Chance

There is something about firing steam boilers that fascinates men who do that work just as there is about railroading or horses that fascinates those engaged in occupations where cars and horses are used.

Perhaps this is one reason why so many firemen have fired boilers all their working lives. The other and principal reason is, that once having learned to fire many men continue to do so because they could not work at other occupations and receive as much in wages as they do while firing. This latter condition is true of most men and plants.

To those not afraid of hard work and who simply want a job, the boiler room offers a good field. But a progressive fireman should consider his work in the boiler room only as preparatory to qualifying him for entrance into the engine room of that plant or of some other plant. To wait for promotion to engi-

neer in the plant where he has learned to fire means too great a loss of time in most cases. If he is a married man he feels that he cannot afford to go into the engine room as oiler on a wage of two or three dollars less per week than he receives as a fireman. On the other hand, if he seeks a position as engineer in another plant he must either lie about his experience in the hope of being taken on or be promptly rejected if he tells the truth. The employer cannot be censured; he wants an engineer, not a fireman. But the fireman wants a position as an engineer, and the more competent, ambitious and self-confident he is the more he is likely to lie about his experience. And there is some excuse for doing so. He wants advancement and finds the channels of progress blocked to him, so he follows the next easiest course.

There is need of greater and better attention to this matter. Surely such conditions work great harm not only to chief engineers and firemen, but to the whole field of power-plant engineering. There should be a general understanding among chief engineers that they will see to it that firemen desirous of advancing in engineering are given the opportunity of acquiring the essential experience. This means that the fireman, or those who wish to advance, would have to do duty in the engine room or with the repair gang. It means, too, that the chief engineer must be sure that a fireman is sincere, so that no shirker would be allowed to do other work just because he wanted a diversion. It is, however, easy to detect a man whose purposes are false. If the helpers in the repair gang and also the oilers were allowed to alternate their work with that of the firemen occasionally, it would be of immense value to all, and incidentally it would help the engineer-in-charge because there would be men available who could successfully do more than one branch of work.

A plan of this kind would mean that a fireman would not need to lie to get a position he was capable of filling. The chief engineer would know where and how best he could use such a man before he hired him.

Best of all, perhaps, is the fact that a man naturally adapted to engineering, but whose economic circumstances prevent him from otherwise getting the necessary experience, would have open to him the means of quickly developing into a competent engineer.

We hope that engineers will work for the realization of the above suggestions. By doing so they will not only make good friends but good engineers.

Falling Expansion Line with Superheated Steam

The author of a recent paper upon superheated steam is troubled by the fact that when superheated steam is used the expansion line of the indicator diagram falls away more rapidly than it would with saturated steam, and that, therefore, in order that the engine shall develop the same amount of power, the pressure must be raised or the cutoff be made later. This does not mean, however, that the engine is using more steam or more heat per unit of power developed; quite the contrary.

The holding up of the expansion line with saturated steam is due to the fact that when the steam is admitted to the cylinder, it gives up some of its heat to the containing surfaces, which have just been exposed to the temperature of the exhaust and are therefore cooler than the entering steam. Being only saturated, that is, having only enough heat to maintain it in the form of steam, it cannot give up any without a corresponding amount of condensation. At the point of cutoff there is therefore present in the cylinder, not dry steam but a mixture of steam and water.

If this mixture were expanded without taking up any heat from the cylinder walls or giving out any heat to them, the line would fall away considerably faster than does the line of the ordinary indicator diagram. As the expansion proceeds, some of the water which is above the boiling temperature at the lower pressure evaporates into steam, raising the pressure above what it would otherwise be. The expansion line of the diagram from an ordinary cylinder with saturated steam, does not represent therefore the relative pressure and volume of a constant amount of steam, but of a varying amount.

While this holding up of the expansion line adds to the area of the diagram and the power developed, it does so at an enormous expense. To heat up a cylinder with steam of boiler pressure and then to use a little of that heat at a lower temperature, to boil water to keep up the expansion line, allowing the greater part of it to keep on boiling out water after the exhaust valve is open, and to make steam to put into the exhaust, is a most inefficient way of fattening the diagram. The perfect engine would avoid this heat leak between the boiler and the condenser. A cylinder which absorbed no heat from and gave no heat to the working medium, would be the greatest improvement to which the steam engine is susceptible. The use of the steam jacket and highly finished internal surfaces are steps in this direction.

The use of superheated steam goes further than either of these. While it cannot preclude the absorption and giving out of heat by iron exposed alternately to high and low temperatures, this transfer is much less active between dry superheated steam than between moisture and the cylinder surfaces and with the use of superheated steam, the giving up of heat to the steam by the cylinder during the expansion is much less energetic and the expansion line is not raised so much above the normal as in the case of the saturated steam. But the heat which would be available for this purpose has not been taken from the entering steam in the first place, and if the author will compute the number of heat units which go to the condenser in both cases, he will find that, notwithstanding the longer cutoff or higher initial pressure, a smaller number of heat units are thus voided per horsepower-hour with superheated steam and its more rapidly falling expansion line than with saturated steam and the fuller diagram.

It speaks well for the carefulness of the boiler inspection in Montana that not a single boiler explosion has occurred during the present governor's administration (Gov. Norris' term of four years expires in January, 1913). In making his report, State Boiler Inspector J. H. Dailey showed also that the work has been growing increasingly difficult yet the department has not only been self-supporting, but in the past year has turned over to the state in excess of its expenses nearly seventy-five hundred dollars. Examples such as this one should be an inspiration to other states.

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Oil Barrel Emptier Improved

Becoming interested in the device for lifting oil from a barrel as described by Mr. Ellsworth in the Sept. 17 issue, I tried to improve it as described below:

The materials used were a 1-in. tee, a ¾-in. short nipple and two 1-in. pipe bushings. One of the bushings was chucked in a lathe, and the wrench grip cut off, leaving the bushing about ⅜ in. long threaded the full length. This part of the bushing is shown at *A*, Fig. 1. Two holes *BB* were drilled in it so it could be screwed down into the thread on the tee. Then the lower end of the bushing *C* was faced off. The next step was to run the bushing *A* into the thread of the tee and screw the second bushing down to lock it in place. The tee was then "chucked" so that the holes in the bushing ran true. The size of this hole was then increased to allow a free fit for the brass tube *D*. The ¾-in. nipple *E* was then screwed into the other end of the tee and the ½-in. hose connection *F* was provided.

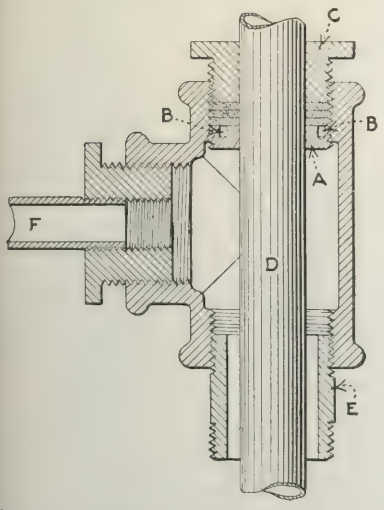


FIG. 1. SHOWING CONSTRUCTION OF AIR OIL LIFT PUMP

After the brass tube had been slipped through the tee the upper bushing *C* was unscrewed and some soft flax packing covered with graphite was wound down into the shallow stuffing-box. This joint proved to be a very good one. Fig. 1 shows the tee assembled.

This air oil lift is used in the same manner as that shown by Mr. Ellsworth. The tee and all its parts are attached to the barrel by screwing the nipple *E* into the bunghole of the barrel. Air under pressure is admitted at *F* and the oil flows from the top of the tube *D*. The outlet end of the tube *D* could readily be fitted with a hose connection if the oil had to be conveyed to a distance which, of course, is limited by the air pressure available.

When one has to open an oil barrel to use a device of this kind, a bunghole auger, Fig. 2, is very handy. All the mill-supply dealers carry these tools, which enable one to easily open a barrel. The business end of this tool looks like an auger bit. Toward the top it assumes the form of a cone with a groove in it which forms the cutting edge. A few gage marks on this cone enable one to get a bunghole bored to fit any desired size of pipe.

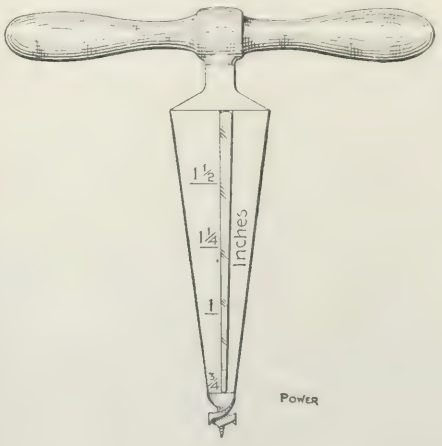


FIG. 2. BUNGHOLE AUGER

JOSEPH DEANE.

Cleveland, Ohio.

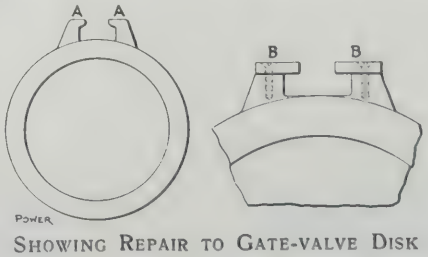
Repair to Worn Gate Valves

Recently we found that the lugs *AA* on the gates of some of our gate valves had become so worn that the nut on the end of the valve stem was likely to pull away from the gate, especially if the valve had been tightly closed.

To avoid trouble we chipped and filed off the lugs. Two pieces of ¼-in. wrought-iron plate were then screwed on each side of the lugs, as shown at *BB*. The repair required but little time and the saving effected more than paid for the cost of making the repair.

G. B. JOSHI.

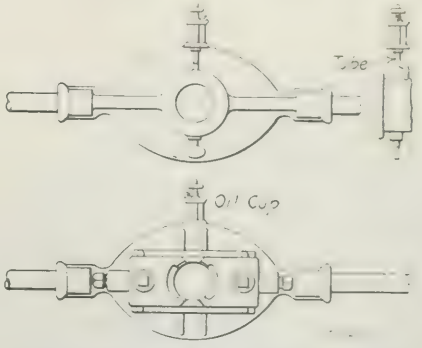
Poona, India.



SHOWING REPAIR TO GATE-VALVE DISK

Novel Oiler for Pipe Dies

Having quite a lot of pipe to thread, I thought of a plan that would enable me to dispense with the oil can that is such a nuisance when threading pipe. I hunted up a discarded oil cup and had the machinist drill and tap the stock (in the back) so that I could attach the oil cup. I bent a piece of copper tube and soldered it at the opening of the oil cup and also at the opening in the stock. The oil feed may be adjusted for different pipe sizes.



PIPE-THREAD CUTTING STOCKS FITTED WITH OIL CUP

Of course, there is room for improvement. The handles of the dies could be used to carry the oil and an adjustable feed could be easily obtained.

ROBERT GEILING.

Metuchen, N. J.

Oil Can Causes Turbine Over-Speed

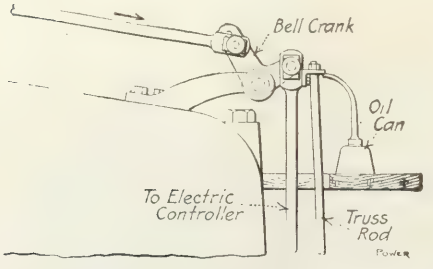
One Sunday afternoon, when the turbine was handling a load that often rose 40 per cent. above normal, one of the firemen suddenly became incapacitated, and a coalpasser was substituted. The engineer was obliged to visit the boiler room frequently to direct the new stoker. It was on one of these occasions, when the engineer had been delayed rather longer than usual, that a breathless

helper reported that the oiler was having trouble.

Hurrying to the engine room, the oiler reported he had tried to operate the emergency valve as the turbine was running away, but it would not move. An examination showed that the latch-lever had released the weight and only the fact that the packing of the valve stem had been tightened that morning had saved a shut down.

Just then the telephone bell rang, and the superintendent inquired if all were asleep and what had just happened to the voltage, as the recorder registered way above normal. The turbine then had dangerously overspeeded and the packing around the emergency valve stem was immediately slackened.

Ascending to the top platform the engineer reached for the oil can with which to oil the governor rod joints, but it was not there. Looking around, he found it laying on its side on the platform. Evidently the slight vibration of the turbine had caused it to fall from the bracket; but what made it roll so far to one side?



How the Oil Can Blocked the Governor

Then it was discovered that the oil can had fallen from the motor bracket, where it was usually kept, landing near the upright reach-rod; and the snout had caught under one arm of the governor-rod bell crank when the turbine was under a heavy load, as shown in the illustration, and had held the governor inoperative until the speed gave it sufficient power to spring the spout, and bounce the can to where it was found.

R. O. Richards.

Framingham, Mass.

Expensive Engine Repair

These indicator diagrams were taken from a 12x24-in. gridiron valve-gear engine, and the distortions are due entirely to the action of the governor, after it had been broken and improperly repaired.

This engine was installed in a woolen mill, and after running satisfactorily for over a year, the governor was accidentally broken. A machinist, called in to estimate on the repair, said it would cost about $20 to get the engine started again. The foreman (a skilled woolen worker) thought this sum too large so called in the mill blacksmith.

After "fixing" the governor, the engine refused to run. A bright thought struck the foreman. "Get one of them things that tells what's the matter with engines," said he. So a man and an indicator were sent for. After repeated trials the engine still worked badly. The entire load was then taken off the engine, when Fig. 1 was obtained.

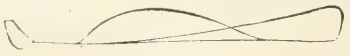


FIG. 1

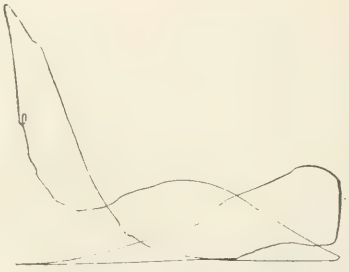


FIG. 2

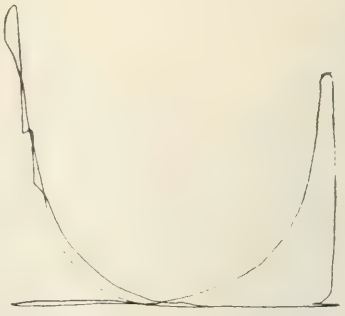


FIG. 3

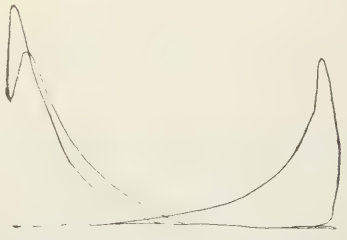


FIG. 4

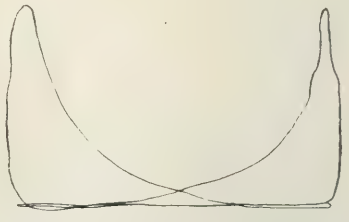


FIG. 5

Diagrams Taken during Adjustments to Valves and Governor

The indicator man then gave it up, and a machinist was sent for. After juggling with the governor and valves, the diagrams Figs. 2, 3 and 4 were obtained. The machinist then started an investigation, and located the trouble in the blacksmith's fixing, and later Fig. 5 was taken. The valves were then reset to their original position and good diagrams obtained.

The cost of the job was as follows: Blacksmith's time, 16 hr. at 32c., $5.12; helper's time, 16 hr. at 22c., $3.52; man and indicator, $10; machinist (shop charge), $20; total cost of the job, $38.64.

J. W. Thayer.

Newark, N. J.

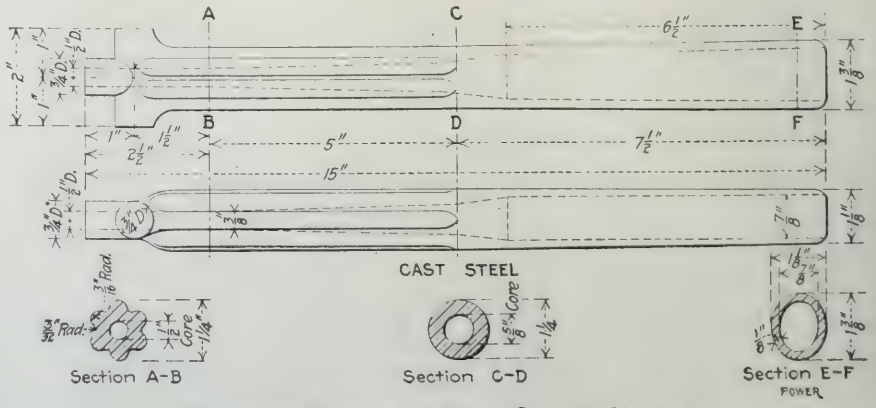
Strong Handle for Babbitt Hammers

The illustration shows a design for a durable handle for heavy babbitt hammers. When made of cast steel it gives long service, because of its strength. New hammers can be molded on the handles as required. The grip is cored out thin and shaped like the usual wooden handle for all kinds of hammers; therefore giving the same advantages in handling. The handle walls gradually grow thicker near the hammer and are ribbed, thus giving great strength where it is needed most. The first cost may be greater than for the same article in gas pipe or tubing, but the advantage to the workmen, and the long life of the handle more than make up for the additional cost.

When made according to the drawing the hammer will be of a good size for engine-room work, such as driving in keys, pins, etc.

W. W. Wysong.

La Porte, Ind.



Details of Handle for Heavy Babbitt Hammer

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Another Old Steam Plant

As a supplement to my article in the July 23 issue I would state that another very old steam plant still in operation has been brought to my notice, particulars of which may interest the readers.

This installation is at the Shipley Colliery Co.'s pits near Derby, England, and consists of a Cornish pumping engine of unknown origin. Authentic records show, however, that it has been in its present location and in satisfactory operation for over 100 years.

The principal dimensions and data are as follows:

Steam cylinder diameter (rebored 23 years ago)	52½ in
Stroke	7 ft.
Weight of beam	9 tons 1 cwt.
Two pump buckets, diameter	14½ in.
Water lift	195 ft.
Speed, strokes per minute	8 to 9 ft.
Duty	44,000 gal. per hr.

In addition to the duty of pit pumping, this engine also draws water from a pond half a mile away, delivering it into a reservoir for boiler feeding. A 10-in. ram pump is driven by the engine beam for this purpose.

The steam supply is the exhaust from a 12x24-in. and a 6x12-in. engine, which take steam at 70 lb. and exhaust against a back pressure of about 5 lb. into an old boiler which is used as a receiver. Steam for the ram pump is normally supplied by the exhaust from these two engines, but when this supply is insufficient, a makeup for the receiver is provided from the boilers through a reducing valve. A safety valve on the receiver is set to open at 8 lb. The vacuum is maintained by a jet condenser and averages about 25 in.

The maintenance costs for this engine are on a par with its steam consumption —low. The engine is said to use, per week:

1 gal. of cylinder oil.
1 qt. of machinery oil.
2 lb. of waste.

Bucket leathers last about three months and cost 14 shillings each. The labor cost for changing them is 15 shillings. Clack leathers last six months and cost 7 shillings each.

Such installations as this do indeed bear out the statement in the editorial of the July 23 issue, that the designers of these old engines "built better than they knew."

JOHN S. LEESE.

Manchester, Eng.

Burning Slack with Forced Draft

Reading V. L. Bowle's letter, "Replacing Stokers with Shaking Grates," prompts me to write of the change we made in our furnaces recently. We have been using the ordinary flat-surface herring-bone grates with natural draft for years. Our plant contains five horizontal tubular boilers, and we have been burning an average of 48 tons of Kentucky mine-run coal every 24 hr. Our evaporation by actual test showed 6.5 lb. of water per pound of coal; feed-water temperature 150 deg. The pressure carried is 100 lb. We recently installed a full equipment of slack burners using forced draft and now we evaporate 7.5 lb. of water per pound of slack. These burners are giving perfect satisfaction and we realize a saving of $25 per day.

J. W. DICKSON.

Memphis, Tenn.

Boiler Explosions

In the Oct. 8 issue, page 540, Lloyd V. Beets, in his letter on boiler explosions, suggests that boiler shells be drawn the same as tubes. This was discussed in POWER some nine or ten years ago by the late J. H. Dunbar, under the heading, "Comments on Power." At that time I believe the subject was called up by an explosion at Canton, Ohio. Several readers agreed that boiler shells would be better if lap welded the same as large size pipe. The trouble was that then no manufacturer made plates of the dimensions necessary to make a boiler shell of say 60 in. in diameter or larger, by the lap-weld process.

It is my opinion, borne out by considerable experience, that boilers made either lap welded or cold drawn, would be prohibitive in cost. If boiler shells were lap welded, they would not be as good as boilers made in the present way, as the metal would not be of a uniform thickness in all parts of the shell. Boilers of the horizontal return-tubular type have considerable stiffness added, due to the lapping of the joints. If made by the lap-weld or drawn process they would lack this feature, and would have a tendency to sag and become distorted when placed in the settings. Then there would be the difficulties encountered in trying to keep the shell round. In time, boiler shells, particularly the drums of water-tube boilers, will probably be welded by electricity or gas, similar to the welding of steam pipes as at present practiced in modern power-house piping, wherein screwed pipe fittings are eliminated on the larger sizes of pipes.

J. F. NAGLE.

Troy, N. Y.

Recollections of Corliss

Mr. Low's article on the Corliss engine in the Sept. 10 issue recalls anecdotes I heard about George Corliss when I was a boy living in Providence.

Mr. Corliss was known in his home town as a very shrewd business man as well as an inventive genius. He built the great Centennial engine and had it placed in the center of Machinery Hall, where it was seen and admired by everybody. Incidentally he made the exhibit a paying investment by taking contracts for furnishing power with it.

Most mechanical geniuses are poor business men, but Mr. Corliss always knew how to make a bargain, and had a very persuasive way of bringing people around to his way of thinking. I was an eye witness of the following incident:

In a large machine shop in Providence where machine tools are made, there was an engine built by another Providence builder. At the time the engine was put in, it was a single engine, but arranged so that another cylinder could be added if the works grew so as to need more power. Mr. Corliss called at this shop one day, to see about buying some tools for his shop. The proprietor took him all over the place, and into the engine room. At that time they were thinking of installing another cylinder, but first proposed to put on a wider wheel to take another belt. It was intended to have the original builder make the new wheel.

The proprietor told Mr. Corliss all about his plans, and they continued to talk until Mr. Corliss had brought the conversation around so there was hardly anything the man could say, but, "Well, Mr. Corliss, you can go ahead and make the new wheel," and incidentally, of course, he took a good order for tools. That engine is running today with a Corliss wheel, but all the other parts are of another make.

W. E. HOPKINS.

Torrington, Conn.

Improper Lubrication as the Cause of Air Receiver Explosions

Soapsuds for air-compressor lubrication have been much spoken of and written about, and Martin McGerry's soapsuds lubricator, as described in the Nov. 26 issue, being in the same issue with Letson Balliet's description of the explosion of an air receiver, on page 810, will make pertinent a few remarks on this subject.

Soapsuds are used for the very purpose of preventing just such an explosion as that described, but many operating engineers, and probably all air compressor builders do not recommend it. It is doubtful if there is much lubricating value in soap, and the effect of the alkali and the water in which the soap is dissolved is to rust the cylinder and its working parts. In Mr. McGerry's case, the use of oil for half an hour before shutting down may be the saving grace that not only prevents serious corrosion but furnishes enough real lubricant, even at long intervals, to prevent cutting.

No doubt, too, the water does much to prevent cutting, and it might be interesting to run the compressor with water only, leaving out the soap but still finishing off with oil, as before, to see how "slippery" such a combination would be. Many steam engines in the navy are often run without any cylinder oil.

These explosions of air receivers are altogether too frequent, and every operating engineer and owner should try to learn how to avoid them. Many who have studied the subject believe that the explosion is made possible by the presence of a mixture of oil and air, or perhaps finely divided carbon and air. Most, if not all, of these explosions occur in the receiver, and not in the compressor. This is peculiar when we realize that the air in the receiver must be cooler than at the instant of discharge from the air cylinder.

A single-stage air compressor, pumping to 100-lb. gage pressure delivers air at a temperature not much below 400 deg. F. In a small machine the relatively large jacketed wall surface cools this down almost instantly, so it is difficult to find such a temperature beyond the discharge valves. A compound compressor, at high-speed pumping to 100 lb. will deliver air in the pipe line at a temperature of not far from 275 to 300 deg. F.

These are not excessive temperatures for oil, except perhaps in the presence of the highly compressed oxygen in the air. Oils are made specially for air-compressor cylinders and only these should be used; yet many, either through ignorance or unwillingness to spend money on the proper material, are using entirely unsuitable oils.

Even with these special oils, however, explosions have occurred, probably due to an excess of air pressure and a very high temperature, or more probably to an excess of oil. Oil should be fed to an air cylinder as sparingly as will insure perfect running—and no more. An excess will produce a deposit of carbon on the valves, springs, ports, pipes and receivers, and may lead to a fatal explosion.

From the different actions observed, it would seem that these ignitions of deposited oil and carbon are of at least two kinds. Air receivers have been known suddenly to become hot to the glowing point, with the beginnings of "bagging" of the red-hot steel. Also, air pressures have been known suddenly to increase to double their original value, and then more gradually to return to normal, without a rupture, though the piping and receivers have been intensely hot. Then again, as in the case described, the ignition has been of the sudden or explosion type, with rupture and perhaps fatal results. In the first mentioned cases, it would seem that the burning had been slower, so that the increase of pressure was not so rapid but that the driven machinery could absorb it by increased speed, or the volume and area of the pipe and tank system were sufficiently large to relieve it fast enough to prevent rupture.

Stage compression is necessary for safety, when compressing air in large- or medium-sized cylinders to pressures above 100-lb. gage. The intercooling not only saves power, with proper design, but also reduces the final temperature and consequently the danger of ignition. With small cylinders, say 3x3 in., several hundred pounds have been reached in a single stage, but it is probable that little or no oil was used and the jacketing and radiation were sufficient to keep down the temperature of so small a quantity of air.

What sets off these explosions is hard to determine. Surely there cannot be any red-hot carbon on the springs, etc., because 400 deg. F. is not a red heat, to say nothing of the more usual figure of 275 deg. It must be started by a slow oxidation, perhaps like the spontaneous ignition in stored coal. If this were the case the explosion might be preceded by excessive temperature in the receiver.

This would seem to be borne out by the fact that the explosions nearly always take place in the receiver, where the air velocity has very materially decreased, rather than in the ports and pipe lines, where the higher velocity might not allow this gradual oxidation to take place. Unfortunately (or fortunately) no one is known to have observed this preëxplosion temperature.

At any rate we know these explosions do occur and probably from ignition of oil or its constituents, and so far the remedies, or rather the preventives, are:

Use only the right oil, made for the purpose, and use little of it; frequently remove all deposit from the springs, valves, etc., but do not use gasoline nor kerosene, or any other fuel oil with which to remove such deposits. Conditions will indicate how often this cleaning should be done; certainly once a month at the longest. Also drain the air receivers often.

H. B. Sidney.

New Orleans, La.

Test of Axial Flow Centrifugal Fan

In regard to the "Test of Axial Flow Centrifugal Fan," by Bratherick in the Oct. 22 issue, I was interested to note that he had discovered a fan that worked with 10 per cent. greater efficiency as a blower than it did as an exhauster. This came as a surprise and it seemed, from my experience in testing fans, that the mere reference of pressures and velocities to the fan proper would obviate any such difference in efficiency. A little computation brought this fact out and there may be readers who would be interested to follow these computations.

I noticed that the ratio of the air velocity to the rim velocity of the wheel bears a constant ratio of 46.6 per cent. This corresponds to a ratio of static pressure to rim-velocity pressure of 27 per cent., taken from a characteristic on a similar fan. This static pressure is due to the restricting of the discharge area of 71 per cent. approximately of the inlet-duct area (it is assumed that the area through the wheel casing is equal to the inlet-duct area). Knowing the rim-velocity pressure, the static pressure is obtained as per tabulation below:

Static Pressure	Air Velocity Pressure	Dynamic Pressure	Air Hp.	B. Hp.	Impact Mech. Efficiency
0.126	1/8	0.251	0.0286	0.057	0..0
0.172	3/16	0.369	0.0516	0.084	0.612
0.247	1/4	0.497	0.08	0.136	0.588
0.312	5/16	0.625	0.112	0.188	0.597

From the dynamic pressure and the volumes it will be seen that the air horsepower corresponding will be as given, which, with the input horsepower, gives the efficiencies shown in the table.

Therefore, these correspond almost exactly with the mechanical efficiency of the fan as a blower. This proves conclusively, at least to me, that the efficiency of a fan is no different when run as an exhauster than when run as a blower.

H. H. Valiquet.

Hyde Park, Mass.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Actual Diamensions of 8-in. Pipe

What are the actual inside diameter, cross-sectional area and cubical content of commercial 8-in. wrought-iron and steel pipe?

W. H. B.

Most American manufacturers quote the actual internal diameter of nominal 8-in. pipe as 7.981 in., making the internal cross-sectional area 50.027 sq.in., or 0.3474 sq.ft. One foot in length of nominal 8-in. pipe, therefore, contains 0.3474 cu.ft. or 2.5987 U. S. gal. (231 cu.in. per gal.), and 2.8785 lineal feet contain 1 cu.ft.

Rate of Combustion

A boiler evaporates 3211 lb. of water per hour. The grate is 6½ ft. by 4 ft. and 9½ lb. of water are evaporated per pound of coal. What is the rate of combustion?

R. M.

As 3211 lb. of water are evaporated per hour and 9½ lb. of water are evaporated per pound of coal the amount used would be

$$\frac{3211}{9.5} = 338 \text{ lb. of coal per hour}$$

The grate being 6½x4 ft. would have an area of 26 sq.ft., therefore the rate of combustion would be $\frac{338}{26} = 13$ lb. of coal per hour per square foot of grate.

Safety Valve on Safety Valve

If the safety valve on a boiler is set to blow off at 100 lb., and then has its discharge connected into another safety valve set to blow off at 100 lb. with the discharge to the atmosphere, what pressure of the boiler will it take for the valves to blow off?

A. J. L.

Each safety valve being set to blow off for 100-lb. pressure per sq.in. more on one side than on the other side of the valve, then to lift the safety valve which discharges to the atmosphere, the pressure discharging from the valve next to the boiler must be 100 lb.; and as the pressure required to lift the valve next to the boiler must be 100 lb. more than the pressure on the discharge side of that valve, there would have to be 200 lb. pressure in the boiler to lift both safety valves.

To Remove Grease from a Boiler

How can deposits of grease be removed from a boiler?

R. R. D.

Grease can generally be cut and washed out by using soda ash, about $\frac{1}{10}$ lb. to a horsepower of the boiler for each application. With the boiler pressure down to zero, soda ash when pulverized can be introduced through the safety valve or other opening. Then with the boiler filled above the highest water level usually carried, the water should be boiled under a pressure of 5 to 10 lb. for about four hours, after which the boiler should be blown off and washed out with clean water. If the oil is coated or gummed to a considerable thickness, two or three such applications of soda ash may be necessary for thoroughly removing all traces of oil.

Danger of Ice Machine Racing

In a refrigerating plant using a Corliss engine type of ammonia compressor with belted governor, would there be danger of the compressor wrecking itself from excessive speed if the governor belt should break or run off at a time when the governor collar had been misplaced?

L. A. D.

If the governor became inoperative a dangerously high speed might be attained before sufficient compression could occur to check the speed. Excessive speed would occur in the event of a release of pressure due to a rupture from any cause, such as an accumulation of excessive compression of ammonia gas.

Degree of Superheating

What must be the degree of superheating of steam at 180-lb. gage pressure in order to show 50 deg. superheat after being throttled to 165-lb. gage pressure?

E. S. F.

There will be some loss of heat by radiation during throttling, but neglecting this loss, the total heat per pound of steam would be the same before and after throttling. Refering to Marks and Davis' steam tables, the total heat contained in one pound of steam raised from 32 deg. F. to 180 lb. absolute (which is practically the same as 165-lb. gage pressure), with 50-deg. superheat is 1227.2 B.t.u. and from inspection of the same tables it may be seen that for a pressure of 195 lb. absolute (which is practically equal to 180 lb. gage pressure), the same number of heat units would be contained by a pound of steam at the latter pressure for some degree of superheating between 40 and 50 deg. superheat, the number of B.t.u. for 40 deg. being 1223.3 and for 50 deg., 1229.2. As the number of B.t.u. for 40 deg. superheat is

$$(1227.2 - 1223.3) = 3.9 \text{ B.t.u.}$$

too small and as 1 deg. superheat at the pressure of 195 lb. absolute, 40 to 50 deg. superheat requires about

$$\frac{1229.2 - 1223.3}{10} = 0.59 \text{ B.t.u.}$$

per degree, then

$$\left(\frac{3.9}{0.59}\right) + 40 = 46.6$$

would be about the superheat required of the steam at 180-lb. gage pressure to be equal to 165-lb. gage pressure and 50 deg. superheat.

Horsepower for Feeding a Boiler

Neglecting losses from friction, what number of horsepower would be required for feeding 210 gal. of water per hour to a boiler carrying 86 lb. pressure per sq.in. and using a 4½x3x6-in. boiler-feed pump, the water being supplied to the pump at atmospheric pressure?

A. H.

The useful work would be the same using any size of pump. Assuming for present purposes that a pump is used having a water piston of 1 sq.in. area and making a stroke of 1 ft., then at each stroke there would be a displacement of 12 cu.in. of water, and pumping against 86 lb. boiler pressure each stroke being 1 ft. in length would require 86 ft.-lb. Therefore each cubic inch of water fed would require

$$\frac{86}{12} = 7.166 \text{ ft.-lb.}$$

Feeding 210 gal. of water per hour would be equivalent to $210 \times \frac{231}{60} = 808.5$ cu.in. per min., requiring $808.5 \times 7.166 = 5793.71$ ft.-lb. per min, or

$$\frac{5793.71}{33{,}000} = 0.175 \text{ hp.}$$

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Involution and Evolution

These subjects are not so difficult to master as their names would seem to indicate.

Involution and evolution deal with the *powers* of numbers. A power of a number is the product obtained by multiplying that number by itself one or more times. For instance, 4 is the *second* power of 2, for

$$2 \times 2 = 4$$

27 is the *third* power of 3, for

$$3 \times 3 \times 3 = 27$$

256 is the *fourth power* of 4, for

$$4 \times 4 \times 4 \times 4 = 256$$

Thus we see that involution is the process of raising a quantity or number to any assigned power.

To let it be known to what power a number is to be raised we use an *exponent.* The *exponent* is the number that denotes how many times the number is to be taken as a factor, or to what power the number is to be raised. The exponent is written above and to the right of the number, as

$$4^2 = 4 \times 4 = 16$$

$$4^4 = 4 \times 4 \times 4 \times 4 = 256$$

The reader should notice that the exponent does not tell directly how many times the multiplication is to be performed (which is always one time less than the exponent) but how many times the factor itself must be taken. Thus 2^2 does not mean to multiply 2 by itself twice, but to multiply two 2's together.

From the expression 4^4 we see that by using the exponents we can express with two figures what would otherwise require four 4's and three multiplication signs. We also see that a product of equal factors is a power.

Powers of the same quantity may be multiplied by adding their exponents as,

$$4^2 \times 4^3 = 4^5$$

i.e., the product of 4^2 and 4^3 is 4^5, for

$$4^2 = 4 \times 4 = 16$$

and

$$4^3 = 4 \times 4 \times 4 = 64$$

Then

$$4^2 \times 4^3 = 16 \times 64 = 1024$$

which is the same as

$$4^5 = 4 \times 4 \times 4 \times 4 \times 4 = 1024$$

On the other hand, powers of the same quantity may be divided by subtracting the exponent of the divisor from the exponent of the dividend, as

$$\frac{4^5}{4^2} = 4^3$$

for $4^5 = 1024$ and $4^2 = 16$, and

$$\frac{1024}{16} = 64$$

which is the same as

$$4^3 = 4 \times 4 \times 4 = 64$$

Any quantity having the exponent 1 is equal only to itself; it shows that the factor is considered but once. But if the factor is already there we do not need an exponent because the exponent 1 does not change its value; hence, all quantities of the first power have no exponent.

The second power of a quantity is called a *square.*

It is so called because the second power of a number represents the *area of a square* of which that number repsents one side. Take a square foot; each side is 12 in. long and we may imagine the area divided off into square inches, as shown in Fig. 1. Here each row represents 12 sq.in. and as there are

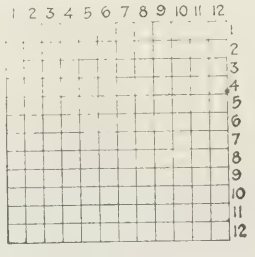


Fig. 1

12 rows the total number of square inches would be

$$12 \times 12 \text{ or } 12^2 = 144$$

For the same reason the third power of a number is called a *cube,* the volume of a cube being indicated by the third power of the number which indicates the length of its side. As an example, take a cubic foot, Fig. 2, the edges measure 12 in. each and each face contains 144 sq.in., as we have seen in our example in Fig. 1. If the cubic foot is divided into blocks of 1 cu.in. each, as in Fig. 2, it will be seen that the cube consists of 12 layers or slices and that there are 144 blocks or cubic inches to each layer or slice. Now the total number of blocks or cubic inches will be 12^3 or

$$12 \times 12 \times 12 = 1728$$

When a minus sign appears before or in front of an exponent it means that the number is to be raised to the power indicated by the exponent and written as the denominator of a fraction the numerator of which is 1. Thus

$$4^{-1} = \frac{1}{4^1} = \frac{1}{4}$$

$$4^{-2} = \frac{1}{4^2} = \frac{1}{16}$$

$$4^{-3} = \frac{1}{4^3} = \frac{1}{64}$$

To raise a common fraction to any power it is necessary to raise both of its terms or the numerator and denominator to the required power. Thus the cube of $\frac{2}{5}$ or

$$\left(\frac{2}{5}\right)^3 = \frac{2 \times 2 \times 2}{5 \times 5 \times 5} = \frac{8}{125}$$

Mixed numbers should be reduced to improper fractions and then both terms should be raised to the required power, the same as with proper fractions.

Example: What is the square of $24\frac{3}{4}$?

$$24\frac{3}{4} = \frac{99}{4} \text{ and } \frac{99 \times 99}{4 \times 4} = \frac{9801}{16} = 612\frac{9}{16}$$

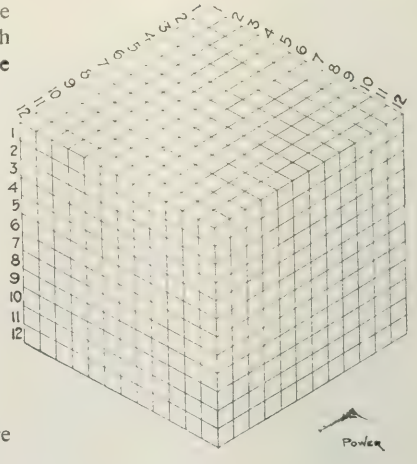


Fig. 2

The above $612\frac{9}{16}$ is the correct answer to $24\frac{3}{4}$ squared. Now suppose we had squared the whole number 24, and the fraction $\frac{3}{4}$ separately and then added the products, we would have obtained an incorrect answer for,

$$24^2, \text{ or } 24 \times 24 = 576$$

and $(\frac{3}{4})^2$ or

$$\frac{3}{4} \times \frac{3}{4} = \frac{9}{16}$$

then

$$576 + \frac{9}{16} = 576\frac{9}{16}$$

an incorrect answer.

This shows clearly that the sum of these two quantities squared is not the same as the square of the whole mixed quantity multiplied by itself.

The reason for this is quickly seen by squaring the number when expressed as a decimal, 24.75, for in multiplying this by itself it will be seen that the fractional part of the multiplier multiplies also the whole number part of the multiplicand and so increases the product over

what it would be if the mixed number were treated in the incorrect way as above.

Evolution

The factor repeated to produce a power is called a *root*, thus,

Four is the square of 2; then 2 is the square root of 4. Also 9 is the cube of 3; then 3 is the cube root of 9.

Evolution is the process of finding the root of a power, and is the reverse or opposite of involution, which is the process of finding a power from a root.

The radical sign $\sqrt{\ }$ placed before a number indicates that the root of that number is to be *extracted*. When the sign alone is used before a number it indicates that the *square root* is to be extracted. For other roots an index, which is a small figure, is placed *above* and to the left of the radical sign to indicate what root is to be extracted.

Thus

$\sqrt{9}$ indicates the *square root* of 9.

$\sqrt[3]{9}$ indicates the *cube root* of 9.

$\sqrt[4]{282}$ indicates the *fourth root* of 282.

The term under the sign, taken with the sign is known as a *radical* and may also be expressed by using a fractional exponent over the number as $4^{\frac{1}{2}}$, which is the same as $\sqrt{4}$. When a fractional exponent has a numerator other than 1, as $4^{\frac{3}{2}}$, it indicates that the number is to be raised to a power as well as having a root extracted. Thus in the example cited, $4^{\frac{3}{2}}$, the 4 is to be raised to its third power and the square root of the power is to be extracted.

$$4^3 = 64$$

and the square root of 64 or

$$\sqrt{64} = 8$$

so that $4^{\frac{3}{2}} = 8$. The operations may be done in either order and the answer will be the same for

$$\sqrt{4} = 2$$

and

$$2^3 = 8$$

Usually, however, it is simpler to raise to the power first.

Examples for Practice

(1) What is the square of 143; 16.84; $\frac{2}{3}$; $18\frac{3}{4}$?

(2) What is the cube of 14; 84.3; $\frac{5}{8}$; $24\frac{1}{8}$?

(3) (a) When the radical sign alone is used before a number, what is meant? (b) What does the index figure mean?

(4) What number is indicated by 24^2; 48^3; 18^4; 8^{-3}?

(5) What do you understand by $43^{\frac{3}{2}}$; $16^{\frac{1}{4}}$?

(6) How many square inches are there in a piece of sheet iron 18 in. square? How many square feet in a wall 14 ft. high and 22.5 ft. long.

(7) How many cubic inches in a cube that is 9 in. on an edge

Answers to Last Week's Problems

(1) 33¾.
(2) 17[illegible].
(3) $85.15.
(4) 6 in.
(5) 11.42 hr.
(6) 165.6 lb.
(7) 6 in.
(8) 120 lb.
(9) 131,625 gal.

A Convenient Safety Valve Chart

By Amos J. Carr

The Massachusetts formula for the area of a safety valve reads:

$$A = \frac{W \quad 70}{P \quad 11}$$

where

A = Area of safety valve in square inches per square foot of grate surface;

W = Pounds of water evaporated per second per square foot of grate surface;

P = Absolute pressure at which steam is generated.

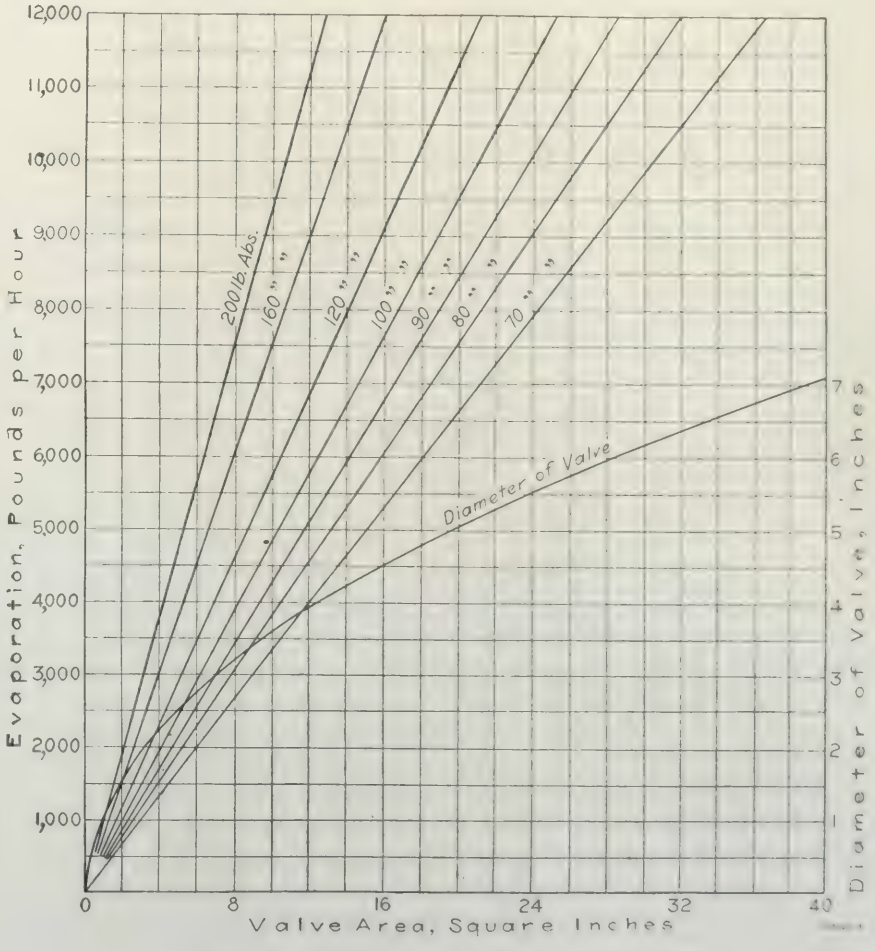
Anyone familiar with Napier's rule for the flow of steam through an orifice will readily recognize the quantity $\frac{W \quad 70}{P}$ which is obtained by transposing, to solve for A, Napier's approximate formula

$$W = \frac{A \quad P}{70}$$

Thus with the exception of the 11, the Massachusetts formula is identical.

It is known that $\frac{W \quad 70}{P}$ will give the necessary opening to prevent an accumulation of pressure, but when it is considered that the effective opening of a safety valve is influenced by several factors and that the lift is directly proportional to the diameter or circumference, and does not bear any definite relation to the area of the valve, it is not quite clear how the factor 11 fits in. The following problem is solved on the assumption that the factor 11 has been so chosen as to give the desired results with a predetermined lift corresponding to a given diameter.

Take an evaporation of 5500 lb. of water per hr., or 1.527 lb. per sec. at an absolute pressure of 160 lb. per sq.in.



Safety-valve Chart for Different Pressures

Substituting the above values in Napier's rule and solving for A gives

$$A = \frac{1.527 \times 70}{160} = 0.668 \ sq.in.$$

A 3-in. valve having a lift of 0.10 in.

gives a ring area of 0.94 sq.in., but with a seat of 45 deg. bevel the ring area must be multiplied by the cosine of 45 deg., or 0.71, which gives

$$0.94 \times 0.71 = 0.667 \text{ sq.in.}$$

as the effective opening. This compares closely with the 0.668 sq.in. obtained by Napier's rule. The Massachusetts formula or the accompanying chart, assuming some grate area, will give the same result, so the factor 11 answers in this case.

A 2-in. valve having the same lift as the 3-in. valve in the above case would have a ring area of 0.63 sq.in. and an effective opening of

$$0.63 \times 0.71 = 0.45 \text{ sq.in.}$$

From the chart it may be seen that this size of valve at 160 lb. steam pressure would be chosen for an evaporation of 2000 lb. of water per hour, or 0.555 lb. per sec. Napier's rule for a flow of 0.555 lb. of steam at an absolute pressure of 160 lb. per sq.in. gives an area of 0.24 sq.in., which is about one-half of the value computed above, assuming a 0.10-in. lift. A 2-in. valve must have a lift of 0.05 in. at the assumed pressure in order that the factor 11 shall suit this case.

Assuming the springs to have uniform compression in the two cases, the 0.05-in. lift with the 2-in. valve would be consistant, because the lift of the valve is proportional to the area multiplied by the pressure. The area of the 3-in. valve is 7.06 sq.in., and of the 2-in. valve a little less than half this figure.

Multiplying the 0.24 sq.in. obtained by Napier's rule by 11 gives an area of 2.64 sq.in., which would require a valve having a diameter of 1.75 in., the same as would have been obtained had the Massachusetts formula been used. A 2-in. valve would be required as there is no size between 1.5 and 2 in.

Construction and Use of Chart

As the evaporation of boilers is expressed in pounds of water per hour and W in the Massachusetts formula represents pounds of water evaporated per second per square foot of grate surface, the value of the factor W as used in an actual case would be:

$$\frac{W_{hr}}{3600 \times N}$$

where

W_{hr} = Pounds of water evaporated per hour;

N = Square feet of grate surface.

By letting A_t equal the total valve area, the formula becomes:

$$A_t = \frac{\frac{W_{hr}}{3600 \times N} \times 70 \times N}{P} \times 11$$

which may be simplified to

$$A_t = \frac{W_{hr} \times 0.21388}{P}$$

Assuming an evaporation of 10,000 lb. per hr., it is only necessary to move the decimal point of the constant 0.21388 four places to the right and divide by 160. In this case an area of 13.4 sq.in. would be required. From the bottom of the chart move upward on a vertical line corresponding to this area until the line opposite 10,000 is reached. Through this point and the zero point of evaporation and areas, draw a straight line and the area of the valve can be quickly determined for any evaporation at the above pressure.

By computing several diameters corresponding to chosen areas, points were established on the chart and through these the diameter curve was drawn. For example, a valve of 24 sq.in. area has a diameter of 5.53 in., which is the value given in the chart.

Should it be desired to know the size of valve for an evaporation beyond the limit of the chart it is only necessary to determine the area of the valve for one-half of the evaporation in question, double it and determine the diameter.

Take an evaporation of 24,000 lb. per hr. at 160 lb. abs. For one-half of this evaporation, 12,000 lb., a valve area of 16 sq.in. would be required; for 24,000 lb. the area would be 32 sq.in. An area of 32 sq.in. corresponds to a diameter over 6 in., so it will be necessary to go back to the 16-in. area, and a valve diameter of 4.5 in., of which two would be used. From the foregoing it is evident that the chart has a wide range of application.

Suppose a plant is being operated with a steam pressure of 120 lb. abs. and 10,000 lb. of water per hr. is being evaporated. An inspector reduces the working pressure to 100 lb. abs. The load demand being unchanged a higher evaporation would be required, this depending upon the change in engine speed, etc., to deliver the same power at a reduced mean effective pressure.

In the first case a 5-in. valve would be required and from the chart it can be seen that there is considerable margin, in that 11,000 lb. could be handled by the 5-in. valve at 160 lb. pressure, whereas assuming that 11,000 lb. must be evaporated at the reduced pressure, a 5.5-in. valve is necessary. Of course the inspector would decide this point, but the chief should be equal to the occasion and know the proper procedure.

State Fire Marshal Ahearn, of New York, reported recently that inspectors of his department had completed examination of every boiler in the state, to aid in strict enforcement of the law relating to the location of boilers in factories. He said the reports showed that a large majority of the boilers were safe. The work supplemented the efforts of the factory investigation commission.

Motor Driven Theater Fire Pump

The fire department of the city of New York in recognition of the necessity of adequate auxiliary fire-protective apparatus in theaters, requires the installation of a fire pump in each theater for use in connection with the regular city service in case of a fire.

A good example of the execution of this requirement is the installation at the Maxine Elliott Theater. The equipment consists of a Deming triplex 8x8-in. fire pump driven by a Westinghouse 20-hp. 220-volt direct-current motor.

The pump has a speed of 50 r.p.m., obtained through a double reduction gearing from the 1300 r.p.m. of the motor, and will deliver at this speed a 1¼-in. stream of water. The capacity of the pump is 250 gal. per min., at a pressure of 100 lb. As the plungers are bronze and the stuffing-boxes are brass lined, rusting cannot take place and the pump is therefore always ready for immediate operation.

The motor is controlled by an automatic starter which is connected through a pressure regulator to the hose reels so that when the hose is removed the pump will start automatically. This pump outfit came from the R. B. Carter Co., 152 Chambers St., New York City, agent for the pump.

For a National Museum

Interest has been revived in the plan to establish a national museum where the developments of engineering and the arts might be preserved. In the original design a large room in the Engineering Societies Building, New York City, was set apart for this purpose, but a lack of sufficient funds has prevented its accomplishment.

The recent visit of the German museum commission of Munich had much to do in interesting the members of the American Society of Mechanical Engineers in this project. The council of this society recently appointed a committee to coöperate with other organizations in establishing a great national museum. The committee is composed of E. D. Meier, H. G. Reist, Ambrose Swasey, George Mesta and George F. Kunz.

The Yarnell-Waring Co., Chestnut Hill, Philadelphia, has assumed management of the factory and sales of the Nelson Valve Co. The identities of both companies will be maintained and the management and officers of the Yarnall-Waring Co. will remain unchanged. D. Robert Yarnall will be vice-president and general manager and Bernard G. Waring, vice-president and manager of sales of the Nelson Valve Co.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

A Merry Christmas to you, and yours! Charles Dickens said it all in these words: "I have always thought of Christmas . . . as a kind, forgiving, charitable, pleasant time; the only time I know of, in the long calendar of the year, when men and women seem by one consent to open their shut-up hearts freely, and to think of people below them as if they really were fellow passengers to the grave, and not another race of creatures bound on other journeys."

TIM HEALY'S JINED TH' SPUGS

Oh, Paddy, dear, an' did ye hear
Th' news that's goin' around?
Tim Healy's jined th' Spugs, bedad.
Sure, he's beat ye to th' ground!

Th' Spugs, says Tim, aare organized
To sthop poor girls from givin'
Christmas prisints to th' boss;
Thus raise th' cost av livin'.

An' whin Society tuk hold
To aid thim girls so maany,
'Twas Tim who gave th' thing éclat,
Wid Morgan's daughter Annie.

Oh, Timmy, man, av ye we're proud,
It's graand worruk, d'ye see?
Ye have th' roight idee, me bhoy.
Here's to ye, Timothy!

Man complains that there's a 60 per cent. discrepancy in the given values of skin-effect coefficients. This is a some serious complaint. Why not consult a physician?

It is proposed that "national and state commissions be formed to fix wages." Just now, when the wife of our bosom is striving to lift the mortgage off the old homestead, and Edythe and Maybelle are knitting our annual light-blue Christmas suspenders, and Cuthbert and Egbert are scrapping over who gives the pipe and who the Ivanhoe package—well, no fixation process will help us any; we're "fixed" already.

Describing a boiler explosion. a correspondent writes that "while the engineer was unhurt, he thought the Lord loved him, so he was through with the engineering field." Be of good cheer, brother, and hang on; nobody loves a quitter.

"Hey, you! drop that monkey-wrench and let the engine run if it wants to."—*Gas Review*. A most undignified editorial utterance, neighbor, but your sound common sense tickles us under the fifth rib. Go it! This is the one column where we don't have to be dignified, thanks be!

Power Plant Costs

By V. C. Bowles

In a power plant of almost any type and size, where modern equipment for measuring performance is not available, at least some sort of record covering operating conditions can and should be kept. Such records will eventually lead to the installing of suitable apparatus for obtaining complete and accurate data. There are very few owners, or managers, but will gladly coöperate with the engineers in such work.

The accompanying daily log and monthly chart show what can be done without the use of recording equipment.

The figures here given as examples are reasonably low, especially as the plant is anything but suited to the requirements of the service.

All data where any doubt may exist as to accuracy, such as volt and ammeter readings, air compressor revolution counter-readings, etc., are subject to a reduction of from 3 to 5 per cent. to eliminate all doubt.

The plant is that of the Clayton & Lambert Manufacturing Co., Detroit, Mich., and furnishes power, light and heat for manufacturing purposes, to eleven different buildings, covering about 8½ acres.

The power plant proper is in one of the main buildings on the outer edge of the group, and contains three vertical water-tube boilers of 100, 250 and 350 hp. respectively. The two larger boilers have stokers of the toboggan-grate type; the small one is hand fired. There are two boiler-feed pumps, one duplex 7x5x7-in. and one single 8x5x10-in., one 500-hp. open

POWER PLANT OF

CLAYTON & LAMBERT MFG. CO.

Date June 9th, 1911 — **DAILY REPORT SHEET**

		DAY	NIGHT	SUNDAY & HOLIDAY
1.	Average steam pressure	100		
2.	Average temperature feed water	204		
3.	Factor of evaporation.	1.043		
4	Cubic feet of water used by meter			
5	Total lbs. water evaporated.			
6	Lbs. water evaporated from and at 212°			
7.	" " " " per lb. coal fired			
8.	Average B. H. P. developed.			
9.	Lbs. coal per B. H. P. H.,			
10.	Lbs. coal burned.	12000	2800	
11.	" " " per square foot grate per hour.			
12.	" ash removed.	780	220	
13.	Per cent ash	6.5	7.8	
14.	Hours run No. 1 engine.	.5	1.5	
15.	" " " 2 "	10.5		
16.	" " " 3 "	11	3	
17.	K. W. H. delivered C. & L. switchboard.	1614	10	
18.	" " Regal "	116	15	
19.	Total K. W. H. generated.	1730	25	
20.	Hours run air compressor.	11		
21.	Cubic feet free air pumped.	55877		
22.	Number hours exhaust steam on heating system.	11	3	
23.	" " " live " " "			
24.	Height of water in sprinkler tank.	92ft.		

REMARKS

CHEMICAL FIRE EXTINGUISHERS. See that they are in their proper stations and filled ready for action. Report at once any broken, unfilled or misplaced Extinguishers and their location.

VALVES. Post valves in yards and alleys.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
O	O	O	O	O	O	O	O	O	O	O	O	O	O	O	O
S	S	S	S	S	S	S	S	S	S	S	S	S	S	S	S

VALVES—Main Drip and Dry Line. If any valve is found shut or with strap broken, state reason for it and report what sprinklers it controls.

FIRE DOORS. Each fire door is to be examined and all cases of broken or weak latches, rails or hinges reported. Report any obstructions found against the doors tending to prevent their quick operation.

FIRE PAILS. Report name of room in which pails are missing from their places, or found partially empty, dirty, or not readily accessible.

WATER PRESSURE today at yard level 40 lbs. from Office Building

Is tank or reservoir full? Yes

Has water been shut off from any part of the sprinkler system since last inspection? No

If so, state reason for it and length of time it was shut off.

Report any cases of leaky sprinkler pipes or heads which appear to have been injured, or covered by corrosion.

GENERAL ORDER. Report here any oily waste, or accumulation of rubbish, or waste material found in any room.

Ernest Beine, Inspector — V. L. Bowles, Chief Engineer

FIG. 1. FACSIMILIE OF DAILY REPORT SHEET

feed-water heater, one 5x6½x10-in. and one 5½x10x10-in. vacuum pump.

The generating equipment consists of one 125-kw. and one 250-kw. generator directly connected to single-valve engines, 16x15 in. and 22x20 in. respectively, and one 150-kw. machine directly connected to a 16x21-in. nonreleasing Corliss engine. There is also one 8 and 14 by 8 by 8-in. steam-driven air compressor.

The heating system contains about 60,000 sq.ft. of direct radiation on which a 17-in. vacuum is maintained. The demand for steam on the system from Oct. 15 to Apr. 15 is in excess of the exhaust steam available, and a pressure of 1½ lb. is required for various purposes, except nights and Sundays, the rest of the year. No charge is made, or credit given, for heat from Apr. 15 to Oct. 15, because no means are at hand for arriving at even an approximate value.

Kenward, W. Va., 1¼ nut, pea and slack coal at $2.30 per ton is used during the winter months. Repeated analyses have shown this coal to contain 2 per cent. moisture, 37 per cent. volatile, 57 per cent. fixed carbon, 3 per cent. ash and 1 per cent. sulphur and have a heat value of 14,400 B.t.u.

A saving of 7 per cent. was effected by purchasing this coal, as it proves better suited to the conditions during the winter, or heavy-load period. In the summer, or light-load period, a coking and slower burning coal gives better results.

No coal-handling apparatus is installed, due to the crowded condition of the plant; therefore all coal is handled by hand, and weighed on scales. On the coal sheet the steam pressure and feed-water temperature are also recorded hourly.

The amounts of water evaporation cannot be obtained, because of the absence of a meter, and the fact that the heating system is in operation the entire year, the returns being pumped direct to the heater. During the winter months the only raw water used is that which enters the condensers, as all steam, except that used for plating purposes is condensed and returned to the heater. The live steam used for plating is not taken into consideration in the accompanying figures, but is from 4 to 5 per cent. of the total evaporation.

Lacking watt meters, volt and ammeter readings are taken hourly to determine the electrical output. The load being comparatively steady, the results are reasonably accurate; 5 per cent., however, is deducted to be on the safe side.

The amount of air pumped is determined by using a revolution counter, and allowing 5 per cent. slippage. As near as the writer has been able to determine, 1000 cu.ft. of free air compressed to 90 lb. gage is equivalent to about 4 kw.-hr. and in reducing the work in pumping the air to an electrical equivalent, this figure is used. It is low, but is preferred to avoid exaggeration.

A draft gage is directly over the writer's desk in the engine room, so that by merely "plugging in" to a series of holes, the draft in either stack, or over any fire, can be seen at a glance, giving a very good idea of the condition of the fires at all times.

A daily report, or log sheet, showing the operating conditions of the plant is carried in duplicate, which is left on file in the engine room. On this sheet is also a record of conditions on the sprinkler system, and all fire risks. This is filled out by the man in charge of the heating system, as his rounds cover the entire establishment.

the heating system never closes from about Oct. 15 to Apr. 15, the coal curve following the temperature curve instead of the load curve.

Due largely to the influence of wind direction and velocity, the coal and temperature lines do not correspond as closely as might be expected. Any deviation not accounted for by atmospheric conditions, therefore, must be due to changed operating conditions, and is the occasion for immediate investigation. A record of wind direction and velocity would add greatly to the value of these charts.

The value of the temperature record is illustrated by the following comparison.

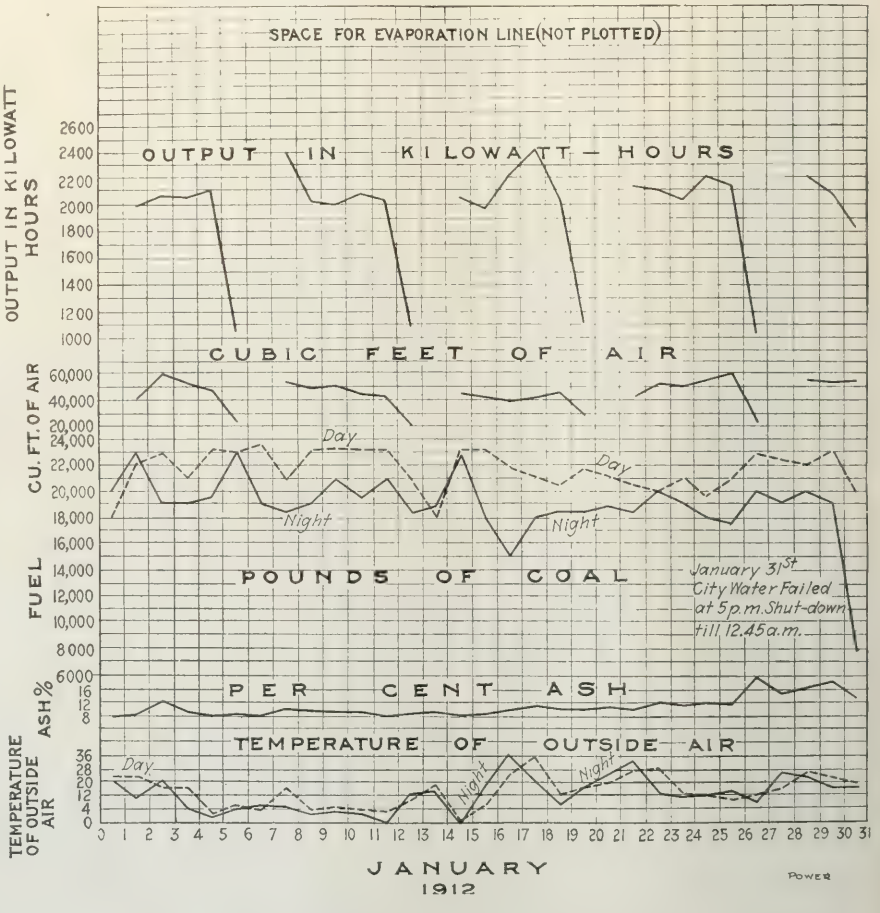


Fig. 2. Monthly Report in Curve Form

A series of monthly charts covering every condition determinable by the means available, is placed in a convenient and conspicuous place in the engine room, for the employees and others interested to observe. The curves on these charts indicate the output in kilowatt-hours, cubic feet of air pumped, pounds of coal burned, per cent. of ash, and a line showing the temperature of the outside air for each 12-hr. period. Provision has been made for an evaporation curve to be plotted when a suitable meter is installed, and a like provision has also been made on the daily report sheets.

A study of the charts shows that the reducing valve admitting live steam to

During January, 1912, 20½ per cent more coal was burned than during the corresponding month of 1911, due principally to the lower temperature which prevailed in January, 1912. The mean temperature for January, 1911, was 27½ deg. and for January, 1912, 13½ deg., or 51 per cent. lower. The difference in coal consumption, however, was not due entirely to the lower temperature, for something like 500 ft. more radiation was added in the fall of 1911.

A complete set of books is carried by the writer, showing the amount and cost of supplies received, when and where used, and the amount, cost and every expense connected with the plant is kept track of so that monthly and yearly cost

sheets can be made up, showing the cost of operation and maintenance of every piece of apparatus in the plant.

Fixed Charge

The cost of the plant proper was $29,000, divided between power equipment, $17,000 and heating equipment, $12,000. For depreciation 5 per cent. and for interest and insurance 7 per cent. are allowed.

The heating equipment embraces all equipment necessary in the plant, if that for power had never been installed. Half of the fixed charge on the heating equipment is charged to power for six months of the year, because power derives benefit therefrom, although it is known that using exhaust steam from Apr. 15 to Oct. 15, more than offsets this figure. Hence it is charged against power, and heat is given the benefit of the doubt.

Half of the cost renewals to boilers, furnaces, stokers, piping and feed pumps, is charged to power, when in reality the greatest amount and cost of such renewals are necessary during, or after, the heavy-heating period, because then everything connected with the steam-generating end is driven to its greatest capacity.

Labor cost is divided so that in summer, or from Apr. 15 to Oct. 15, all service is charged to power, while for the remainder of the year, power is only charged with that necessary for the maintenance and supervision of power equipment, the rest being charged to heat.

As shown by the accompanying cost sheets, the load factor is very low. This, with other conditions which obtain, is hardly conducive to a low cost per kilowatt-hour.

The report sheets, charts and cost sheets, both separately and collectively, are working out very satisfactorily, and are developing new and valuable features every day. The accompanying copies are self-explanatory and will no doubt bring out points not touched upon here.

POWER PLANT COST RECORD

Clayton & Lambert Mfg. Co

August 1911

	Power		Heat	
	Day	Night	Day	Night
Kw.-hr. at switchboard	39,570	2793		
Air reduced to kw.-hr.	3,556			
Total kw.-hr.	43,126			
Cu. ft. of free air pumped	1,482,547			
Coal used, tons	137 8	46.7		
Cost of coal used	$316 94	$107.41		
Cost of service	205 22	79.62		
Cost of oils, rags and supplies	24.11	12.06		
Cost of replacements to				
Boilers	0 85			
Stokers				
Furnaces				
Pumps	3 75			
Piping				
Engines	4 00			
Fixed charge	230 00		$60	
Total cost	784 87	139 09	60	
Cost of coal per kw-hr.	0 0073	0.0383		
Cost of service kw.-hr.	0 0047	0.0285		
Cost of supplies kw.-hr.	0 0002	0.0043		
Cost of replacements kw-hr.	0 0005			
Fixed charge kw.hr.	0 0053			
Net cost kw.-hr.	0 018	0.0498		
Net cost of all power per kw.-hr.	0.021			

January 1st, 1911 to January 1st, 1912

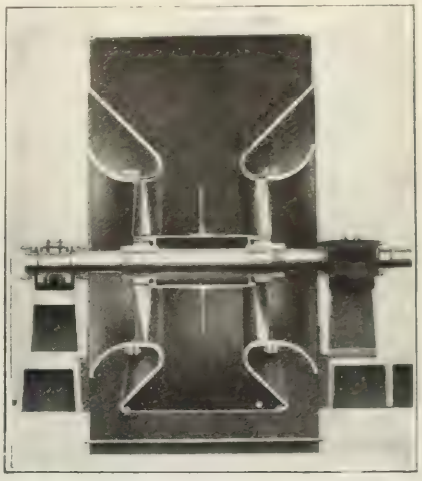
Kw.-hr at switchboard	475,641	50,390		
Air reduced to kw.-hr	57,795			
Total kw-hr	533,436			
Cu.ft. of free air pumped	14,448,990			
Coal used, tons	704 8	214 85	1392.4	1341.9
Cost of coal used	$1621 04	$494 15	$3202.52	$3086.37
Cost of service	1932 39	452 02	1095.56	988.04
Cost of oil, rags and supplies	247.29	127.16		
Cost of replacements to				
Boilers	1 75			
Stokers	154.93		197.49	
Furnaces	83.75		83.75	
Pumps	6.35		8.08	
Piping			9.30	
Engines	41.00			
Fixed charge	2400.00		1080.00	
Total cost	6488.50	1073.33	5676.70	4074.41
Cost of coal per kw-hr	0.00348	0.0098		
Cost of service kw.-hr	0.0036	0.0089		
Cost of supplies kw.-hr	0.00046	0.0025		
Cost of replacements kw.-hr	0.00054			
Fixed charge kw.-hr	0.0045			
Net cost kw-hr	0.01216	0.0213		
Net cost of all power per kw.-hr	0.01295		All Heat	9751.11

Turbine Driven Blowers

These are built in two types, the smaller sizes with double helical runners and the larger with increase-pitch propellers. The manufacturers claim for both types a high unit economy of steam per air horsepower, especially on forced-draft pressures. Both types have the runners opposed and balanced for high speed and have a central deflector to prevent cross flow, as shown in the illustration.

The runners are removable endwise and may be reversed on the shaft for right- or left-hand discharge. The bearings are lead-bronze, or babbitt lined, ring-oiled and dust proof. The helical runner type has a solid cast-iron volute, and the propeller type is made with cast-iron sides and steel volutes.

The former is designed especially for forced draft for forges and furnaces and for use with underfeed stokers. The propeller type of blower has blades with an increase-pitch helical surface to reduce the shock losses due to impact at entrance. The kinetic efficiency of these blades is said to be high and in connection with the volute casing, reduces the discharge velocity to produce a pressure instead of eddies.



Section through Blower

Losses from back flow are prevented and a strong construction is had by dovetailing the blades into a solid hub. As the pull of centrifugal force is radial there is practically no blade distortion at high speed. A peculiarity of this make of blower is the use of interchangeable wheels in the same casing, to produce different pressures for approximately the same speeds and capacities. It is made by the McEwen Bros., Wellsville, N. Y.

Barometric Condenser Practice

By Miles Sampson

The barometer injector condenser has in recent years met with favor in those power plants which are so situated that a plentiful supply of cold condensing water is available, either by using pumps or a dam and its natural static head.

By using water under pressure for starting, this condenser will lift water from a depth of 20 to 25 ft. below the cones, condense the steam, and discharge into the hotwell against the atmospheric pressure. Forced injection, however, must be handled with great care. If the

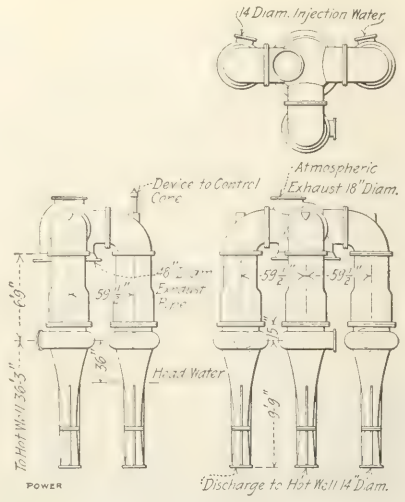
lift can be reduced to 10 ft. or less, the installation will be more satisfactory. By using a priming pipe between the injection and discharge pipes and by by-passing the cone, water is led into the discharge pipe, and a vacuum is formed. The water is then drawn over through the cones and the condenser is in operation. In such an installation the complications due to leaky tubes, power-using pumps and moving machinery are avoided, and the results obtained are surprisingly good, comparing favorably with the best pump equipments in less favored localities.

The triple-condenser arrangement as herewith shown serves a 3500-kw. horizontal turbine that is directly connected to an alternating-current generator. The turbine takes steam at 150 lb. gage and 50 deg. F. superheat. One winter day with a load of 4100 kw. and with the injection water at 36 deg. F., the vacuum was 28.4 as shown by a mercury column. At a steam consumption of 16.5 lb. per kilowatt-hour the total steam condensed would be about 68,000 lb. per hour. The total water passing per hour is probably not less than thirty times this amount, which is equivalent to a flow of 550 cu.ft. per minute. Such installations obviously require an ample supply of water.

It has been found necessary in practice to obtain certain conditions to insure a satisfactory installation. The levels are perhaps the most important feature, and should be carefully predetermined. The cone should be well above the head water, say, at least 3 ft. and at 36 ft. the hotwell overflow has been found to give the most satisfactory results. One company, to reduce the possibility of sucking water back into the engine, raises the exhaust head at least 4 ft. above the cone.

The hotwell should have a volume above the lower end of the discharge pipe equal at least to the combined volumes of the pipes above it. It will then be impossible for a slug of water to be drawn back into the exhaust pipe by the breaking of the water seal. If the hotwell is outside, provision should be made for draining it in cold weather.

Where possible, the injection piping to the cones should have no 90-deg. turns; these turns sometimes form air pockets and thus contribute to troublesome operation. The exhaust piping should be as short as conditions will allow. The double-deck arrangement, with the turbines placed over the boilers, has many advantages in the operation of this type of condenser.



TRIPLE BAROMETRIC CONDENSER FOR 30,000-KW. TURBINE

An interesting feature of the working of this arrangement is the unequal division of the load between the three legs of the condenser. With conditions seemingly identical, one of the three legs fails sometimes to carry hardly any of the load. In the instance cited, the temperatures of the discharge water were 60, 50 and 48 deg. F., and the divisions of the load were approximately 48, 28 and 24 per cent., respectively. One condenser was doing practically one-half the work. Careful repriming will sometimes help to equalize the load, but no positive remedy seems to have been found.

To recapitulate, the barometric condenser is simple and cheap to install and maintain; it is without moving parts and their unavoidable power consumption. It can easily be made to produce, in temperate climates, an average vacuum of 27.5 to 28 in. As this condenser requires only the observance of a few simple conditions and a plentiful supply of condensing water, it well deserves the recognition it has received from engineers.

Second Hand Boiler Explodes

A second-hand boiler exploded at Headric, Okla., at about 10 a.m., Nov. 8, seriously injuring two men and demolishing the boiler and engine. It was a 66-in. by 15-ft. return-tubular boiler and had been in service at Headric 11 years.

Apparently the explosion was due to corrosion and pitting of the shell longitudinally from 8 to 14 in. below the normal water line, where the shell was but ⅛ in. thick. The shell opened from one end to the other along this line. Both heads were torn loose from the shell, shearing some of the rivets and pulling some of them through the rivet holes in the rear head. The front head tore all the way around just outside of the tubes, leaving nearly all of the tubes connected to both heads. This portion of the boiler was carried about 100 ft.; the shell about 75 ft. in another direction, as illustrated herewith.

WHAT WAS LEFT OF THE EXPLODED BOILER

Nearly all of the braces pulled loose from the shell, or were broken.

The edges of the torn shell seemed to be crystallized and brittle. A working pressure of 90 lb. was carried, and the safety valve worked freely at 95 lb. The safety valve could not be found, and an examination of the shell showed that it had only been screwed into the boiler shell three threads.

The flywheel of the slide-valve engine was broken into several pieces, the hub and parts of the spokes being left on the shaft. The engine bed was cracked through just in front of the cylinder.

"Big Eats" for Chicago Engineers

The Hotel La Salle, in Chicago, one f the largest and best in the country, ave its entire staff of 1260 employees banquet on Thanksgiving Day.

In order to avoid interruption of ser- ice the employees were fed in groups nd at different hours. The engineering epartment was served at 1:30 p.m. and ad a table set in the engine room be- ause it would be impossible for the en- ire watch on duty to be absent from he plant at one time. Thus, instead of ating succulent turkey to the strains of n orchestra, the engineers ate it to the nore familiar music of the industrious ynamos, engines and pumps.

Thirty covers were set, and in addition o the engineers, oilers, firemen and coal assers, the plumbers, machinists and lectricians, who are also under the upervision of the chief engineer, sat own to the feast.

A generous menu was served, and, al- hough there was a wealth of laughing nd joking and good-natured banter, none eglected to do fullest justice to the good hings provided. The occasion will be ong and pleasantly remembered by all resent.

Practical Refrigerating Engineers' Convention

On Dec. 5, 6 and 7 the Practical Refrigerating Engineers' Association held its third annual convention at Atlanta, Ga., with headquarters at the Kimball Hotel. The first session opened on Thursday afternoon. W. R. C. Smith, of the *Southern Engineer*, acting for Mayor Candler, welcomed the engineers to Atlanta and W. H. Ross, of *Ice and Refrigeration*, responded.

In his presidential address, F. H. Ladd, of Key West, Fla., spoke of the remarkable growth of the association. Starting in 1910 with but a scarce half dozen, no less than 67 regular members and a number of honorary and associate members had been enrolled at the 1911 meeting. During the past year the association had shown a healthy growth and it

PRACTICAL REFRIGERATING ENGINEERS AT ATLANTA

HOTEL LA SALLE ENGINEERS' THANKSGIVING FEAST IN THE ENGINE ROOM

is hoped that the meeting in Atlanta will add materially to the membership in the eastern section of the Southland. At present the membership is largely made up from Texas and Alabama. An examination is required for admittance and already owners and managers are showing their confidence in men belonging to the association.

Friday morning, W. H. Ross, in behalf of the American Association of Refrigeration, extended a cordial invitation to attend the third international congress of refrigeration to be held in Chicago during September, 1913. It was suggested that the Practical Association fall in line with others in the field and hold its annual convention in Chicago at this time. The opportunity of making the present organization a national association, as was originally intended, would never be better.

W. J. Rushton, who is an Atlanta man and president of the American Association, spoke briefly about the congress and the benefits to be derived by those attending.

While those at the convention realized the importance of attending in a body, September was a busy month for the refrigerating engineer and the trip was long. It was decided not to hold the annual convention at this time, but a committee was appointed to see that as many of the members as possible would go to Chicago.

With a view to extending the association at the congress, provision was made to amend the bylaws and provide for state chapters, the president or head of which would be a vice-president of the national association. The advantage of installing local chapters in such states as Illinois, Ohio and New York, where refrigerating engineers are more numerous, was fully appreciated and the association has great hopes of accomplishing its purpose at Chicago.

On Friday afternoon two papers on the steam boiler were read: one by Norman Register on its installation, care and management, and the other by William Francis on preparing a boiler for inspection. Next morning the question box, which had been left in the convention hall during the meeting, was opened and inquiries on soldering ice cans, measuring the contents of a tank lying on its side and calcium chloride losing its strength were fully discussed.

The entertainment program consisted of a visit to the Atlanta Brewing & Ice Co.'s plant where the conventioners did justice to an oyster roast with plenty of Atlanta's favorite brew on the side and incidentally a spirited contest between one of the negro employees and the brewery's goat. Keith's vaudeville was the attraction Friday night, and on Saturday afternoon the new raw-water ice plant of the Universal Ice Co. was inspected. In the evening an informal dinner at the Mechanics & Manufacturers Club brought the convention to a close.

Hot Springs, Ark., was chosen as the meeting place for 1913. The following officers were elected: President, Carl Nessler, Texas City, Tex.; vice-president, J. Higginbotham, Scranton, Miss.; secretary and treasurer, J. B. Embrey, Shreveport, La.; third place on board of directors, D. L. Gilbert, Dallas, Tex.; third place on board of examiners, W. A. Owen, Shreveport, La.

Car Heater Explosion

An accident of a serious nature, though fortunately not attended with disastrous results, occurred on a car of the Fort Wayne & Northern Indiana Traction Co. on the evening of Dec. 6. The car was passing through Logansport, Ind., about 11 o'clock and had just discharged the last passenger when the hard-coal heater of the hot-water heating system exploded with great violence, wrecking the forward end of the car and setting fire to it as well.

According to reports the motorman, who was protected by a partition, was thrown down amid a shower of glass, though not seriously injured. The heavy windows of the front end were blown out, the partition between the baggage and smoking compartments was torn loose and the heater was reduced to scrap. Had the accident occurred with a crowded car the results might have been very serious.

In answer to a request for an account of the accident and its causes, the company stated that it did not want to give out any information for publication, as it would not be fair to the manufacturers of the heater nor of any decided benefit to the users; users evidently meaning purchasers rather than the traveling public. About two years ago this traction system, then the Fort Wayne & Wabash Valley Traction Co., was forced to reorganize because of a succession of disastrous wrecks and consequent damage suits.

SOCIETY NOTES

The American Society of Agricultural Engineers has planned the program of its annual meeting at the Great Northern Hotel, Chicago, Friday morning, Dec. 27. Among the interesting topics will be "Isolated Gas Lighting Plants," by Eugene Becker, and "The Principles of Fuel Oil Engines," by C. F. Hirshfeld.

OBITUARY

George A. Kimball

George Albert Kimball, chief engineer of the Boston Elevated Railway Co., died on Dec. 3 at his home in Arlington, Mass., aged 62.

The transportation system of Boston was largely the work of Mr. Kimball. The Cambridge subway and elevated extension to Cambridge were designed and their construction was supervised by him. The layout of the street-car service in Boston and suburbs served by the elevated company is also largely the result of his efforts, as he was connected with the West End Street Ry. Co. before it was merged with the elevated.

In collaboration with Howard A. Carson, formerly chief engineer of the Transit Commission, and with Edmund S. Davis, the present chief engineer of that board, Mr. Kimball helped to design the East Boston tunnel, the Tremont street subway and the Washington street tunnel, which were built by the Transit Commission and subsequently leased to the elevated.

Alfred P. Boller

Alfred Pancoast Boller, formerly chief engineer of the Manhattan Elevated R.R. and consulting engineer for the Department of Public Parks of New York, died on Dec. 9 at his home in East Orange, N. J. He was born in Philadelphia in 1840 and was graduated from the University of Pennsylvania in 1858, later receiving his degree of civil engineer from Rensselaer Polytechnic Institute, Troy, in 1861. He was associated in 1862 with the engineering work on the Lehigh Canal and later went with the Philadelphia & Erie R.R. In 1865 he became chief engineer of the Hudson River R.R.

Mr. Boller was vice-president of the American Society of Civil Engineers, president of the American Institute of Consulting Engineers, and a member of the Institute of Civil Engineers of London, England. He is survived by his wife, two daughters and three sons.

PERSONAL

Peter Neff, of Canton, Ohio, has resigned his position with the Arctic Ice Machine Co. to open an office as a consulting engineer. Mr. Neff will specialize in refrigeration.

J. C. McQuiston, advertising manager of the Westinghouse Electric & Manufacturing Co., on his recent retirement as president of the Pittsburgh Publicity Association was presented a handbag containing a silver-mounted traveling set in recognition of the work he has performed for the association.

S. Rosenzweig, mechanical engineer of the Erie City Iron Works, on his present lecture tour of Western universities, will include the universities of Cincinnati, Illinois, the Armour Institute of Technology and probably Purdue University. Mr. Rosenzweig will also speak before the A. S. M. E. branch of the Brooklyn Polytechnic Institute and Yale, Harvard and Lehigh universities on Jan. 4, 7, 10 and 14.

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POWER

Vol. 36 NEW YORK, DECEMBER 31, 1912 No. 27

How an unsuspecting public is menaced by dangerous lap-joint, return tubular boilers located under sidewalks. Doubly dangerous are the boilers when operated by incompetents.

Lentz System Applied to Steam Engine*

By Sigfried Rosenzweig

The Lentz engine which is now used extensively in Europe, employs valves of the poppet type both for admission and exhaust. These permit the employment of high superheat as a result of which some exceptionally low steam consumption rates have been obtained.

Thermodynamics teaches that the efficiency of an engine increases when the temperature limits between which it works are increased. That occurs in a steam engine when steam enters the cylinder at the highest temperature and leaves it at the lowest possible. Hence, to obtain economy in a steam engine three factors are of greatest importance: First, high steam pressure; second, high superheat, and third, high vacuum.

The greater the steam pressure in a steam engine, the greater is its capacity to develop power. By increasing the steam pressure from 100 to 200 lb., the output of the engine can be practically doubled, or it can do the same work as before with much less steam. Steam at 200 lb. pressure contains only 10.8 B.t.u. per lb. more than steam at 100 lb.; so if the coal has 14,000 B.t.u. per lb., only about $^1/_{1000}$ part of a pound of coal is necessary to raise the steam pressure from 100 to 200 lb. As the high-pressure boiler costs very little more than the low-pressure boiler, these facts forcibly point toward the adoption of high steam pressures. Modern stationary engines use boiler pressures from 175 to 200 lb.

That the use of superheated steam raises the efficiency of an engine is generally acknowledged. The reasons for it are various, chief among which is the reduction of initial condensation and the leakage past valves and piston. As these two items represent from 25 to 50 per cent. of the total steam consumed in an engine, their reduction is important.

When dry saturated steam enters the cylinder, it has to come in contact with surfaces which have just been in contact with cold exhaust steam and which are covered with a film of moisture. As water serves as a ready means of transferring heat from the steam to the metal a great quantity of the incoming steam is wasted by condensation. The use of superheated steam reduces this initial condensation, extensive experiments having shown that superheated steam, because of its low density, is a poor heat conductor and that quite naturally this property becomes more and more marked the higher the superheat.

Professor Doerfel, of the University of Prague, found that by increasing the superheat from 46 to 256 deg. F. the loss during the admission period decreased from 8400 to 1900 B.t.u. Higher superheat will probably, apart from radiation, prevent almost entirely the loss of heat due to the exchange of heat between cylinder walls and steam. These experiments were confirmed by another authority, Seemann, who found that by superheating from 0 to 360 deg. F. the heat given up to the cylinder walls decreased from 36.2 to 9.3 per cent. Another experiment showed that with a superheat of about 210 deg. F. it made no difference as to the initial condensation whether the engine was running condensing or noncondensing, though in the former case the temperature drop due to the expansion of steam amounted to 195 deg. against 135 deg. F. in the latter.

These experiments, which were confirmed by many other authorities, show conclusively that it is advisable to use high superheat to obtain high economy. The author from his own experience has found that a superheat of from 200 to 250 deg. F. is quite commercial and gives very satisfactory working results. The higher initial cost for installing superheaters is easily counterbalanced by the smaller piping, as superheated steam allows of considerably higher steam velocities, by the greatly reduced heating surface in the boilers due to the vastly superior economy in the steam engine, by the greater safety of the power plant, due to the absence of water in the pipes and engine, and by the greatly reduced coal consumption.

To illustrate this point may be mentioned the power plants of the electricity stations located in Vienna and in Rome. The former has twenty piston-valve engines of 1000 hp. each. When the superheaters were installed, the high-pressure cylinders had to be reconstructed and were fitted with Lentz valves and gear. The outlay of about $80,000 was recovered within one year from the saving of coal. In the power station at Rome the installation of Lentz engines and superheaters increased the output of the plant by about 30 per cent. The introduction of superheat further simplified greatly the design of steam engines, so that compound engines with highly superheated steam successfully replaced triple and quadruple expansion engines, thus producing a saving in initial cost, of floor space and running expenses.

The application of superheat has been restricted for various reasons, chiefly because of the bad effect of the high temperature on valves unsuitable for superheat, but these obstacles were never sufficient to blind engineers as to the merits

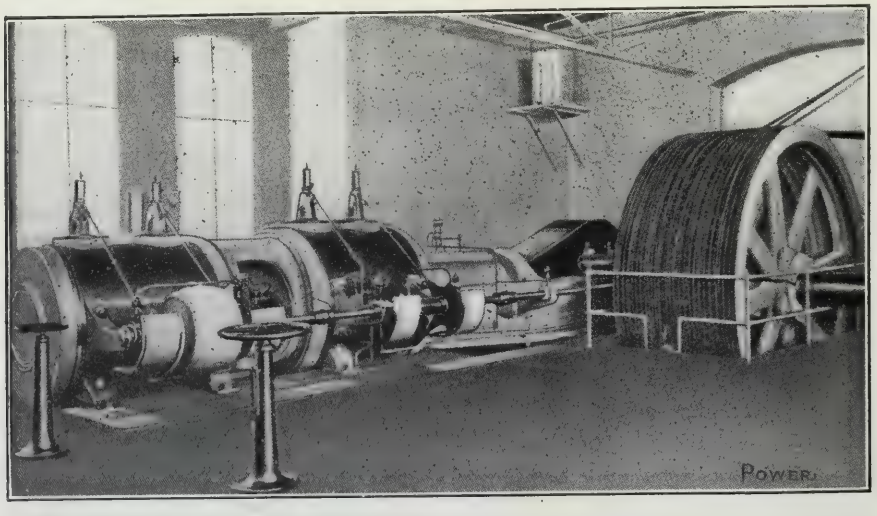


FIG. 1. VALVE-GEAR SIDE OF LENTZ ENGINE

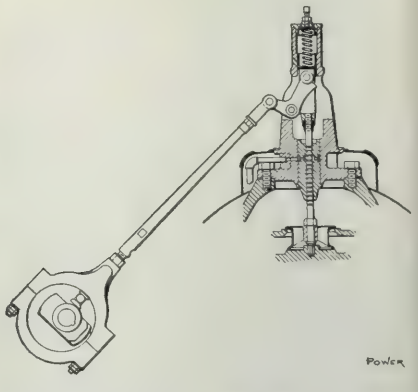


FIG. 2. SECTION THROUGH VALVE

*From a paper read before the Ohio Society of Mechanical, Electrical and Steam Engineers.

of superheated steam. Owing to the expansion of the metal all valves which have to rely upon the surrounding walls for steam tightness are unsuitable for superheat. For high steam pressures and high degrees of superheat a balanced and frictionless valve has been used, and the deficiencies of the slide, piston and Corliss valves can only be remedied by the adoption of the balanced multiple-seated poppet valve.

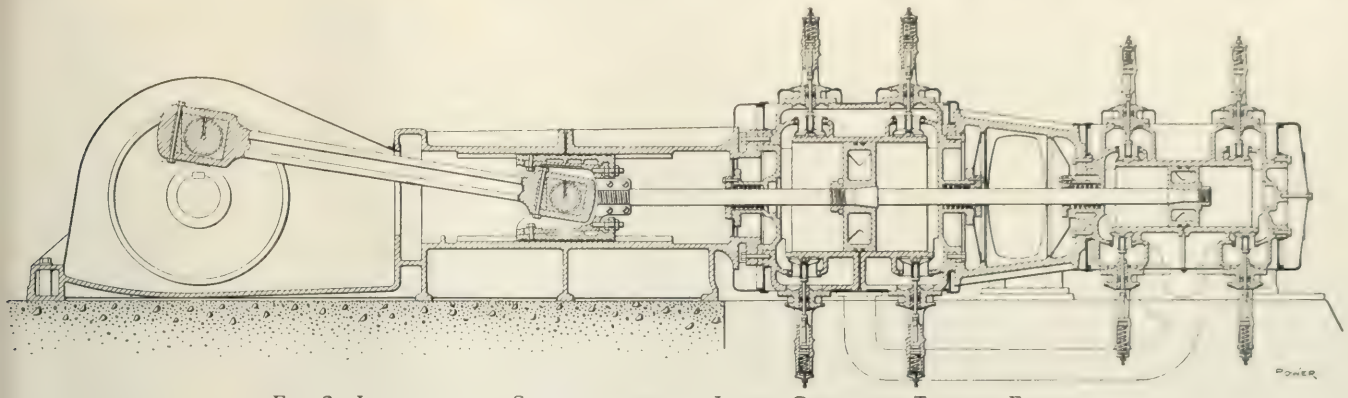


FIG. 3. LONGITUDINAL SECTION THROUGH LENTZ COMPOUND TANDEM ENGINE

The advantages of the poppet valve are no rubbing surfaces, hence no lubrication and its adaptability for high temperature; small weight and a practical balance, hence its adaptability for high speed and high steam pressure; no possible wear, consequently the economy of the engine is maintained for an indefinite period.

Probably the best representative of this type of engine is the Lentz, a view of which, from the valve-gear side, is shown in Fig. 1. A layshaft runs alongside the engine, driven from the main shaft by a pair of bevel gears. All the eccentrics are keyed on this layshaft, each separate eccentric operating one valve by means of an eccentric strap rod and lever. In this way the different valves are entirely independent from each other and can be set very accurately according to the requirements of steam distribution.

Fig. 2 shows the different parts of the internal valve-gear, the valve, the spindle with its grooves, and spindle guide with roller and cam lever. All the parts, as pins, bushings, rollers and cams are case-hardened to prevent wear. The valve is entirely balanced but for the width of the seat, which even in the larger sizes does not exceed ⅛ in., and hence takes very little power to operate. The valve-gear consists only of a roller and cam lever. The latter is oscillated by means of the eccentric rod and is so curved that it opens the valve and closes it very rapidly. No dashpots are used as the whole operation is positive. On account of the noiseless operation of the valve-gear and its great simplicity it is possible to operate the Lentz engine safely at high speeds, a limit to which has not been reached at even 350 r.p.m. A feature is the absence of any spindle packing, as the spindles are simply ground into cast-iron bushings and provided with grooves, thus producing a labyrinth system in which the condensed steam and oil collects and by which the escape of any steam is successfully prevented. A similar principle is used for the piston-rod packing, which is made entirely of cast-iron rings but so arranged as to follow the floating action of the piston rod.

A section through the cylinders of a tandem compound engine is shown in Fig. 3. The low-pressure cylinder is placed next to the bed. This arrangement gives the high-pressure cylinder a chance to expand freely without being obstructed by the distance piece and the low-pressure cylinder. The opening in the distance piece is made large enough to remove the back cover and piston of the low-pressure cylinder without disturbing the alignment of the engine.

All the low-pressure and high-pressure exhaust valves are driven from fixed eccentrics, the high-pressure steam eccentrics only being influenced by the governor.

Fig. 2 shows also the arrangement of the high-pressure steam eccentric, which is arranged to slide on a block keyed to the shaft. When the load changes, the governor slides the eccentric on the block, and alters the angular advance and the throw of the eccentric according to the requirements of the load, while the linear lead remains constant. The governor is placed between the two steam eccentrics and influences their position without the help of a belt or chains.

The governor is shown assembled in Fig. 4. It consists of two pendulums pivoted to a support which is keyed on the shaft. By means of links the pendulums are connected to the governor inertia body, which can move freely around the shaft, being connected with the shaft only by a flat circular swing which counterbalances the centrifugal forces of the pendulums. The potential energy stored up in the inertia body is converted to kinetic energy as soon as the slightest change of load and consequently of speed occurs. As the inertia forces and the centrifugal forces act in the same sense their combination produces a governor of a very sensitive and instantaneous action. It is possible to change the speed of the engine while in motion by means

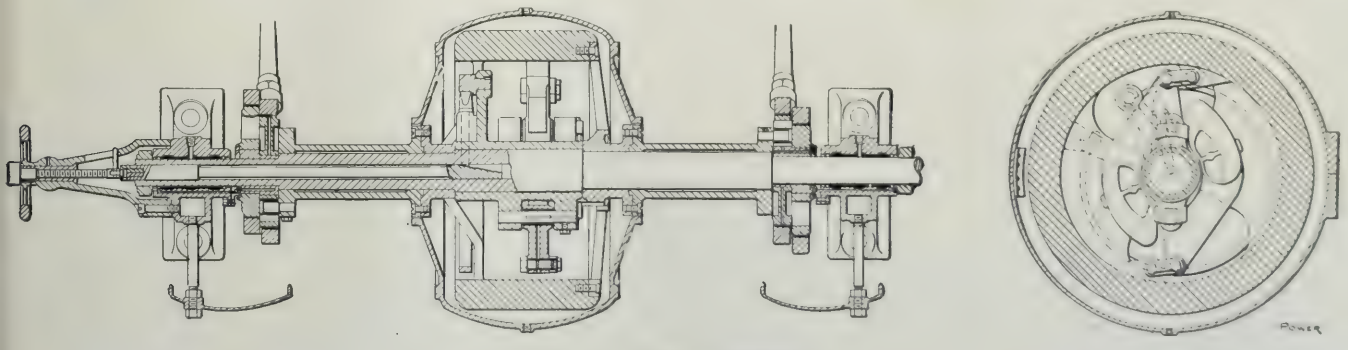


FIG. 4. LENTZ GOVERNOR

of a radial pin which presses against the spring and changes its tension. The radial pin can be moved by means of a shaft and handwheel which is placed at the end of the layshaft.

Extensive tests have been carried out on a Lentz simple engine having a 15-in. cylinder and 21-in. stroke. This engine showed a steam consumption of 19.2 lb. per i.hp.-hr. with an average steam pressure of 145 lb. gage, an average superheat of 47 deg. F. and a back pressure of 2 lb. This performance gives a Rankine efficiency of practically 76 per cent.

A tandem engine carefully tested gave the following results: The first test with saturated steam, 170 lb. boiler pressure, and 26 in. vacuum, gave 366 i.hp. with a steam consumption of 12.3 lb. per i.hp.-hr. The second test with the same conditions as before but with superheated steam of 150 deg F. showed a steam consumption of 10.4 lb. per i.hp.-hr. These results were obtained from an entirely new engine within two days of first running.

The Lentz system has also been extensively used in connection with semi-stationary engines. a combination of an engine with boiler, superheater, condenser, air pump, feed-water heater and feed pump, in other words, a compact, high-class and self-contained power plant which is extensively used in isolated plants in Europe and South America. On account of its compact arrangement and the absence of any piping and losses incurred by it, the economy of these engines is remarkable. One of these engines was tested by Prof. Grassmann, of the Technical College of Carlsruhe, Germany. The 200-hp. unit gave a steam consumption of 7.4 lb. and a coal consumption of 0.92 lb. per b.hp.-hr., figures which probably stand as world records. The mechanical efficiency of the engine was 93 per cent. and the combined efficiency of boiler and superheater 82 per cent.

Power Plant Records

By G. E. Miles

The articles published from time to time have clearly shown that the best way for an engineer in an isolated plant to keep the central station from getting his business is, first, to operate his plant at its highest efficiency and, second, to be able to show his employers what the power is costing them.

It has also been shown that daily records are necessary to enable the engineer to accomplish the above results, and with the hope that some may be helped, the following suggestions and sample daily report are submitted. The fact that this form was designed for, and used by, a small central station should deter no one from adapting it to the use of an isolated plant. Often the best way to fight fire is with fire, and certainly the isolated plants can learn much from the central stations that will enable them to better compete with the latter.

At the top of this form are spaces for each engineer to record the time of his going on and off watch. The first column of blank spaces is for the record of the load on the plant. If recording wattmeters are used of a form that can be read closely every half hour, the load can be best determined from them, as the output for the period is desired rather than the instantaneous load at the time of reading. If there are no wattmeters from which the load can be determined, the load can be estimated from the ammeter readings.

The second column is for the steam pressure and the third and fourth show which engine was running. The latter is conveniently done by drawing a line from the time the engine is started until it is stopped. The fifth, sixth and seventh columns are for the arc-lamp circuits. Isolated plants will rarely need these columns for arc lamps, but they may need them for other purposes.

From the data in the first column, curves can be plotted showing graphically the load on the plant at any time. These curves, although not essential, are nevertheless very convenient.

At the top on the right-hand side the weight of coal used is recorded. Many engineers will balk at this because they have no means of weighing their coal, but means can be provided at a small expense and these data are absolutely essential.

At the plant where this form was used small platform scales were adapted to the purpose by building a platform for the scales large enough for a wheelbarrow to be run upon it. As the coal had to be wheeled into this plant, the coal was weighed with little extra trouble. At another plant an ordinary washtub was provided with a bail and hung on a steelyard that would weigh 150 lb. of coal at a time. When small coal is used it can be measured in a tub with sufficient accuracy if a tubfull is frequently weighed to keep a check on the weight. Weighing the coal and keeping the records will cause the engineer additional labor, but the satisfaction of knowing

DAILY STATION REPORT

ENGINEERS

NAME	FROM	TO

DATE

19

A.M.	K. W.	Steam	Eng. No.1	Eng. No.2	Cir. No.1	Cir. No.2	RR	P.M.	K. W.	Steam	Eng. No.1	Eng. No.2	Cir. No.1	Cir. No.2	RR
12								12							
30								30							
1								1							
30								30							
2								2							
30								30							
3								3							
30								30							
4								4							
30								30							
5								5							
30								30							
6								6							
30								30							
7								7							
30								30							
8								8							
30								30							
9								9							
30								30							
10								10							
30								30							
11								11							
30								30							
12								12							

COAL

12 to 6 .....

6 to 6 .....

6 to 12 .....

Total .....

WATT METER READING

Midnight .....

6 a. m. .....

Noon .....

6 p. m. .....

Midnight .....

WATER METER READING

Noon .....

6 p.m. .....

12 midnight .....

6 a.m. .....

Boil. No. .... On... Off. ...

Cylinder Oil, ...Qts

Engine Oil, ...Qts

Waste, ...Lbs

Packing, ...Lbs

Boilers Cleaned.....

Repairs, ...

REPORT INTERRUPTION OF SERVICE ON OTHER SIDE

CHIEF ENGINEER

what results he is securing in the operation of his plant will amply repay an ambitious engineer.

The next spaces are for the record of the wattmeter readings at six-hour periods. If no wattmeters are provided, the output for these periods can be computed from the readings recorded in the first column, provided the load is not subject to frequent and large changes. If there are such changes, a recording wattmeter is necessary to determine the output with reasonable accuracy.

Where boiler-feed water is paid for on a meter basis, the readings of the water meter should be recorded in spaces provided for that purpose.

In the absence of wattmeters the load on a direct-current plant may be easily found by multiplying the current in amperes by the voltage and dividing by 1000 to reduce to kilowatts. The load on an alternating-current plant cannot be as easily or as accurately determined without wattmeters as the power factor is seldom known, and when three-phase generators are used the phases are seldom balanced.

The following approximation will serve to give an engineer a comparison from day to day, and will be much better than no information as to what the plant is doing: Multiply the amperes by the primary voltage and the product by the power factor for single-phase circuits; the power factor will be between 75 and 95 per cent. For three-phase circuits take one-third of the sum of the amperes on three phases and multiply by the volts and this sum by 1.732, also multiply by the power factor. Care should be taken to get the correct primary voltage, which is the voltage as shown by the meter multiplied by the ratio of the transformer supplying the meter. This ratio is usually 20, so that when the meter reads 110 volts the primary volts are 2200, but if the meter reads 115 volts the primary volts are 2300.

An engineer may make the measurements mentioned and record the data without deriving much benefit from doing so, but if he studies the figures and computes the cost of power in pounds of coal per kilowatt-hour and then compares the results at different times he will be able to see what conditions give the best results and also to keep a check on the work of his subordinates.

All this means more work for the men, who probably feel that they are doing more than they are being paid for, but it will pay in the end. It is the man who knows what he is doing, and can do, who can confidently ask his employer for the pay he knows he is earning.

The Horsepower of a Pipe Line

By R. S. Bayard

The relation between static and friction head, flow and velocity. Determining the proper velocity and flow to get the greatest horsepower out of a line of given size with a given static head.

"How much water will flow through a 2-in. pipe under 100 lb. pressure?"

Many readers will recognize this as a familiar question, which they themselves have asked or have been called upon to answer. The trouble is that the question is not complete. Upon inquiring where the pressure is measured, the usual reply is: "Oh, right at the end, or perhaps 6 in. back from the end." With water flowing out of an ordinary open pipe without any end obstruction it is absolutely impossible to obtain a pressure of 100 lb., or any other pressure above atmospheric, at the end, or even 6 in. back.

If the water is flowing out rapidly its energy of velocity will exert a pressure in the direction of its motion, as in driving a Pelton waterwheel; but, without a contracted nozzle, the pressure against the walls of the pipe at its end must be atmospheric.

To produce a flow, there must be pressure somewhere. Then, where should this pressure be measured, and what becomes of it at the end of the pipe?

If a pipe wall offered no frictional resistance to the movement of the water, no pressure at all would be required to make it flow along the pipe to its open end, and in fact, it would be impossible to build up any pressure even with a pump, no matter how much water was being handled, nor how long the pipe.

What necessitates pressure, or "head," to cause a flow through an open-ended pipe is friction between the water and the pipe walls. This friction depends upon the length and diameter of the pipe, and the velocity of the water through it. The greater the length, the more the friction, and so with the velocity. Increase in pipe diameter, other things remaining unchanged, decreases the friction.

So, before answering the original question, it must be known how long is the pipe and at what point the pressure is measured. As an example, consider what might be a Western mountain hydraulic power plant. Imagine a pipe line down the side of a mountain and let the lower end of the pipe be 450 ft. below the dam which catches the water and directs it into the pipe. Also let the length of the riveted steel pipe be 6000 ft., or a little over a mile, and its diameter 16 in. Such a combination of conditions is not out of the ordinary.

Assume that the pipe is of uniform diameter throughout its length, and for the present has no waterwheel nozzle at its lower end, but is provided with a gate valve giving when open practically no obstruction to the flowing water.

First, imagine that the gate valve at the lower end is closed, that there is no flow at all and that the pipe is full to its upper end, 450 ft. higher than the gate valve. Under these conditions the pressure at the gate valve is equivalent to the "head" of water above it. It is

$$450 \times 0.433 = 195 \textit{ lb. per sq.in.}$$

Pressures in such a pipe line are usually expressed in head of water, or feet of vertical water column. Thus, with the gate valve at the lower end closed, the head of water is 450 ft. Now, assume that the valve is partially opened, allowing a slight flow of water. The reading of the pressure gage on the upstream side of the valve will scarcely be affected, but if the valve is opened wider, allowing an appreciable flow, the gage reading will decrease. The reason is that the flowing water produces a friction force on the walls of the pipe, which diminishes the previous gage pressure. Part of the 450-ft. head is being expended in "friction head."

If the valve is opened still wider, the flow will become greater, increasing the velocity of the water along the pipe walls and increasing the friction. The increased friction head will further decrease the original static head, so that the gage reading will be still lower.

Up to this time the partially opened valve has exerted appreciable resistance to the flow of the water, this becoming less the greater the flow, because the friction head helps to hold the water back. As the valve opening is increased the gage reading will continue to decrease until finally the valve will be opened wide and the gage will register no pressure.

Under these conditions the flow of water will be so great that all of the static head will be absorbed in friction, with the exception of a very few feet, which is represented by the increased velocity of the moving mass of water. At this time it may be truly said that the quantity of water is all that will flow through the given pipe with the original given pressure, and this quantity would be the answer to such a question as that originally asked.

To get a visual idea of these condi-

tions, the accompanying curves have been plotted. In this diagram, horizontal distances represent cubic feet of water flowing per minute through a 16-in. pipe, 6000 ft. long, with a total static head of 450 ft., while one of the vertical scales so marked represents feet head of water pressure. For the curve marked "Head Lost in Friction," the left-hand scale gives the feet of head absorbed in friction and the corresponding quantity of water flowing may be read from the lower scale.

It may be observed that for a flow of 200 cu.ft. per min., the friction head is about 12 ft., which subtracted from 450 ft. leaves 438 ft. available head, as shown by the curve marked "Available Head at Lower End." When the flow is 500 cu.ft. per min., the friction head is about 65 ft., leaving an available head of 385 ft. It is this available head that exerts the pressure on the gage at the lower end of the pipe. Its reading under the last condition would be about

$$385 \times 0.433 = 167 \text{ lb. per sq.in.}$$

This means that the friction is absorbing 28 lb., and the remaining 167 lb. is being resisted by the partially open valve at the exit. At this rate of flow the velocity is so low that its energy represents a negligible part of the head.

The curve further shows that with the gate fully open, and offering no resistance at all to the flow, the head lost in friction will reach the full value of the original static head of 450 ft. All the original head has now been used up in forcing the water at high speed along the pipe whose friction holds it back. The gage will show no pressure at the lower end of the pipe. This takes place when the flow is practically 1400 cu.ft. per min.

When the water is flowing at the above rate its velocity through the 16-in. pipe will be

$$V = \frac{Q}{A};\ V = \frac{1400 \times 144}{201 \times 60} = 16.72 \text{ ft. per sec.}$$

the area of a 16-in circle being 201 sq.in. At this speed the velocity head, or portion of the static head used to impart this much energy to the moving column of water, is

$$h = \frac{V^2}{2g} = \frac{16.72^2}{2 \times 32.16} = 4.35 \text{ ft.}$$

As this quantity is so small compared to the total head of 450 ft. and as the actual coefficient of friction must be guessed at for a new installation, in the absence of experiments under the actual existing conditions, this quantity may safely be neglected. This is especially so for working conditions, as less than half this quantity would be allowed to flow to a waterwheel, because to do any work from its original head of 450 ft. some of this head must be left available at the pipe exit or wheel. With half the quantity, there would be half the velocity and so one-fourth the velocity head, or not much over 1 ft. This is far within the "guessing limit" of the actual friction head.

The available head at the pipe exit brings up the question of what velocity to use and what flow to allow to get the greatest horsepower out of a pipe line of a given size, with a given static head. To figure available power of the flowing water, it is necessary to multiply the pounds of water flowing per unit of time by the head available at the bottom end of the pipe after the friction head has been deducted.

As an example, with the 6000 ft. of 16-in. pipe with 450 ft. of static head, let the flow be 500 cu.ft. per min. The curve gives a friction loss of 65 ft. and so an available head of 385 ft. The water horsepower then would be

$$W.hp. = \frac{500 \times 62.4 \times 385}{33{,}000} = 364$$

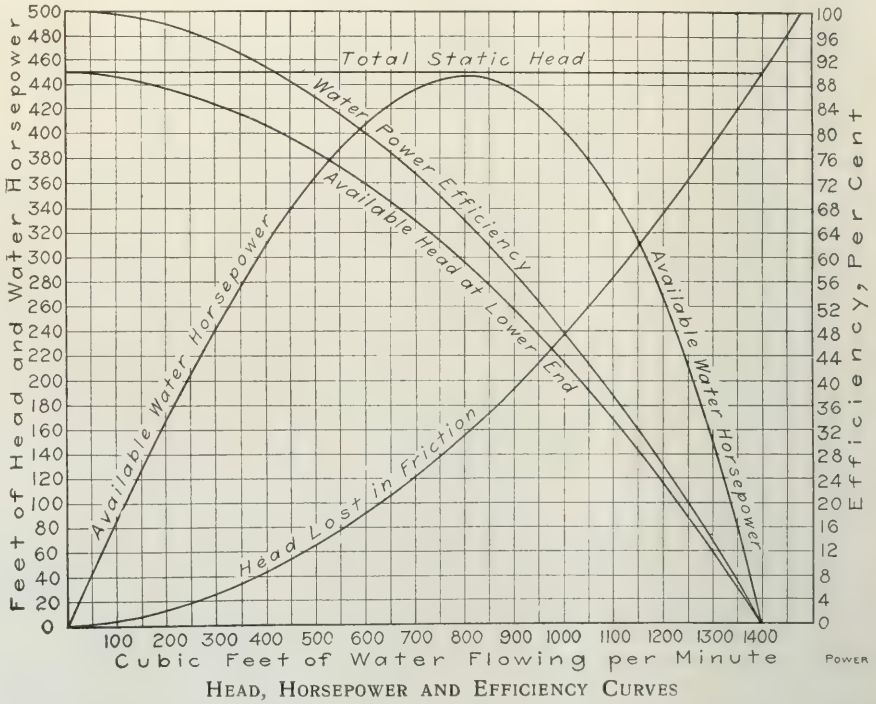
moving mass. At the maximum flow of 1400 cu.ft. per min., the velocity head was found to be equivalent to 4.35 ft., so the power would amount to about

$$\frac{1400 \times 64.4 \times 4.35}{33{,}000} = 11.9 \text{ hp.}$$

This is a very small percentage and as a matter of fact, it is doubtful if this pipe could ever deliver 1400 cu.ft. per min., because the curves show the entire 450 ft. of static head used up in friction at this capacity; whereas part of the available head, after deducting for friction, must be represented in the velocity of the flowing mass. At 1390 cu.ft. per min. the velocity would be 16.6 ft. per sec., and the velocity head would be

$$\frac{16.6^2}{2 \times 32.16} = 4.28 \text{ ft.}$$

At this point also the friction loss is just about this much short of the total



HEAD, HORSEPOWER AND EFFICIENCY CURVES

If the flow is allowed to be 800 cu.ft. per min., the lost head of friction will be about 155 ft. and the available head 295 ft. The horsepower will be about 446. A waterwheel would not be able to deliver all of this, due to its own imperfections and so this available power would be further reduced by the mechanical efficiency of the wheel. The further reduction would amount to somewhere around 20 per cent., with a first-class wheel.

It will be observed that there are two conditions of flow where there would not be any horsepower available at all. These occur when there is no flow, with all the head available, and when there is the maximum flow, with all the head used up in friction.

Under this last condition there would be some power due to the velocity head already spoken of, or the energy of the 450 ft. of static head, so all of the static head would be represented in friction and velocity. However, all of these close figures are "hair splitting" on such work and the two theoretical points of no horsepower available are about as the curves show.

What, then, is the best velocity to employ in such a pipe line to develop power, or what is the maximum power available with a given size and length of pipe, and at what water velocity will this be obtained, assuming that the necessary volume of water is available?

To answer this question, a glance at the chart shows that the maximum horsepower is about 447 and it occurs when the friction head amounts to about 155 ft. This loss by friction is just a little over one-third of the total head available. It can be mathematically shown that the maximum horsepower will be

available in a given pipe line, when about one-third of the static head is sacrificed to friction.

Such a condition may be said to give the maximum efficiency of the pipe, though it certainly is not the maximum efficiency of the use of the water. A larger pipe would, of course, give the same flow with less friction loss, or a greater flow with the same loss, but it also would be true of the larger pipe that it would deliver its maximum power when the friction loss through itself became one-third the total head. It is only a matter of how much money it is desired to spend on the pipe line, and whether or not it is more economical to pay for a bigger pipe, used below the point of its maximum horsepower and thus use the available head to better advantage.

By assuming that there is no friction loss and the water power is being used at 100 per cent. efficiency, and so on down to zero efficiency with all the head lost in friction, a curve such as that marked "Water Power Efficiency" in the chart is obtained.

From the efficiency curve, it may be noticed that the water-power efficiency is about 66 per cent. at the point of maximum water horsepower. This naturally follows from the fact that one-third of the static head is sacrificed to friction. As previously stated, there will be the further loss of efficiency of the waterwheel, or turbine, amounting to 20 per cent. or more, giving an overall efficiency of somewhere around

$$66 \times 80 = 5280 = 52.8 \textit{ per cent.}$$

Of course, if the money is available, and a higher efficiency is wanted, a pipe may be used that will reduce the friction loss to, say 10 per cent., giving an overall efficiency in the seventies. These matters are all subjects for study to suit results to costs, and each case requires individual analysis and judgment by the designer.

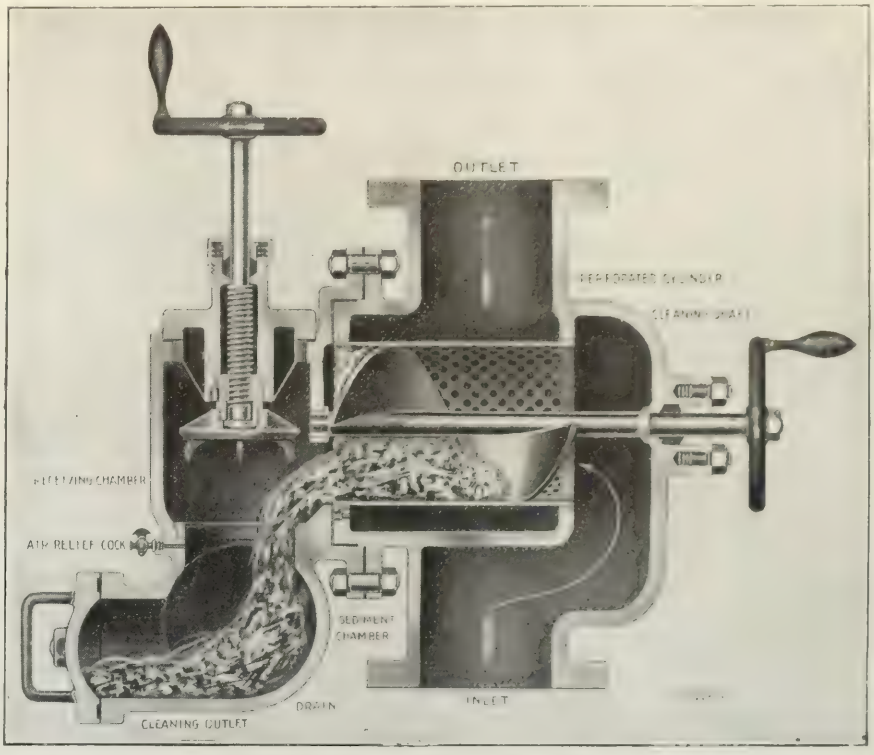
Holt Automatic Strainer

This device is designed to automatically remove all kinds of solid foreign matter from water supply lines without interrupting the flow of water. It consists of a cast-iron casing, having an inlet and outlet opening and a perforated brass cylinder, having open ends, interposed between the inlet and outlet of the casing. The liquid to be strained flows into one end of the perforated cylinder and passes out through the perforations into the outlet of the casing.

Foreign matter is retained on the inside of the perforated cylinder until the cleaning shaft is rotated. The blade of the cleaning shaft, which has a $\frac{1}{32}$-in. clearance between it and the inside wall of the perforated cylinder, sweeps the inner surface and forces the foreign matter out of the opposite end from that through which the liquid enters the receiving chamber, from where it falls by gravity into the sediment chamber.

There is only one valve, and this need be operated only occasionally, when the sediment chamber is to be cleaned out. In practice, all that is required to keep the strainer cleaned, is to occasionally turn the handwheel connected to the cleaning shaft about two complete revolutions, and whenever necessary, clean out the sediment chamber, both operations being performed without interrupting the flow of water through the strainer.

This strainer is made by the Rosedale Foundry & Machine Co., North Side, Pittsburgh, Penn.



SECTION THROUGH AUTOMATIC STRAINER

Water Tank Support Collapses

On the morning of Nov. 12, a 45,000-gal. wooden tank collapsed at Pile Brothers Starch Works, Indianapolis, Ind., putting the plant out of commission. In falling it completely wrecked the pump house, and four double-acting water pumps were badly damaged and will require extensive repairing before they can be used again. Fortunately nobody was injured. The falling tank also broke a 4-in. steam line and three flanges on a 10-in. steam line, making it necessary to shut off steam on all buildings. Water from the tank ran into the engine room and flooded the wheel pits.

Four columns supported the tank, each made in three sections of 9-in. iron pipe, ¼ in. thick and 20 ft. long. The joints were made with couplings, ⅝ in. thick and 9¾ in. long. Diagonal and horizontal braces ran from opposite corner couplings. The horizontal braces were made of 4-in. pipe, ⅜ in. thick, and the diagonal braces were of 1 in. round iron, threaded at the ends, and screwed into the cast-iron fittings. Each rod had a turn-buckle in the center.

There were indications that one or more of the diagonal braces were subjected to a stress greater than their elastic limit, which allowed one or more of the columns to buckle, breaking one of the couplings near the top and leaving the tank unsupported on one side. The damage is estimated at $5000.

The first mention of the occurrence of coal in the United States, according to the United States Geological Survey, is made in the journal of Father Hennepin, a French Jesuit missionary, who in 1679 recorded the site of a "cole mine" on the Illinois River, near the present city of Ottawa, Ill. The first actual mining of coal was in the Richmond Basin, Va., about 70 years after Father Hennepin's discovery in Illinois, but the first records of production from the Virginia mines were for the year 1822, when, according to one authority, 54,000 tons were mined.

Reports of the anthracite coal trade are usually begun with the year 1820, when 365 long tons were shipped to Philadelphia from the Lehigh region. Prior to this, however, in 1814, a shipment of 22 tons was made from Carbondale, also to Philadelphia. It is probable that the actual production prior to 1820 was between 2500 and 3000 tons.

The production for 1911 was 496,221,168 short tons.

Progress of Wolf Locomobile Engines

By F. E. Junge

In the last fifty years steam pressures have increased from eighty or ninety pounds to over two hundred pounds. The average horsepower of the locomobiles manufactured at the Wolf shops has increased from 8 hp. to 80 hp. They turned out 90,000 hp. of locomobiles last year.

The firm of R. Wolf, Magdeburg-Buckau, Germany, has just published a memorial, edited by the well known historian Conrad Matschoss, which gives a comprehensive survey over the field of its various activities during the last fifty years. The following is a short abstract of the development as it relates to the "Locomobile," or integral steam plant in which the engine boiler and accessories are mounted together in a concrete unit.

There are at present 88 types of locomobile engines built by the firm, differing in detail in so far as they are destined for use either as stationary or portable machines, simple or compound, for use with superheated or saturated steam, condensing or noncondensing, and, of course, varying in capacity within the range of these various applications. Though elasticity of production is an indispensable condition of German manufacture, which has to satisfy the exacting requirements of extensive markets at home and abroad, there is a growing tendency toward standardization, an attempt to make the various parts for the different types interchangeable, in order to reduce the total number of parts used, and thereby the cost of manufacture, to a minimum.

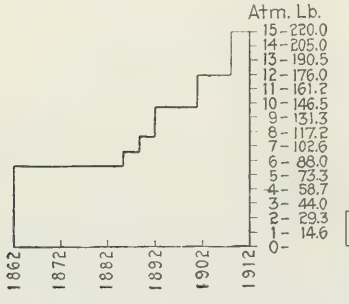


FIG. 1. INCREASE IN BOILER PRESSURES

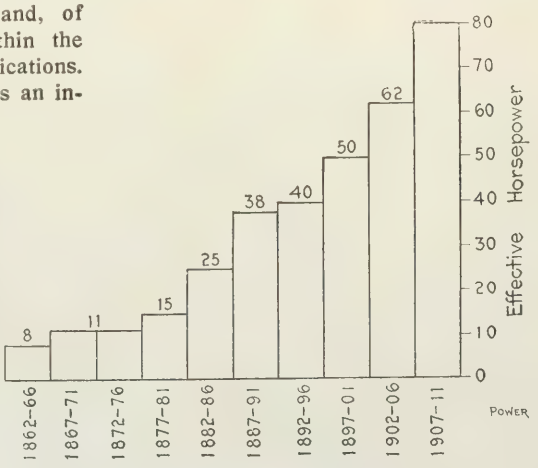


FIG. 3. GROWTH OF AVERAGE HORSEPOWER OF LOCOMOBILES

Fig. 1 shows the gradual increase of boiler pressures employed in the various periods of 50 years. It is seen that during the first 20 years the boiler pressure remained at 6 atmospheres (88 lb.) while today a pressure of 15 atmospheres (220 lb.) is employed in some types. The indicator diagrams assembled in Fig. 2 will give a better idea of the evolution of the locomobile and of the improvements realized than would a survey of the details of construction. As the diagrams are all drawn to the same scale a comparative analysis is possible. Of the denominations used, n means the number of revolutions per minute, p is the boiler pressure, p_i the mean indicated pressure and t the temperature of the steam when entering the high-pressure cylinder.

In this table diagrams 1 to 4 refer to single-cylinder locomobile engines using saturated steam. No. 1 was obtained from an engine having flat slide valves and throttle regulation. These engines have been built since 1862, the card being taken in 1878. The weaker lines in the diagram represent lower loads. Diagram 2 is from an engine having a Meyers hand-adjustable cutoff valve and throttle regulation. The type has been built since 1874, the card was taken in 1878. Diagram 3 refers to an automatic cutoff engine with Rider cutoff valve and Porter governor. The card was taken in 1880. No. 4, taken in 1892, refers to the same system of governing as the foregoing, but to a boiler pressure of 10 atmospheres (147 lb.).

Nos. 5 and 6 refer to single-cylinder locomobile engines using superheated steam. The distribution of the steam is in all types effected by means of a piston valve with elastic rings, the travel of this valve and its angle of advance being adjusted by a governor acting directly on the driving eccentric. The first engines of this type were built in 1898. The diagrams numbered 6 were taken at various loads.

Diagrams 7 to 10 refer to compound locomobiles using saturated steam. Nos. 7 and 8 were obtained from engines using Rider valves on the high-pressure and flat slide valves on the low-pressure side. Nos. 9 and 10 were taken from engines built since 1892, having Rider valves on the high-pressure and Trick canal slide valves in the low-pressure cylinder.

Diagrams Nos. 11 to 14 refer to compound locomobiles using superheated steam. They are all equipped with piston valves controlled by the governor. Nos. 11 and 13 refer to engines having flat slide valves and 12 to one having Trick's slide valve in the low-pressure cylinder. No. 14 is from the latest type of engine in which the exhaust from the high-pressure cylinder as well as the admission into the low-pressure cylinder are controlled by the same piston valve.

In the 50 years which have passed since the inauguration of the firm the maximum output per unit of locomobile engines manufactured has increased from 20 hp. in the period between 1862 and 1866 to 900 hp. in the period between 1907 and 1911. If the total capacity of all locomobiles turned out in the same

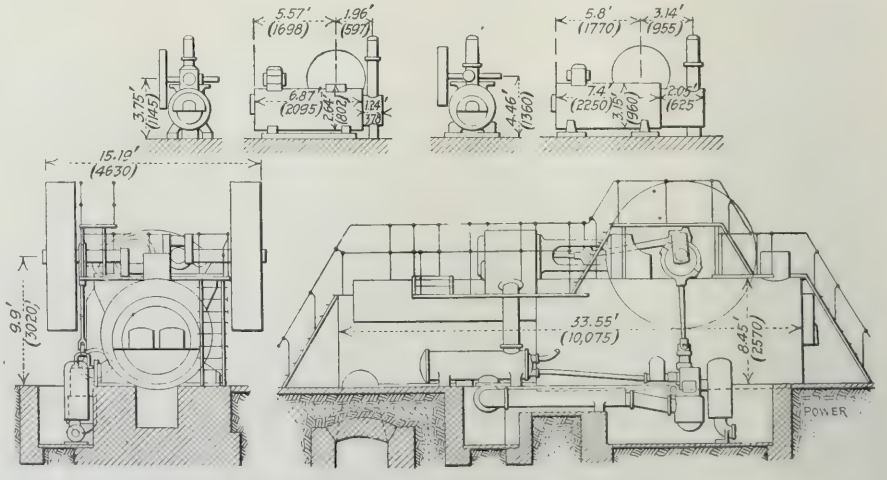


FIG. 5. SMALLEST AND LARGEST STATIONARY LOCOMOBILE

period of time is divided by the total number delivered, the average capacity of the locomobiles manufactured is obtained. Fig. 3 shows how remarkable has been this growth of average output; from 8 hp. in the first five years to 80 hp. in the last five years. The total capacity of all locomobiles turned out by the Wolf firm in 1911 was 90,000 hp.

Though, owing to its peculiar combination of boiler and engine in one unit, the locomobile type of engine is limited in its power output, this remarkable growth of capacity per unit has been effected without changing the principal design and construction of the engine. Fig. 4 gives a graphic representation of this development, showing the smallest portable locomobile ever built as well as the smallest and the largest standard type in stock. Fig. 5 shows the corresponding development of stationary locomobile engines.

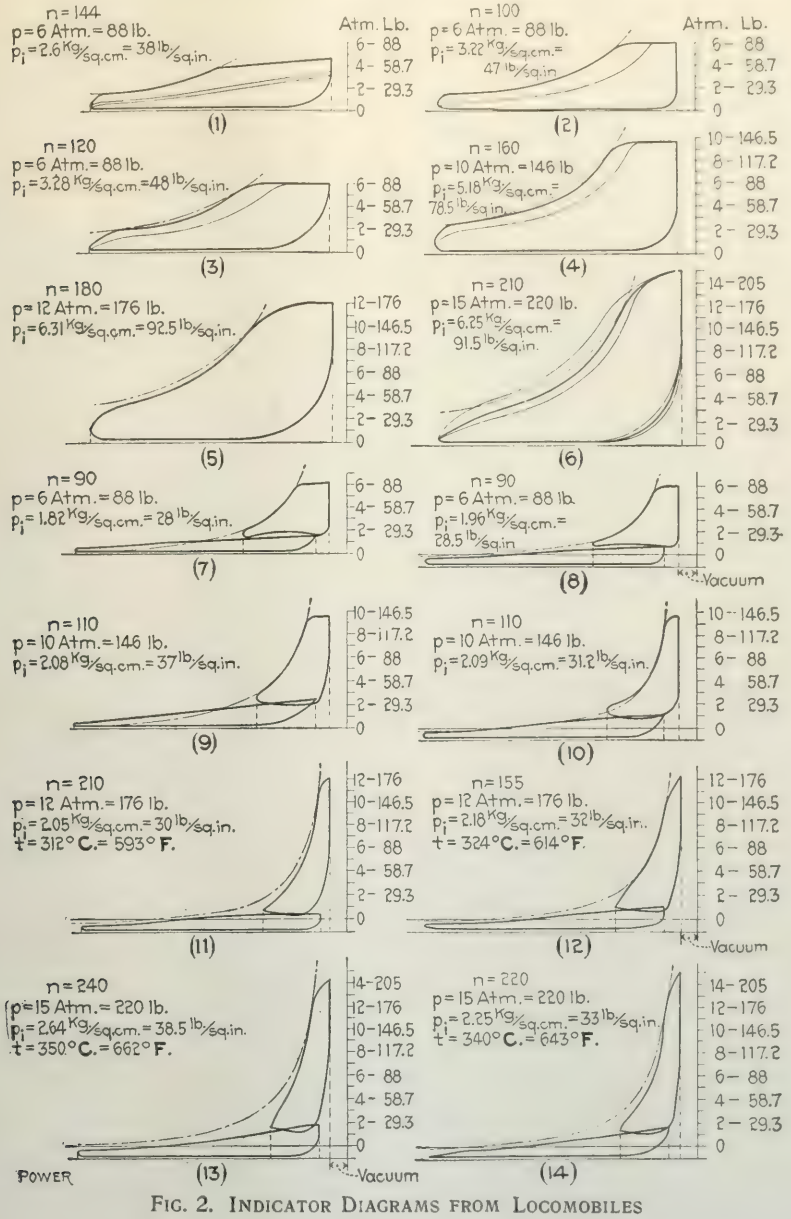


FIG. 2. INDICATOR DIAGRAMS FROM LOCOMOBILES

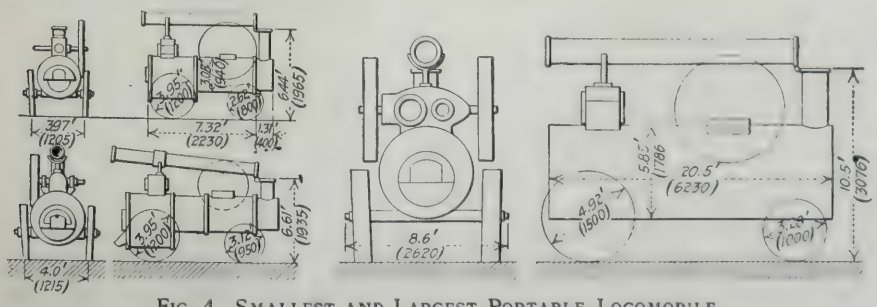


FIG. 4. SMALLEST AND LARGEST PORTABLE LOCOMOBILE

Considerable attention has been devoted to the reduction of fuel consumption, one of the most important factors of operating cost. How successful has been this attempt is shown in Fig. 6, giving the results of tests made by eminent experts in the course of 30 years. Fifty years ago the consumption of coal per horsepower-hour was hardly less than 5 kg. (11 lb). Today guarantees of not to exceed 0.5 kg. (1.1 lb.) for the larger types; or one-tenth the amount of the original consumption may be had. With unusually high degrees of superheat consumptions as low as 0.358 kg. (0.79 lb.) per hp.-hr. are obtained with 100-hp. units.

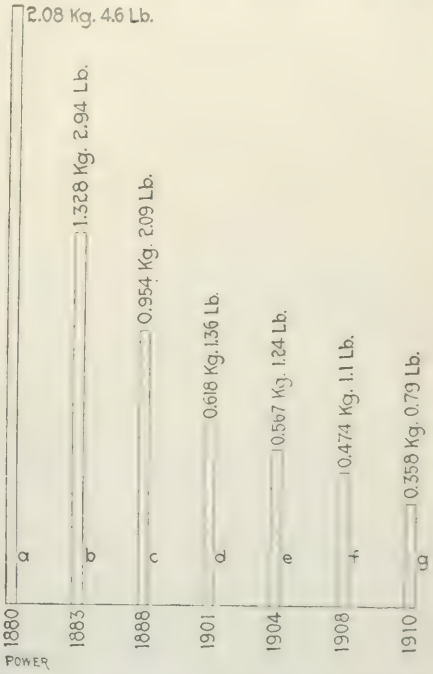


FIG. 6. REDUCTION IN COAL CONSUMPTION

It is interesting to compute the saving thus realized in pecuniary equivalents. With coal prices such as obtain in Germany the annual saving in fuel cost thus realized by improvements in the course of 20 years for a 50-hp. locomobile amounts to 3031 marks or $722. The total saving of coal so made possible represents a considerable conservation of the natural wealth.

This survey over a field of industry in which American manufacture has not even made a feeble attempt to compete, cannot be concluded without reflecting, in the briefest possible manner, upon the causes which exist and are responsible for the lack of initiative in this line of endeavor. For one who has been concerned with the comparative study and analysis of technical progress in Germany and in the United States for the last ten years, it seems unfortunate that, owing to the abundance of fuel resources there is no urgency from within, and, owing to the restrictive tariff of 45 per

cent. on manufactured goods, there is no urgency from without, compelling manufacturers to concentrate their energies in the direction of efficient service to consumers.

It is safe to say that the stagnation which is making itself so unpleasantly felt today in various lines of mechanical engineering in the United States, among others in the realm of high-grade power-plant apparatus, is due primarily to this lack of influx and stimulus from abroad which, when admitted would force many a manufacturing concern to abandon its old-fashioned cut-and-try methods and move on, if it wanted to be sure of a competency and a profitable business.

Unless these conditions change, American consumers are certain to receive inferior service for their money, and American manufacturers are sure to be left behind in the conquest of foreign markets. Because, as long as there is an alternative to unsatisfactory dealing, foreign consumers will ask for the best things money can buy, not for the things which it is most profitable for manufacturers to sell. If they can choose between an inefficient standard engine and an efficient modern machine they are sure to invest their money where it is likely to yield them the greatest immediate returns. The present system of seclusion reduces to an inferior status both the American producer and the American consumer and is bound to work infinite harm to the country, unless by a reasonable change of commercial policy, it is amended—the sooner the better.

Setting Corliss Engine Valves

By J. K. McIntyre

Directions for setting the valves of a Corliss engine for the first time, according to the method employed by erecting men.

The valve-gear adjustments are illustrated by drawings.

Most engineers know how to adjust the valves of a Corliss engine already in service, but the method of setting the valves and making adjustments for the first time on a new engine is perhaps hazy to all but a few of the most experienced engineers.

In the following the shop method of setting the valves of a single-eccentric noncondensing engine is described.

Wristplate Movement

First, adjust the eccentric rod for length, so that the rocker-arm travels an equal distance each side of the central position; see Fig. 1. Then turn the eccentric on the shaft until the rocker-arm is plumb, set the wristplate central and clamp it in position. Adjust the length of the hook rod to suit, and scribe a line on the hub of the wristplate and another to correspond on the wristplate stand, the latter mark being made light so that it can be changed later.

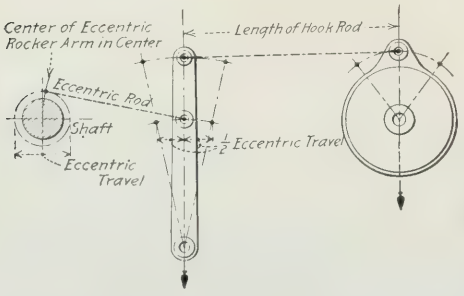


Fig. 1. Rocker-arm and Wristplate in Central Position

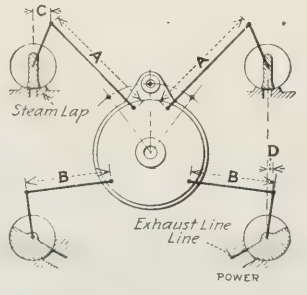


Fig. 2. Steam and Exhaust Valve Adjustment

Throw the eccentric ahead to the outermost travel and mark the wristplate stand in line with the mark on the wristplate hub, to show the extreme travel on one side of the center; turn the eccentric to the opposite extreme and scribe a mark on the wristplate stand to show the opposite extreme travel. This applies only to new engines. Next adjust the length of the steam and exhaust connections *A* and *B*, to suit *C* and *D*, Fig. 2. By using the dimensions *C* and *D* rather than following the working drawing for the length *A* and *B* any variation in the location of the ports is overcome.

Setting for Lap

With the wristplate plumb, set the steam and exhaust valves to the dimensions given on the working drawing and then mark the valve stems for the keyways, cut and drive the keys into place.

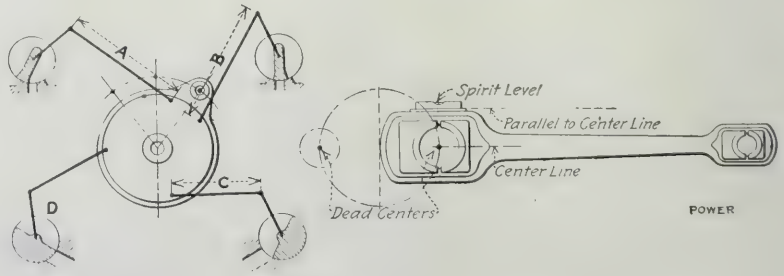


Fig. 3. Steam and Exhaust-valve Adjustment. Full Port Opening

Fig. 4. Leveling the Crank

While setting the steam valves, it is important to see that the releasing gear latch is locked up with the steam arm as when the valve is in operation.

When resetting the valves of an old engine, in which the keyways have been cut in the valve stems, and also to determine the steam "lap" as nearly as can be done without specific figures, proceed as follows: Set the wristplate plumb and adjust the exhaust rods *B*, Fig. 2, so that the exhaust-valve edges are line and line with the port edges; turn the wristplate to its extreme right-hand travel and adjust the length of the steam-valve rod *A*, Fig. 3, bringing the edge of the valve to a fullport opening. Turn the wristplate to the opposite extreme travel and repeat for valve *B*. Then center the wristplate and note if the steam laps at each valve are the same; if not, equalize them.

Adjusting the Dashpot Rod

Next, for either new or old engines, turn the wristplate to the right-hand extreme travel and adjust the dashpot rod for the valve *B*, bringing the hardened-steel block of the releasing-gear hooks to a hooked-on position with $\frac{1}{32}$-in. clearance; turn the wristplate to the opposite extreme travel and adjust the dashpot rod for the valve *A*.

Turn the crank shaft until the crank is nearly on the head-end dead center. Lay a spirit level on that part of the connecting-rod which is parallel with the center line of the rod, Fig. 4, and turn

the crank shaft until an exact level is indicated. Then turn the eccentric in the direction it is to run until the head-end steam valve opens to the proper lead. This should be as follows:

Diameter of cylinder in inches	Lead in inches
10	3/32
12 to 16	7/64
18 to 22	1/8
24 to 26	9/64
28 to 30	5/32

Scribe a line on the shaft and one on

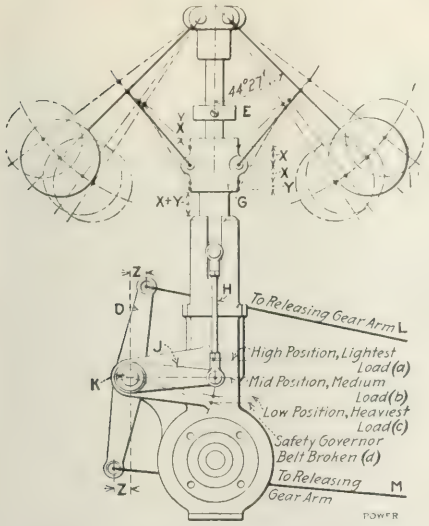


FIG. 5. GOVERNOR ADJUSTMENT

the eccentric hub as a check against slipping and draw up one setscrew in the eccentric sufficiently to prevent its slipping. Turn the crank to the crank-end dead center, level as before and observe the position of the crank-end steam valve. If the hook-rod pin is at the top of the wristplate, the crank-end steam valve should show less lead than was given to the head-end valve, or even a slight lap. If the hook-rod pin is at the top of the wristplate, the crank-end steam valve should show less lead than was given to the head-end valve or even a slight lap. If the hook-rod is connected to the lower part of the wristplate, the crank-end steam valve should show too much lead. Either of these conditions is to be corrected.

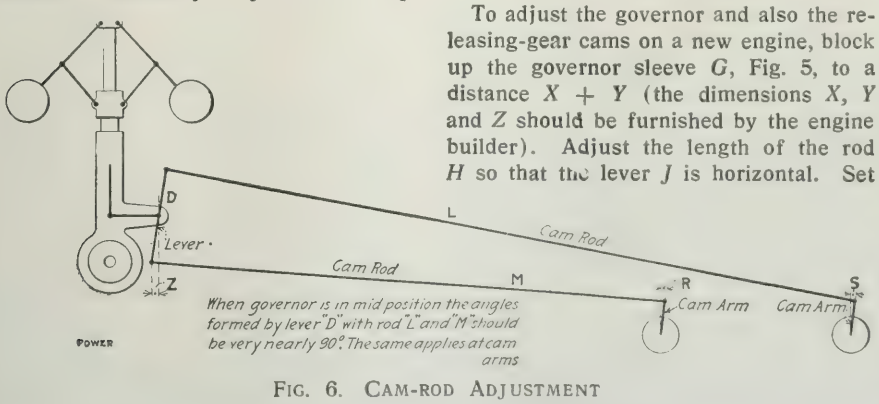


FIG. 6. CAM-ROD ADJUSTMENT

Slacken the eccentric setscrew and turn the eccentric in the proper direction to move the crank-end steam valve a distance equal to one-half the required correction. The other half of the correction is to be made by lengthening or shortening the eccentric rod. If the amount of correction required is very small, it may be done by adjusting the length of the connection from the wristplate to the crank-end steam valve. When the valves at each end are adjusted for the proper lead, tighten the setscrews in the eccentric and correct the travel marks on the wristplate bracket. With the crank on each dead center, the exhaust lap should be noted and a correction made to equalize any difference by adjusting the lengths of the connections to the exhaust-valve arms.

There are four important points in the travel of any governor; *a*, the high position, Fig. 5, which trips the valves before they have advanced far enough to open the ports; *b*, the mid position, which is used only for adjusting the positions of the cam arms before locating the cams; *c*, the low position, which draws the cams back to allow the wristplate to rock through its full travel without tripping the valves; *d*, the safety position, which brings the safety cams into operation if the governor belt breaks or runs off its pulleys. The high safety position is for the safe operation of the engine and the low position determines the extreme load-carrying capacity of the engine.

GOVERNOR ADJUSTMENT

To adjust the governor and also the releasing-gear cams on a new engine, block up the governor sleeve *G*, Fig. 5, to a distance $X + Y$ (the dimensions *X*, *Y* and *Z* should be furnished by the engine builder). Adjust the length of the rod *H* so that the lever *J* is horizontal. Set the lever *D* to the dimension *Z* as given on the engine builder's drawing and key the levers to the shaft *K*. Adjust the cam rod *L* and *M*, the proper length to suit the dimensions *R* and *S*, Fig. 6, as shown on the erecting drawing.

Lower the governor from the mid position to the low position. (There should be a device on the governor to hold it up to this position) and turn the wristplate to its extreme travel toward the crank shaft and adjust the cam *J*, Fig. 7, on the head end of the cylinder. This should be slipped around the cam-arm hub *L* until the leveled edge comes to a position 3/32 in. from the hardened-steel block *G* and fasten *J* to *L*. Turn the wristplate to the opposite extreme travel and repeat for the crank-end cam. With the governor in the *low position* the cams should *not* trip the releasing gear.

Next raise the governor to its high position and block it. Move the wristplate slowly and at the same time observe the steam valves. With the governor in this position the cams should trip the releasing gear before the valves come to their opening points. If this condition is not obtained, raise the governor a trifle higher. Allow about 1/16 to 1/8 in. steam lap at the moment of tripping the valve. When this is provided for, bring the collar *E*, Fig. 5, down to the sleeve *G* and fasten.

SAFETY CAMS

To adjust the safety cam *H*, Fig. 7, lower the governor to the low position and block it. Turn the wristplate toward the

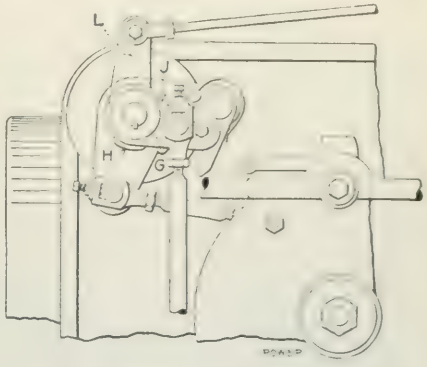


FIG. 7. VALVE-GEAR ADJUSTMENT

head end to its extreme travel. Set the head-end safety cam to within 3/32 in. of the hardened-steel block, Fig. 7. The holes in these safety cams are slotted to allow for some adjustment independent of the cam arm. The fastening bolt should be located to allow for making this adjustment in either direction. To test the safety cams, lower the governor to a position 1/4 in. below the low position and turn the wristplate to its extreme travel in both directions. The releasing-gear hooks should *not* move the steam valves. If the engine is old and not provided with safety cams, they should be added. To allow for this, lower the governor to its very lowest position and then block it up 1/2 in. Let this position be the low position and adjust the cam rods to move the cams to within 3/32 in. of the hardened-steel hook blocks when the wristplate is turned to its extreme limits of travel. Then raise the governor until the steam valves trip at about 1/16 in. lap.

Block the governor in this position and bring the set collar *E*, Fig. 5, down on the sleeve *G* and fasten the set collar to the governor spindle. Lower the governor to the low position 1/2 in. above the very lowest or safety position and adjust the safety cams, as for a new engine. If the valves are carefully set very little, if any, adjustment will be required when the indicator is applied to the engine.

Power Plant of Sectionalized Machinery

By George O. Green

Mule-back transportation is frequently used in mountainous districts to move mining machinery, which is built, if possible, in sections weighing not over 200 lb. Heavier pieces must be carried by gangs of laborers.

Parts of machinery which cannot be made to meet this maximum weight are sometimes packed on a cradle slung between two mules in tandem. Steel cables, weighing 600 lb., are brought in by three mules in tandem. The cables are divided into six coils of 100 lb. each and two coils are placed on each mule, with a short length of cable hanging loose between them.

An extra long and heavy cable was once carried into a Mexican mine by uncoiling it and placing laborers at a distance of 10 ft. apart. At a whistle signal, given by a foreman, each man lifted a portion of the cable and carried it until another signal was given to stop and rest.

The trail leading to the mines of the South American Development Co., near Zaruma, Province of El Oro, Ecuador, is a notoriously bad one. Machinery can be transported by river steamer to Santa Rosa, but for the remaining distance to the mine, 45 miles, everything has to be packed in by either one of the methods mentioned.

The greater part of this stretch of trail is the bed of the Santa Rosa River, which is forded 49 times in a distance of 13½ miles. After leaving the river an ascent of 7000 ft. is made in the course of a three hours' ride.

It is impossible to give any idea of this trail from a photograph, but Fig. 1 shows a bad part. Fig. 2 is a view of a mule arriving at the mines laden with the top section of a steel form used in the construction of a concrete pipe now in course of construction for a hydraulic power plant.

The main part of the plant consists of a 40-stamp mill, three compressors, and a cyanide plant, power for which is obtained from 10 waterwheels under a head of 38.8 ft. There are also pumps, hoisting engines and rock-drills, which are run by compressed air. This machinery was all transported in sections. The water for driving the waterwheels is taken

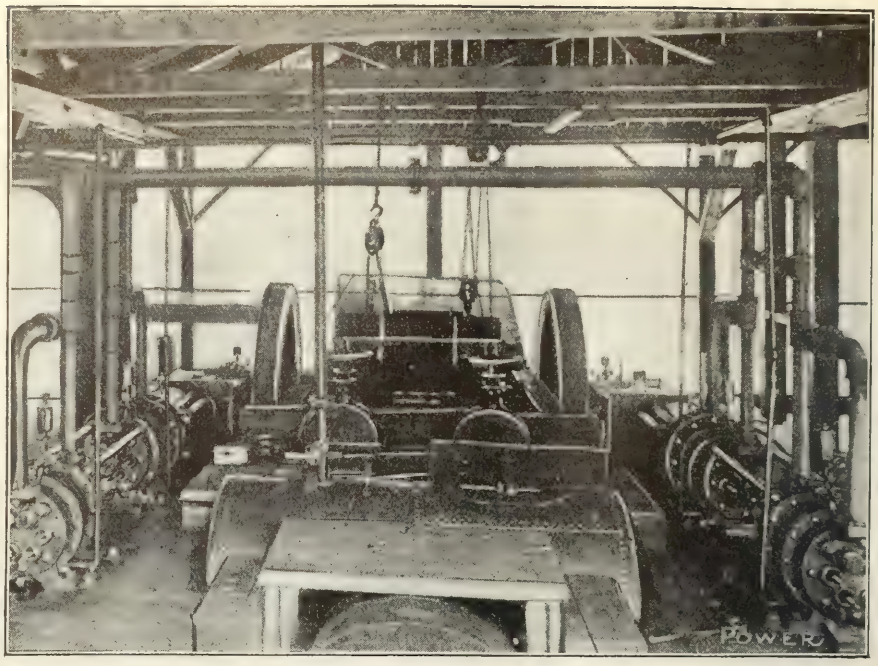


Fig. 5. General View of a Duplex Tandem Air Compressor

Fig. 1. A Bad Piece of Trail

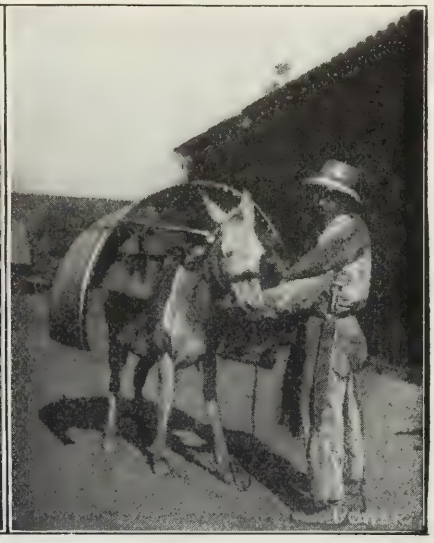


Fig. 2. Mule Arriving at the Mine

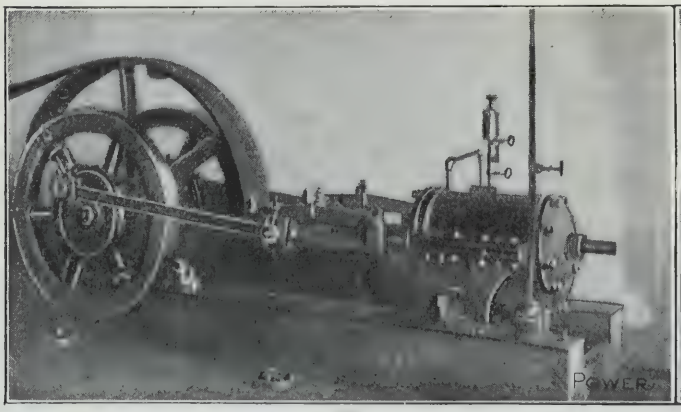


Fig. 3. One of the Air Compressors

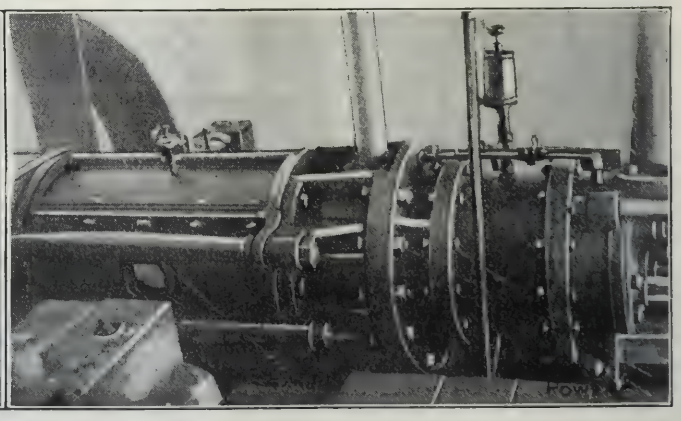


Fig. 4. Sectional Construction

from the Amarillo River through a canal about a mile long.

In Fig. 3 is shown a compressor, an excellent example of sectionalized machinery. Fig. 4 is a view of a cylinder, guide and distance piece of one of these compressors. The distance piece is in two sections; the top and middle piece are plainly shown. The bottom section is similar to the top section. The cylinder is composed of the usual cylinder

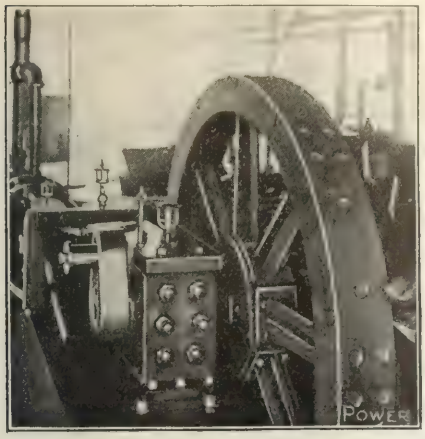


FIG. 6. FLYWHEEL BUILT IN SECTIONS

heads, fitted with poppet inlet and outlet valves, three outer casing sections and a liner. The liner is a driving fit in this outer casing, but is made so as to leave an annular chamber to serve as a water jacket. Fig. 5 shows one of the sectionalized duplex, tandem air compressors.

The flywheel, Fig. 6, is made up of 42 castings. The main bearing and the two top and bottom nuts on the stay rods which hold the main bearing and the guide sections together are also shown. Similar rods on each side hold the front and rear ends of the cylinders together, as in Fig. 4.

One side of the compressor is held in alignment by a steel skeleton frame, embedded in the concrete foundation, with holes drilled to match holes in the cylinder and different sections of the working parts of the machines.

These compressors run at 100 r.p.m., compressing air to a pressure of 100 lb. per sq.in. One has been running for 14, and the other for seven years.

Three natives attend to the lubrication, cleaning, pressure regulating, stopping and starting the compressors. This is about all they are capable of doing, any attempt to teach them to pack piston rods or to take up bearings has proved a failure, but they are good, faithful workmen and easily managed. Their pay amounts to one gold dollar a day.

A clearing house for the issuance of information respecting engineering standards will be established by the American Society of Mechanical Engineers. The prominent engineering societies have been invited to coöperate.

Automatic Motor Controller

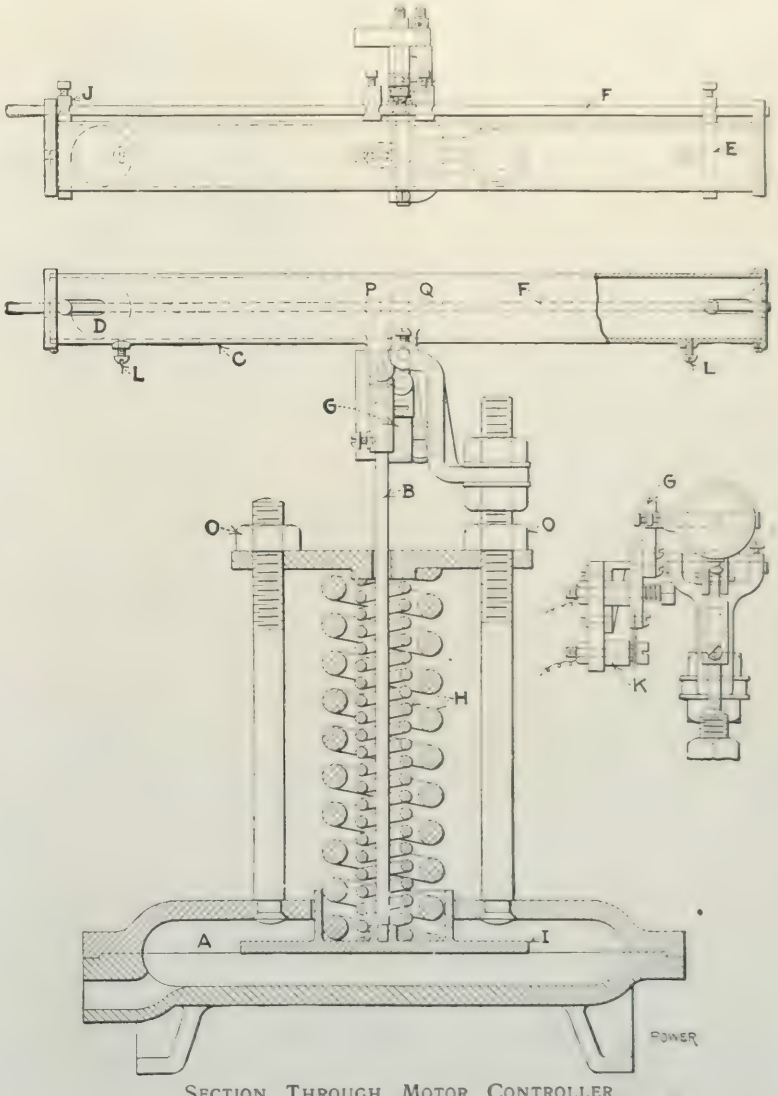
This device is designed to maintain a uniform water level in tanks, standpipes or reservoirs by automatically controlling the action of the motor driving the pump.

The pressure from the tank coming underneath the diaphragm *A* forces the rod *B* upward, tilting the cylinder *C*, when the ball *D* will roll to the opposite end, hitting the pin *E*. The latter being attached to the horizontal rod *F* causes the stop *P* on the rod to move the switch *G*, thus stopping the motor.

When the water level in the tank is lowered, the springs *H* force the diaphragm *A* downward. The rod *B* which is attached to the diaphragm plate *I* tilts the cylinder *C* in the opposite direction, causing the ball *D* to strike the pin *J*. This also being attached to the horizontal rod *F* shifts it in the reverse direction, causing the stop *Q* on the rod to move the switch *G* into contact with the post *K*, which starts the motor.

The adjusting nuts *OO* above the springs *H* regulate the average height of water maintained in the tank, that is, its general level, and the range between the maximum and minimum levels may be varied by turning the setscrews *LL*, shown at each end of the cylinder. These act as obstructions to the free rolling of the ball so that the more they are screwed in the more must the cylinder tilt before the ball can roll over them.

This device is manufactured by the Golden-Anderson Valve Specialty Co., Pittsburgh, Penn.



SECTION THROUGH MOTOR CONTROLLER

The twenty-first annual revised edition of Hendricks Commercial Register of the United States for Buyers and Sellers has just been issued. This edition numbers 1574 pages and contains upward of 385,000 names and addresses.

The value of the Commercial Register for purchasing purposes is not confined to its complete classifiation alone; it also gives much information following the names of thousands of firms that is of great assistance to the buyer, and saves the expense of writing to a number of them concerning the article required.

The book will be expressed to any part of the country on receipt of $10 by the publishers, S. E. Hendricks Co., 74 Lafayette St., New York.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Fan Motors and their Control

By George J. Kirchgasser

In most fan applications it is necessary to be able to vary the speed to meet all conditions of service. Motors for either direct or alternating current can be used where speed variation is required, except the induction motor of the squirrel-cage type. This is satisfactory only where a constant speed is required.

To operate a constant-speed fan of small size it is only necessary to close the switch as shown in Fig. 1. If the motor is above ¼ hp., however, the switch in the fan circuit must first be closed and the motor brought up to speed by means of the starting rheostat. The connections for this purpose are shown in Fig. 2, and a rheostat for use with a direct-current motor is shown in

Fig. 1. Connection for Small Constant-speed Motor

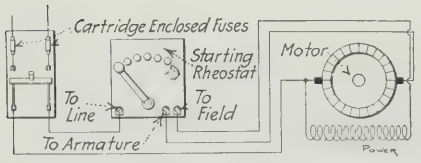


Fig. 2. Connections for Starting Medium-sized Motors

Fig. 3. This type of starter is designed only for accelerating the motor and its resistance is in circuit only during the acceleration period. The speed-regulating rheostat referred to later is similar in appearance but is differently designed.

By referring to Fig. 4 it will be noted that this illustration of a Cutler-Hammer speed regulator is similar to the starting rheostat shown in Fig. 3. The principal difference is in the design of the resistance. For the regulator the resistance must be capable of carrying the full load current for long periods while the resistance of the starting rheostat is only in service at intervals, when starting. To make the action of these regulating rheostats clear a reference to the characteristics of motors may be pertinent.

Where direct current is available a shunt or compound motor may be employed, although the shunt type is used in practically all cases. There are two ways of varying the speed of these motors; by inserting resistance in circuit with the armature of the motor and by inserting resistance in circuit with the shunt field winding.

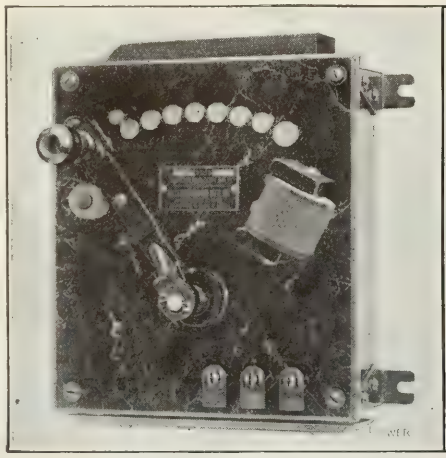


Fig. 3. Standard Direct-current Starting Rheostat

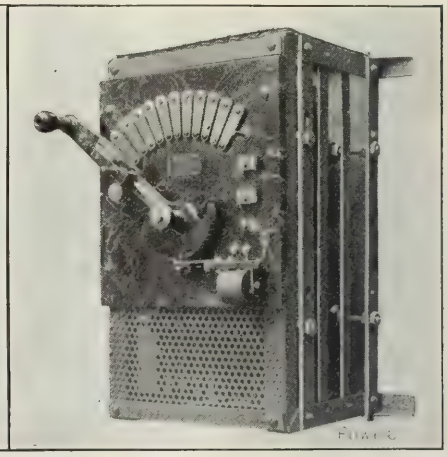


Fig. 4. Hand-operated Speed Regulator

Inserting resistance in the armature circuit by means of a regulator, such as shown in Fig. 4, reduces the speed of the motor in steps from its normal running speed to somewhere near one-half speed, usually. The amount of speed reduction depends upon the amount of resistance, the position at which the movable lever of the regulator is located and also to a certain extent upon the load the motor is driving.

When resistance is connected in various amounts in the shunt field circuit the current flowing in this field is decreased and causes the motor to increase in speed above normal, the percentage increase depending upon the amount of resistance used. Sometimes regulators are used which combine the two systems, that is, they permit of reducing the motor speed below normal by

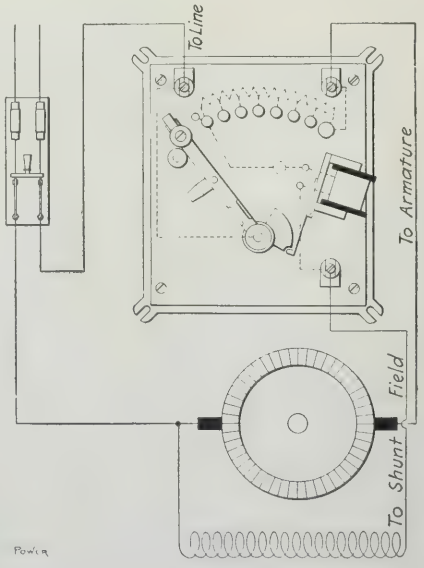


Fig. 5. Regulator Combining Field and Armature Resistance

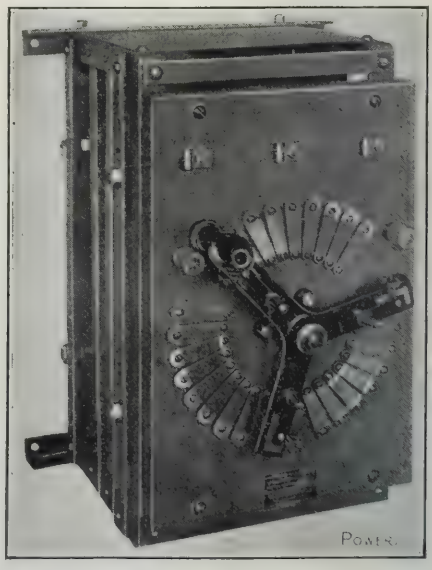


Fig. 6. Regulator for Three-phase Motors

means of armature resistance and provide for increasing the speed above normal by insertion of resistance in circuit with the shunt field of the motor. A regulator of this type is represented in Fig. 5.

Regulators used with motor-driven ventilating fans or for similar service are referred to as doing "fan duty." This is to distinguish from "machine-duty" regulators in which the resistance is differently proportioned. A "fan-duty" reg-

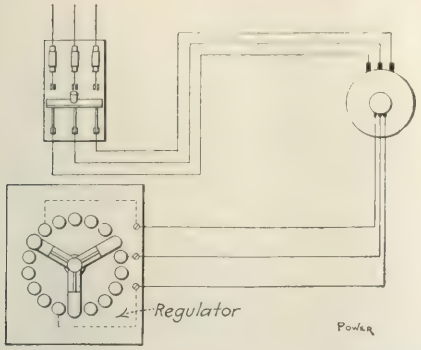


FIG. 7. CONNECTIONS FOR THREE-PHASE MOTOR REGULATORS

ulator is used in service where the torque decreases as the speed decreases, as is the case of a motor-driven ventilating fan. The current required when running at a speed less than the normal is less than the current required when running at the normal speed. For this reason the ventilating fan offers one of the best applications for control by means of an armature resistance type of regulator

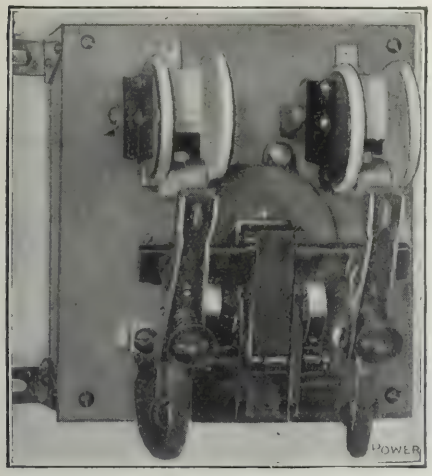


FIG. 8. MAGNETIC SWITCH STARTER

because at low speeds it is not necessary to dissipate a large part of the current in heating the resistance.

With many classes of service this is not the case, and full current is required at half speed, three-quarter speed or full speed, the motor running at the lower speeds simply because a part of the energy is dissipated in the form of heat.

Single-phase and polyphase motors are often used for driving fans, the former usually for the small sizes. Of the polyphase motors those of the squirrel-cage type are used only in cases where a constant speed is satisfactory. Where variations in speed from 50 per cent. below normal to normal are required and alternating-current supply is provided, the slip-ring type of motor with a special regulator is well suited. Fig. 6 shows such a regulator commonly known as the faceplate type, which has three sets of contacts. The steps of resistance for varying the speed can be inserted in circuit with the phase windings of the rotor, by means of a single lever operating the three sliding shoes. Equal amounts of resistance are cut in or out of circuit in the three phases (or two phases if a two-phase motor), thus maintaining a balanced running condition. The connections in Fig. 7 show how this resistance is cut in and out of circuit.

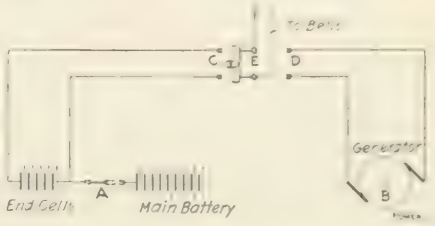
The drum-type regulators, similar in appearance to street-car controllers, are well adapted for larger fans, and are particularly suited for fan systems used in grain and malt elevators, in textile mills and wherever flying dust or other particles are present. Being entirely inclosed, nothing can come in contact with the current-carrying contacts.

In some fan installations it is desirable to control the starting and stopping, and sometimes the speed, from a point distant from the fan, such as another floor of the building. For the starting and stopping a magnetic switch starter, such as shown in Fig. 8, which can be actuated from push-button stations, is suitable. But in hotels, municipal buildings, schools, etc., where the various rooms are used at different times for different purposes, an automatic fan controller arranged so that from each room or group of rooms the speed of the fan can be changed, is of particular advantage. Control equipments giving these features are being installed to a large extent.

CORRESPONDENCE

End Cells for Bell Circuits

In plants having a storage-battery system, much annoyance and expense can be avoided by utilizing the end cells for operating elevator signals, call bells, annunciators, house phones, and any other apparatus requiring a low voltage. The number of cells that are required to make up the necessary voltage should be detached from the main battery as in the sketch. They should then be connected with the main battery by a single-pole switch of sufficient capacity to safely carry the full charging current. This detached group of cells is then connected to one side of a double-pole, double-throw switch as at *C*. The other side of the double-pole switch *D* is connected to a small motor-generator *B* and the middle point *E* to the main feeder of the low-voltage system. The object of the heavy single-pole switch between the end cells and the main battery is to prevent a high voltage backing up into the bell system should a ground occur.



BELL CIRCUIT ON BATTERY

When it is necessary to charge the end cells, the motor-generator is started and is connected with the bells by means of the double-throw switch. This consequently disconnects the end cells from the bell system.

The heavy single-pole switch at *A* can then be closed and charging can be continued as long as is necessary. The only object in having a motor-generator is to prevent interruption in the bell system, during the charging of the end cells.

FREDERICK HANSA.

New York City.

Blown Fuse

Mr. Piper's answer to Mr. Koppel's question which appeared in POWER for Nov. 5 is undoubtedly true, but the conditions stated by Mr. Koppel cannot be fulfilled in any three-phase system. Evidently the three voltage determinations were not made simultaneously or under identical conditions. This fact is easily proved by taking three space vectors, one of length equal to two, one of length equal to one, and one of zero length; upon trying to arrange these vectors in any manner it is readily seen that between three points the condition of relative potentials of two, one and zero is utterly impossible.

HENRY E. RANDALL, JR.

Boston, Mass.

Cleaning Battery Jars

Many engineers using storage batteries with glass jars have probably had considerable trouble keeping them free from spray and dirt.

When my jars get dirty, I wipe them off with a piece of waste moistened in kerosene. The oil softens the dirt and leaves the jars with a greasy surface, which will not allow the spray to stick. Care should be taken to get off all superfluous oil, as that would gather dust.

H. K. WILSON.

New Bedford, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

A Bituminous Gas Producer Plant

By A. A. Potter

The producer plant here described was installed in 1908 by the Smith Power Co., of Lexington, Ohio, as part of the laboratory equipment of the Kansas State Agricultural College. As this institution is far from the natural-gas fields of Kansas and was not near an artificial gas plant, the producer gas had to be adopted for use in open burners at the various laboratories.

Plant Equipment

The producer plant is shown in elevation and plan in Figs. 1 and 2 respectively. The essential parts of this plant are a gas generator *B*, condenser *H*, centrifugal wet scrubber *E*, centrifugal fan *K*, gas-pressure controlled reducing valve *Q*, tar-collecting tank *W*, dry filter or purifier *V*, gas holder (not shown) and pipe lines.

Details of the generator are shown in Fig. 3. It consists of a steel shell 6 ft. in diameter and 10 ft. high, lined with one course of firebrick and one course of ordinary brick, making a total thickness of about 9 in. The diameter of the fuel bed is 54 in., and, assuming a gasification of 10 lb. of coal per square foot of fuel area this producer is capable of gasifying 159 lb. of coal per hour and will generate about 10,000 cu.ft. of gas. The heat of combustion of the gas made is about 150 B.t.u. per cu.ft. (high value). Fuel is charged into the retort *C* and is admitted to the shell of the generator by means of a quick-opening gate valve. The retort *C* is provided with a water-sealed cover, this arrangement enabling the operator to charge the producer while the plant is in operation, without the danger of admitting air or of allowing gas to escape. Coal entering the shell is distributed by means of the hood *J'*, the inside of which serves as a gas collector; the gas outlet is at *J*. A swinging grate *Z* supports the fuel bed and is suspended from the shell by four chains, the shaking motion of the grate being produced by the hand lever *L*. Doors are provided at the bottom for the removal of ashes. The generator is also arranged with poke holes on top and with peep holes at the various zones, so that the temperature and condition of each zone may be determined. The inlet for the steam and air is shown at *A*.

Gas leaving the generator at the outlet *J* enters the condenser, which is shown in cross-section in Fig. 4. The condenser consists of a number of deflecting rings which throw the gas and water alternately from side to center, cooling the gas and condensing part of the volatiles.

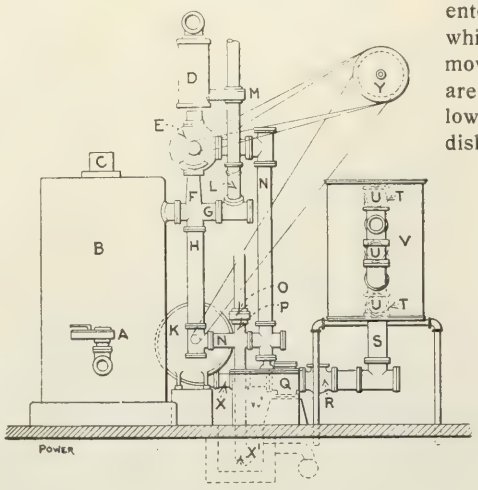


Fig. 1.

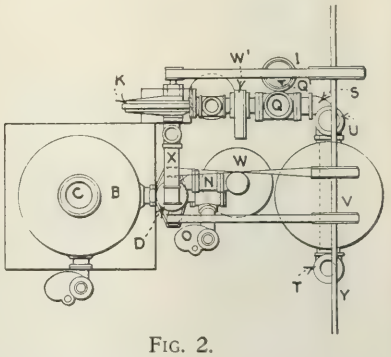


Fig. 2.

Elevation and Plan of Producer Installation

Referring again to Fig. 1, the gas, water and impurities, after leaving the condenser *H* are separated in the tee at *N'*; the gas is carried upward through the 6-in. pipe to the scrubber *E*, while the water and the condensed volatiles fall to the tar-collecting tank *W*. The water from this tank is allowed to pass off into the sewer, while the tar settles to the bottom and is drawn off.

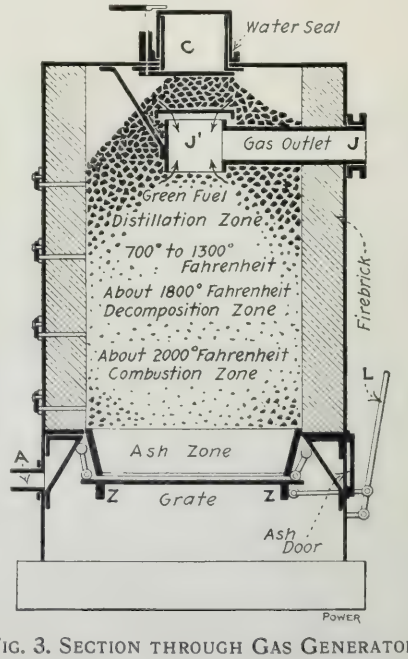


Fig. 3. Section through Gas Generator

The gas passing up the 6-in. pipe *N* enters the centrifugal wet scrubber *E* which is shown in detail in Fig. 5. The moving parts of the centrifugal scrubber are two disks *T* and *T'*, carried by hollow shafts. About the peripheries of these disks are blades or vanes ½ in. wide and spaced ¼ in. apart. The disks are of different diameters, so that one is inside the other, and are driven in opposite directions by open and crossed belts. The gas enters the center of the scrubber at *S'*, is thrown by centrifugal force to the circumference of the disks and in passing between the blades is intimately mixed with water which is admitted through the hollow shaft *R*.

Leaving the centrifugal scrubber the gas passes up to the separating drum *D* (see Fig. 1), where its velocity is decreased and part of its moisture is removed. It is then led through the pipe *X* to the centrifugal fan *K*. The fan *K* produces a suction on the fuel bed and also pushes the gas through the pipe *X*, the gas-pressure controlled reducing valve *Q*, the check valve *R*, and the pipe *S*. The function of the reducing valve *Q* is to regulate the pressure and amount of steam passing into the gas generator at *A*.

From the pipe *S* the gas is admitted to the purifier *V* through the three inlets *U*. This type of purifier, sometimes called a dry filter, is used for the separation of sulphur from the gas, and is shown in cross-section in Fig. 6. It consists of a steel shell having three inlets and two outlets. Inside of the tank are four wooden lattice-work shelves, supported on rings attached to the shell. Four-inch

layers of iron sponge are placed on each of these shelves. This iron sponge (iron oxide) is prepared by mixing iron filings, sawdust and sal ammoniac. The fan *K* forces the gas through the filter frames in the purifier; the gas leaving the purifier through the outlets *T* passes through the venturi-tube meter to the gas holders and mains.

FIG. 4. SECTION THROUGH CONDENSER

The fan and centrifugal scrubber are driven by a 13½-hp. electric motor. There is no boiler or vaporizer with this plant, the steam for the producer being supplied from the main boiler plant of the college.

Besides the above equipment, this plant is provided with a Junkers constant-pressure gas calorimeter for the determination of the heat of combustion of the gas made, scales for weighing the coal and ashes, a water meter for the measurement of the scrubber water, a steam meter for the determination of the steam used in the process, manometer tubes and thermometers at the various points in the plant, a Heraeus pyrometer for the determination of the temperatures in the various zones of the producer, and electric meters for the measurement of the power required to drive the centrifugal fan and wet scrubber.

CARE AND OPERATION OF PLANT

The general operating details can be seen by referring to the weekly log sheet. (Fig. 7.)

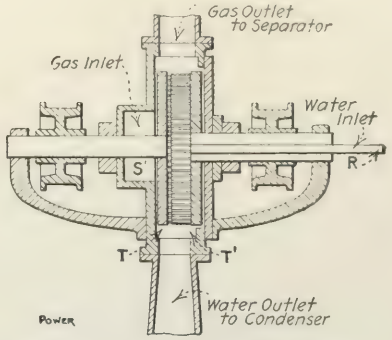


FIG. 5. CENTRIFUGAL SCRUBBER

In the operation of a gas-producer plant there are two processes, the blowing and the gas-generation process. Starting with the producer idle, the valve *M*, connecting the fuel bed with the atmosphere, will be found open. The first step is then to work the fuel bed with a poking bar, so as to settle the coal in the generator. Coal is then hoisted to a platform which is lowered by means of a hand lever on a weighing scale, and the total weight of coal required to fill the producer is recorded. A sample from each shipment of coal is taken and this is quartered down to an amount sufficient to fill a two-quart can and sent to the laboratory for analysis. After the generator is filled with coal, the water to the scrubbers is turned on, the fan is started, the air valve *A* and the purge valve *O* are opened, and the valve *M* is closed. This starts the blowing process, which requires 10 to 30 min., the ashes are then shaken down, the clinkers are pulled from the grate and the ash-pit cleaned. The electric meters are then read and the flow of water to the scrubber is adjusted to the most economical rate, which with this size of producer has been found to be about 1 cu.ft. per min. When the fuel bed appears hot as viewed through the peep holes of the generator, steam is turned on, and the purge valve is throttled down sufficiently to produce a slight pressure on the pilot lamps located at the fan outlet. If the gas flame at the pilot lamp indicates that the producer is generating a gas of the proper quality, the purge valve *O* is closed and the valve *Q* is opened, allowing the gas to pass through the dry

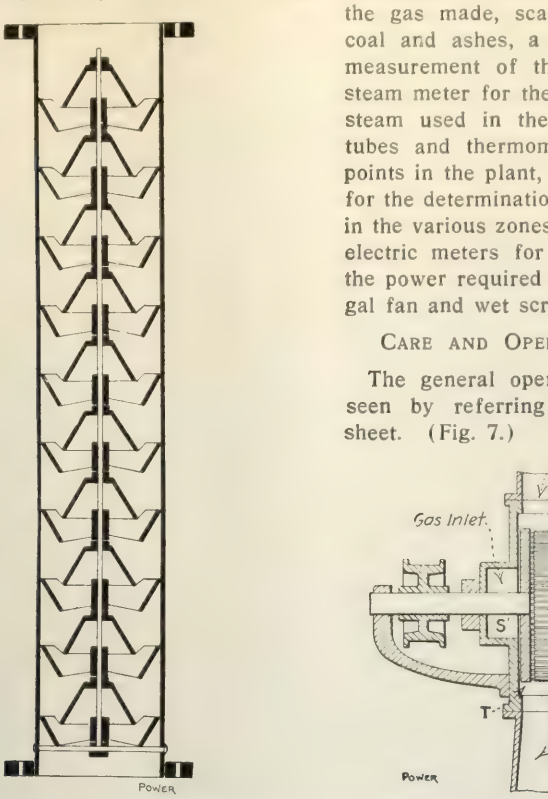


FIG. 6. SECTION THROUGH PURIFIER

WEEKLY LOG OF GAS PRODUCER _ RATED CAPACITY 100 HORSEPOWER

Run No.	Date	Gas Generation: Time From	To	Total	Current Volts	Amps	Kw.H.	Scrubber Water Meter Start	Stop	Cubic Feet	Steam Gage Pressure	Amount Pounds	Gas: Average Flow	Total Cubic Feet Generated (Corrected	Pressure	Calorific Value High	Low	Coal: Lot Number	Weight Pounds	Ashes Pounds
1																				
2																				
3																				
15																				
16																				
	Total																			
	Average																			

Run No.	Temperatures: Degrees Fahrenheit — Scrubber Water Initial	Final	Rise	Gas: Producer Outlet	Scrubber Outlet	Per Cent. Combustion Zone Producer	Heating Fuel Bed During Blowing Process: Time From	To	Total	Current Volts	Amperes	Kw. H.	Scrubber Water Meter Start	Stop	Cubic Feet	Remarks	Operator
1																	
2																	
15																	
16																	
	Total																
	Average																

POWER

FIG. 7. WEEKLY LOG SHEET

purifier and the venturi meter to the gas holders.

In stopping the plant the motor switch is opened, the steam is shut off, the valve at the purge pipe *O* opened, the valve *Q* leading to the purifier closed, the flow of water to the scrubbers is stopped and the natural-draft valve *M* opened. The air valve at *A* and purge-pipe valve *O* are then closed and the producer is left in this condition ready for starting.

To the foregoing directions for starting and stopping may be added—if the producer remains idle for any length of time it may be necessary to start a fire in the fuel bed. If the bed contains considerable fuel this can be best accomplished by using gas to ignite the coal. If the producer is to be started for the first time or the bed is thin, a fire can be started in the usual manner and the fuel bed built up.

The operator, besides following out the directions as outlined, records on the weekly log sheet (Fig. 7) the time required for the blowing process, the power necessary to drive the auxiliaries during that time in kilowatt-hours, and the water used in the scrubbers. When the blowing process is completed and the gas generated is admitted to the holder and pipe lines, the operator takes the various readings shown on the weekly log under "gas generation." This includes the time the producer is generating gas, the power required for driving the auxiliaries, in kilowatt-hours, the amount of cooling water used in the scrubbers, the steam necessary for the process, the rate of gas making, the cubic feet of gas made, and the gas pressure. Also one analysis of the gas is made and the temperatures of the gas in the combustion zone as well as at the producer and scrubber outlets are taken.

It has been found that the operator can attend to the operating details of the plant and have sufficient time to take the readings and keep up the records.

From the weekly records calculations are made of the cubic feet of gas generated per pound of coal, the cost of fuel, water, steam, electricity and labor per thousand cubic feet of gas and per thousand B.t.u.

These records and the results of the coal analyses constitute a mass of valuable information regarding the operation of this type of producer. It is the intention to publish these results in bulletins of the engineering-experiment station of the Kansas State Agricultural College. It has also been found that the taking, keeping and working up of complete records are conducive to better work on the part of the operator.

On this page Mr. McGouvy speaks of gas-engine builders who are willing to guarantee a horsepower-hour on 8300 B.t.u. We have yet to meet such reckless builders.

Gas Power Don'ts

By F. Webster Brady

Don't expect a horsepower from a half-pint of gasoline.

Don't buy an engine too small for your work.

Don't use galvanized-iron pipe for the oil connections.

Don't put a rubber gasket under the cylinder head.

Don't use baling wire for the battery connections.

Don't foul the vent hole in the gasoline tank.

Don't use butterine in the bearings.

Don't test the batteries by holding a screwdriver across the terminals.

Don't climb into the flywheel to start the engine.

Don't tramp on the oil pipes and igniter wires.

Don't forget to drain the water jacket the night before Xmas.

Don't kick up a dust about the carburetor.

Don't fail to strain all the gasoline.

Don't set the timer at "advance" before cranking.

Don't use rubber hose for gasoline connections.

Don't pour water into the carburetor.

Don't neglect the strainer in the oil pipe.

Don't expect air cooling to be automatic.

Don't crank with a push and with the right hand.

Don't cuss the engine 'cause it smokes.

Don't buy an engine without adjustable bearings.

Don't leave the engine under the rain spout when laying it away.

Don't trust an inexperienced man to operate your engine.

Don't dismantle the magneto simply to gratify an idle curiosity.

Don't operate without a muffler just to spite your neighbors.

Don't forget that not all engine pounds are caused by loose parts.

Cleaning Blast Furnace and Coke Oven Gases

In the *Proceedings* of the Société de l'Industrie Minérale, M. A. Gouvy emphasizes the importance of cleaning blast-furnace and coke-oven gases before they are used even for burning under boilers. Cleaning the gas so that it does not contain more than 0.5 gram per cu.m. (0.218 grains per cu.ft.) greatly increases the evaporative power of any boiler heated with it. With the unclean gas about 2 cu.m. are required per kilogram (32 cu.ft. per lb.) of water evaporated as compared with only 1 to 1.2 cu.m. per kg. (16 to 19 cu.ft. per lb.) with clean gas. The effect of cleaning on the evaporative efficiency is due to the fact that the dust is an excellent nonconductor and if present is deposited upon the heating surfaces, greatly impeding the transmission of heat through them to the water. In a Westphalian works where the cleaning plant reduces the dust contents to less than 0.25 gram per cu.m. (0.109 grain per cu.ft.) the consumption of gas is only 0.8 cu.m. per kg. (12.8 cu.ft. per lb.) of steam superheated to 250 deg. C. (482 deg. F.).

The dust deposit has another prejudicial effect in acting as a flux and making the firebrick waste rapidly. The cost of the cleaning should not exceed 0.027 franc per 100 cu.m. (0.15c. per 1000 cu.ft.), including all charges.

For engine running the gas should be further purified so as to bring its dust content below, as a maximum, 0.03 gram per cu.m. (0.0132 grain per cu.ft.) and in favorable conditions this figure has been reduced to 0.005 gram per cu.m. (0.002 grain per cu.ft.), in which case the engines do not require to be stopped for cleaning purposes any more than do steam engines. Gas-engine builders will generally guarantee a consumption not exceeding 2100 cal. (8300 B.t.u.) per i.hp.-hr., and in practice it may be taken that the consumption will not exceed 2800 cal. (11,100 B.t.u.) per b.hp.-hr., which with blast-furnace gas having a calorific value of 950 cal. per cu.m. (107 B.t.u. per cu.ft.) corresponds to a consumption of about 3 cu.m. (106 cu.ft.) of gas per b.hp.-hr. This is really a maximum value, but a low gas consumption is of secondary importance as compared with reliability.

Preheating for Internal Combustion Engines

On page 565 of the Oct. 15 issue an article appeared on "Preheating for Internal Combustion Engines." I do not want to dispute this matter for I have been working on this subject for several years. I have had patents pending in the United States Patent Office covering this field for some time. What my patents cover is: "The field of obtaining ignition of the fuel by means of the heat of compression added to an initial preheat." This preheat can be obtained by the heat of the exhaust or by an external heater, depending upon the type of the engine required.

I am not a wealthy man and, considering the expenses of experiments, etc., have not made as much progress as I should like to have made. However, I have worked quite a way beyond the scope of the article in question and I ask you to publish this so that if any of your readers are taken with this idea of preheat they will not infringe on my patents unwittingly.

John F. Wentworth.

Quincy, Mass.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Economic Operation of Ice Making Machinery by Use of Thermometers*

By V. R. H. Greene

In practical refrigeration the gases most commonly met with are CO_2 and ammonia. Ammonia is used more extensively because it can produce the refrigerating effect at a lower pressure for the same water temperature and because in the evaporation of one pound of the liquid to a gas a larger number of heat units can be absorbed. The cycle of the ammonia gas in an ordinary ice plant is as follows:

Ammonia gas is drawn from the ice-tank coils through the suction valve into the gas pump or compressor. On the return stroke of the piston the gas is compressed and discharged under pressure into the ammonia condenser, where the heat of liquefaction is removed and the ammonia gas reduced to liquid. This liquid is drained from the condenser pipes to a receiver, which is under the same pressure as the condenser, and on this account the ammonia gas remains in a liquid state. From the receiver the liquid flows to the freezing coils, but the supply is regulated by a delicately adjustable valve. Once in the tank coils under reduced pressure the liquid changes to a gas and in so doing absorbs from the brine nearly as much heat as was taken up by the water flowing over the ammonia condenser.

In any vapor, temperature of liquefaction and the corresponding pressure bear a fixed relation to each other, and the temperature of the ammonia in the freezing-tank coils depends upon the extent of reduction of pressure in the low-pressure side of the system. Just as it is necessary to have two different levels to make water flow from one vessel to another, so there must be two different temperatures to cause a transfer of heat from one body to another. In a freezing tank the most economical brine temperature that can be maintained is 12 to 14 deg. F., and to produce such a temperature the temperature of the ammonia gas in the freezing coils must be about 0 deg. F. This 12- to 14-deg. difference between gas and brine is required to cause the necessary heat transfer from the brine to the liquid ammonia in the coils.

*Abstract of paper read before the Eastern Ice Association, New York City.

Any table giving the properties of ammonia gas will show that an ammonia temperature of 0 deg. F. corresponds to a pressure of approximately 15-lb. gage. This is why a back pressure of 15 lb. is most commonly met with in ice-making practice.

Refrigerating effect is produced wholly by the evaporation of a given number of pounds of liquid ammonia into a vapor. This vapor is then drawn into the compressor and compressed and discharged into the condenser. As there are more pounds of ammonia contained in a cubic foot of its vapor under a high pressure than under a low pressure, it is advisable for the economical operation of the plant to carry the suction pressure at the tank as high as possible consistent with the brine temperature required. With a high back pressure the more closely are the gas molecules compressed, and, therefore, the more pounds of ammonia are introduced into the compressor cylinder and discharged into the condenser per cubic foot of vapor handled by the compressor.

However, a high suction pressure means a high suction-gas temperature, and this tends to reduce the degrees difference between the brine and gas, resulting in a higher brine temperature. As it is uneconomical to make ice with a brine temperature much higher than 14 deg. F., owing to the length of time required to freeze a standard ice can, some method must be devised to keep the brine bath at 14 deg., and at the same time make it possible to run with as high a back pressure or suction temperature as possible. It is known that a greater rate of heat transfer is obtained between two liquids or gases in motion than at rest, and to help increase this rate freezing tanks are provided with agitators.

The first step, then, toward freezing-tank efficiency is to operate the tank agitator as fast as possible without causing the brine to slop into the ice cans. The next step is to place thermometers in the suction pipe near the entrance into the compressor cylinder and find the smallest degree difference which it is possible to carry and maintain a brine temperature of 12 to 14 deg. F. When this temperature is found, regulate the operation of the plant thereafter by the thermometer, paying attention to the expansion valve regulation only to maintain this suction temperature at the machine. If the temperature is right the suction pressure must be right, for unless the gas is superheated it will correspond to that pressure which is equivalent to the suction temperature maintained.

There is another loss which the thermometer on the suction eliminates. As previously stated, under a pressure of 15 lb. saturated ammonia vapor has a temperature of 0 deg. F. It is possible, however, to maintain a pressure of 15 lb. at the freezing tank and then allow the suction gas, after it leaves the tank on its way to the compressor, to pass through hot engine rooms in uninsulated pipes and become heated up away beyond the temperature corresponding to its pressure. When this happens the gas expands under heat, just as do other substances, and becomes rarefied, so that for every cubic foot of gas pumped by the compressor very few pounds of ammonia are delivered to the condenser per cubic foot of gas pumped. This results in an overspeeding of the ammonia compressor in proportion to the superheat in order to discharge the required number of pounds of ammonia into the condenser. By regulating the temperature of the gas at the point of entrance to the compressor, this loss is eliminated.

The only possible loss of efficiency that can result from thermometer operation in the suction side of the machine, is caused by carrying too much liquid ammonia back from the freezing tank into the compressor cylinder, making a semi-liquid pump instead of a pure-gas pump. When this is done the compressor displacement is spent upon removing the liquid which has become evaporated into a vapor in the compressor itself, and not the vapor which has done useful refrigerating work in keeping down the brine temperature. This will show at once a capacity loss, which can be discovered by the drop in temperature of the discharge gases to the condenser. No liquid ammonia should be allowed to enter a compressor cylinder.

When vapor containing no liquid ammonia is compressed from a suction pressure of 15 lb. to a condenser pressure of 185 lb., which is common in practice, a temperature of about 250 deg. F. is created. This temperature is based only upon the vapor entering the compressor at the temperature corresponding to the back pressure of 15 lb., which is 0 deg. F. If the thermometer is placed in the discharge pipe near the compressor and reads about 250 deg. F., it will indicate that the entering vapor temperature

is the right one, and that no liquid is present in the compressor. Should the discharge temperature go up to 270 deg. F., it will be known at once that the suction gas is entering the compressor at a temperature above zero, and is therefore superheated and rarefied. This shows that the expansion valves on the freezing tank are becoming clogged and not enough liquid is being fed into the tank coils. If, on the other hand, the discharge temperature falls below 250 deg. F., too much liquid is passing into the coils, meaning that the compressor is becoming a liquid and not a gas pump, and that this flow must be checked. It will then be necessary to pinch down upon the expansion valve which appears to be feeding in excess of all the others, until the proper discharge temperature is reached.

In a 30-ton plant which seldom made over 20 tons a day, the compressor was overhauled, new valves installed and excessive expense entailed with no increase in output. When a thermometer was placed in the discharge line at the compressor a temperature of only 104 deg. F. was registered, proving excessive ammonia fed into the tank coils. The expansion valves were closed down until the discharge temperature rose to 240 deg. F., which proved to be the best temperature, and from that time on 32 tons of ice per day have been pulled from this plant. Nothing more was done to the machine than to raise the discharge to its proper point for efficient operation.

From the gas compressor the ammonia is discharged to the condenser, which usually consists of a series of 2-in. pipe coils into which gas enters at the bottom and passes upward in counter-current flow to the cooling water, which flows downward over the outside of the pipes. Just as there was required a difference in temperature between the ammonia gas and the brine in the freezing-tank coils to produce the necessary flow of heat, so in the ammonia condenser the water which is showered over the outside must be cooler than the condensing vapor on the inside. In average practice this difference is between 10 and 15 deg. F., so that with an initial water temperature of 70 deg. F. the liquid might be expected to have a temperature of 80 to 86 deg. F., depending upon the type and efficiency of the ammonia condenser.

The same ammonia table which showed that for a back pressure of 15 lb. the saturated gas temperature was 0 deg. F., also shows that for an ammonia temperature of 80 deg. F. a condenser pressure of 138 lb. must be expected, or for a temperature of 85 deg. F., 152 lb. If the actual head pressure as registered by the gage is higher than that corresponding with the liquid temperature, provided, of course, that the gage reading is correct, it will be known that the system is foul with noncondensable gases which should be removed at once so as to reduce the head pressure to the proper point and thus reduce the horsepower required to compress the gas. In an efficient design of ammonia condenser, particularly one in which a strictly counter-current principle is involved, the water heats up as it passes over the ammonia coils, and the temperature of the outlet water corresponds closely with the liquid temperature. When this is the case, it is only necessary to place a thermometer in the overflow pipe in the condenser pan to learn the temperature of the liquid ammonia.

Of late manufacturers have placed on the suction and condenser pressure gages, directly over the pressure dial., a series of red figures giving the corresponding temperatures. This does away with the necessity of referring continually to the ammonia tables. It is only necessary to see that the outlet water from the condenser pan checks up with the red figures over the pressure-gage line for the correct head pressure, and that the temperature corresponding to the suction pressure on the gage checks with the suction thermometer at the cylinder of the compressor.

Refrigerating Problems

One of our readers in Suva, Fiji Islands, recently asked the following questions, which with their answers may be of general interest:

1. In cold-storage rooms, what are the practical objections, if any, to the perforated floor and ceiling system of circulation? Where the air is cooled by a battery of coils, what is the best arrangement to avoid contamination of the goods stored?

2. For a daily output of three tons of ice, what would be the approximate size of a brine tank, the cans being 8x15x28 in. and the bath temperature 25 deg. F., with a forecooler in use?

3. What would be the approximate size of a coil room 7 ft. high and of an ammonia compressor to maintain 10,500 cu.ft. of space at an average temperature of 25 deg. with the atmosphere at 85 deg. F. and circulating water 70 deg. F., assuming the insulation to be fairly good?

1. Perforated floors and ceilings for circulating the air in cold-storage rooms are very rarely used in modern construction for these reasons: Perforations in the floor weaken it and good circulation of air is only secured when the perforations are so placed that the goods cannot cover them. It is also difficult to proportion the number and sizes of perforations to obtain an even distribution of air throughout the room. Perforations in the floor are also undesirable on account of the dirt which falls through them and collects in the air ducts. Whenever refrigeration is secured in cold-storage rooms by mechanically circulating the air from a central battery of cooling coils, the air is purified at the point where it is cooled, as it deposits its moisture there and the condensed water vapor contains most of the foul gases in solution. Where the "Linde" system is used for cooling, the brine flowing over the outside of the cooling coils purifies the air, but in this case sometimes the calcium is carried along with the air and deposits on the goods that are stored in open crates or are loose. If the air is properly divided during cooling, either the dry-coil or wet-coil method of cooling should purify the air sufficiently for cold-storage purposes. If it does not, any amount of filtering will not improve its quality.

2. A brine tank to hold a sufficient number of 100-lb. cans, 8x15x28 in., to make three tons of ice per day when operated 24 hr., the temperature of brine being 25 deg. F., will be about 10 ft. wide, 24 ft. long and 30 in. high, inside measurements. There will be required about 150 ice cans of the size above mentioned and in the tank should be placed not less than 300 ft. of 1¼-in. pipe per ton of ice-making capacity. To obtain this output with the above tank, the brine must be properly agitated by means of a propeller placed at one end of the tank and the tank partitioned lengthwise through the center to facilitate circulation.

3. A cold-storage room of 10,500 cu.ft. volume lined with an insulation permitting not more than 3 B.t.u. heat transfer per square foot per degree Fahrenheit difference of temperature per 24 hr. will require 2½ tons actual refrigeration to maintain it at 25 deg. with a temperature of 85 deg. F. on the outside. In this calculation no refrigeration is included for materials to be cooled or frozen in the room as no quantities are given. For determining the number of feet of 2-in. pipe necessary in a coil battery to refrigerate this room by cold air, mechanically circulated, it can be assumed that 100 ft. of 2-in. pipe will take care of one ton of refrigeration provided the air passes the pipes of the cooling coil at a speed of about 1000 ft. per min. For 2½ tons of refrigeration, 250 ft. of 2-in. pipes will be necessary under the above conditions. Two cooling coils, each containing the required length of 2-in. pipe should be installed when the plant is continually operated so as to allow for defrosting one coil when the other is in use. To the above must be added the refrigeration necessary to cool and freeze whatever goods are to be stored in the room at 25 deg. F., if they are not already cooled and frozen.

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POWER

Pipe Line Inspection

A factor, closely allied with power-plant safety, that has been ignored is the inspection of steam pipe lines.

Boiler inspection is maintained in several states and in many cities, both in compliance with an inspection law, and also by insurance companies as a business. Why then stop with the boiler? Why not know that the steam lines are correctly designed, installed and cared for?

It is true that fewer accidents occur from steam pipe than from boiler failures, but there are more of them than would occur if the piping were put up right and kept in condition.

When a power plant is installed, in some instances the boilers are inspected before leaving the shop, by a representative of the purchaser, to guard against inferior material and workmanship. They are again inspected before a boiler-insurance company will issue a policy covering the risk against explosion. This is as far as the inspection goes. The steam piping can be put up in any manner that suits the designer or contractor. It may contain pockets, dangerous bends and cast-iron fittings unprotected by suitable means of caring for expansion and contraction strains.

When such piping is once in place it is there to stay until an accident occurs. It may be the cracking of a fitting, the fracture of a blank flange that is altogether too light, or it may be the pulling apart of a screwed joint. Anyone of these could be avoided if the pipe work were under the eye of an inspector. Starting right and keeping right means a lot in any work, and especially in power-plant construction.

A second demand for inspection occurs where the piping was originally designed for a pressure of, say, one hundred pounds, and which is later forced to carry a pressure of a hundred and fifty pounds. This is the same as using standard weight pipe instead of extra-heavy pipe; it is not an uncommon practice.

There are engineers operating plants, the high-pressure piping of which they know to be unsafe, because it was never designed to carry the pressure that has been forced upon it.

It is seldom that a piece of pipe line is removed because it is considered unsafe. Joints may be found leaking badly, the drainage may be faulty or none at all provided, and the means of suspension may be such that the pipe and fittings are always under a strain, yet these conditions will generally be allowed to continue.

Because steam pipe lines are small in diameter and are considered safe for any pressure to which they will ever be subjected, is no reason for confining the inspection to the boilers, and losing sight of the dangers existing beyond the stop valve. There is no excuse for neglecting the inspection of steam lines at intervals. If steam-piping inspection were made a part of the power-plant routine fewer accidents would occur, and if this had been the practice aforetime many men now dead would be living.

Reporting Plant Conditions

Under the stress of operating service a change in shifts in the power house sometimes represents a critical period in the daily routine. As one set of men lay down their tasks and another set assumes them, it is clearly desirable to make the transition as smooth as possible and to leave no loose ends or forgotten duties to cause subsequent trouble. Verbal instructions and suggestions are obviously of advantage to the incoming shift, but sometimes these are neglected or hurried over to the detriment of the service. The station log is usually the main reliance of men going on duty, in the absence of specific information from the outgoing force, but some plant managers have recognized the need of a definite statement of conditions from those leaving the work and have prepared forms for this purpose which are fulfilling a highly useful function.

The information included covers in a

representative case a day report and a night report signed by the watch engineer in each case, and the data set forth include such questions as—hour at which watch engineer assumed charge; anything reported by predecessor as being out of order; anything reported as wrong during previous shift, with any stops required; whether gage-glasses were tested; plentifulness of coal supply; disposal of ashes during shift; condition of water supply, engines, generators, pumps and stokers; violations of orders during shift; condition of piping and connections to boilers; whether boilers and tubes were blown off at the proper time.

A similar but much more abbreviated report is prepared by the engineer in charge of the night watch. The regular station log of hourly and other readings is not interfered with by the report of plant condition thus required, and the latter is beneficial in fixing the responsibility for the handling of the plant at stated hours, besides serving as a memorandum of emergency work which should not be allowed to run over into a later shift's period. Thus the incoming force is in position to meet the service conditions most promptly and efficiently.

Such records are of the greatest value as evidence of the care with which a plant is operated, and if the manager has occasion to use them in a court case, for example, the slight cost of preparing them becomes as nothing in proportion to their helpfulness.

Climbing the Ladder

Those with an ambition to succeed and reach the top of the ladder in their chosen work, should learn early to obey instructions. It is human nature to regard one's own ideas on how to manage or to do given things better than those of the one in authority, and when he orders a thing done a certain way the tendency is for the subordinate to want to substitute another way.

He fails to realize that probably his chief devoted some time thinking and planning this particular thing, before even mentioning it to him, and has looked at it from all angles. Very likely the subordinate's plan was rejected along with a dozen others, before the final one was adopted.

It is not the intent to discourage anyone from having ideas. They may be good ones and very valuable and acceptable if advanced at the proper time and in the proper way. But no matter how good the ideas may seem to their originator, he should not deliberately go contrary to his orders, even in the smallest detail.

As a rule it is a characteristic of the really successful that they do exactly as they are told, no matter whether their chief is a strawboss, a foreman, a superintendent, or a board of directors. In fact, the higher a man goes the more important it is that he obeys his orders to the very letter.

Contrary to the plaint (usually from the lazy) that the good places are all taken, there never was a greater demand for men who can carry out the plans of the "men higher up." Almost invariably the latter became what they are by first obeying the instructions of others. They are exemplifications of the old adage, "Before you can command you must learn to obey."

There is no line of work in which this is more true than in the power-plant field, nor are the opportunities to rise to the top more numerous or more rapid elsewhere.

The Average Owner

Conditions in the engine and boiler room of the small isolated plant are often not worth boasting about. In fact, the farther the visitor stays from the power plant, the greater peace of mind he will usually enjoy. The equipment is sometimes second-hand and very often in a dilapidated condition. Some of it may even be in a dangerous operating condition. Everything, including the starter-and-stopper, points to the fact that the power plant is considered only as a necessary evil and, instead of taking its place as a valuable adjunct to a prosperous business, it is only endured because it is impossible to run the business without it.

The owner of a small but prosperous and growing manufacturing plant in the Middle West recently informed his visitor that if his factory were located in a city where central-station power was available, he would not take a second thought before he would do away with his troublesome power plant altogether and purchase power from the central station. In the next breath he asked whether, in the opinion of his visitor, an open feed-water heater would have any effect in reducing the amount of scale formed in the boiler. It is apparent, therefore, that this man wished to be informed, but that his mind was so taken up with the cares of his business that he had little opportunity to study the needs of his power department.

Later an opportunity was given the visitor to meet the engineer and fireman. He was informed that this man's duties were to start and stop the engine, blow the whistle, put coal in the furnace, clean out the boiler of scale on Sunday, sweep up the shop, run the drill press when required and be general handy man. His wages were fifteen dollars per week.

This is not an exaggerated case. It is the picture of the average owner of the small plant and the average steam engineer. Both need awakening. The average owner is open to conviction if his engineer can tell him with the authority that comes from knowledge the advantages that will be gained from much needed improvements. For example, it would only be necessary to put the case properly to prove to the owner that his power would cost him practically nothing if the exhaust steam from the engine were used for heating the shop rather than being allowed to waste into the atmosphere. This owner needed only to be convinced that a feed-water heater would be a profitable investment for his money before installing one.

A general toning up of the power plant would react on the engineer and, instead of being a roustabout, his duties in the power department would be so important as to require all of his time and attention.

There are a thousand and more power plants in the country similar to this one. The average owner is so busy making his factory pay that he neglects the power department. If he would but stop to consider, he would realize that the expense for his power has its share of influence upon his profits and that a little better engineer at a higher salary may save many times his extra cost by increasing the plant economy.

It is two hundred years ago this year that the first successful reciprocating steam engine was erected by Newcomen and Savery. Where would the world have been today if this had never happened?

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Repairs to a Balanced Piston

The balance piston, Fig. 1. was supposed to help carry the weight of the piston valve and valve-gear of the 38x54-in. high-pressure cylinder of a marine engine. The balance piston was found in the storeroom. evidently having been condemned by some former engineer. Indicator diagrams and a little calculation showed that the piston had too much area for the work required of it. The weight of the valve-gear is about 1350 lb. and

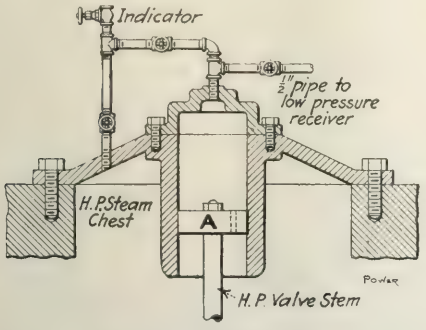


FIG. 1. BALANCE PISTON FOR CARRYING WEIGHT OF VALVE GEAR

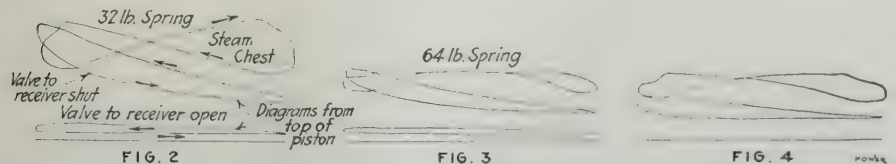
the diameter of the balance piston is 6 in. Fifty pounds per square inch would easily support this weight. but the boiler pressure of 100 lb. was applied. and as the top of the balance cylinder was connected by a pipe to the condenser, the pressure might increase beyond this amount. The pipes shown were connected to apply the indicator, and the pipe to the condenser was changed to connect to the low-pressure receiver, which carries a pressure of about 10 lb.

As the bottom of the balance cylinder is open to the main steam chest, the problem was to form enough back pressure on the top of the piston to reduce the mean effective pressure to the proper amount that would just balance the weight of the valve-gear. Diagram, Fig 2, was the first one taken, and it quickly explained the violent hammering which followed the attempt to use the whole power of the balance piston. There is considerable difference between the maximum and minimum pressures in every revolution, caused by fluctuations of pressure in the steam chest by the main throttle being partly closed. However. the noise was easily eliminated by closing an outlet to the receiver, but as will be seen from Fig. 2, the variation in pressure became greater by doing so, the steam on top of the piston being compressed on the up-stroke for lack of a free outlet. Should the valve to the receiver be closed altogether, this pressure would even exceed that in the steam chest, as shown by the shaded part of the diagram near the top center.

Fig. 3 shows that live steam was admitted to increase the back pressure. After some experimenting it was found that a ¼-in. hole *A*, Fig. 1, drilled through the piston supplied enough steam to maintain this back pressure while keeping the valve to the receiver open. Fig. 4 shows the final result. With this arrangement the piston has a mean lifting effect of about 1000 lb. without at any time taking the whole weight off the eccentric.

O. OLSEN.

New York City.



DIAGRAMS TAKEN BEFORE AND AFTER CORRECTING THE TROUBLE

Firing Boiler Furnaces

The following is a summary of my observations of methods of firing boilers worked at only a small fraction of their rated capacity. Boilers worked in this manner are very common in electric-light plants in small towns, especially plants supplying a day circuit. A few suggestions are also given.

In one town of less than 2000 inhabitants the load is so light for extended periods during the day that it scarcely shows on the ammeters. This plant is equipped with a first-class Corliss engine and two 66-in. by 16-ft. boilers, all in excellent condition. Only one boiler is in use at a time and it is always run with the ashpit doors and the damper wide open, regardless of the load that is being carried, the engineer claiming it will waste coal to partly close the dampers. In observing his method of firing, I found that the front part of the grates was well covered but the rear part was much neglected. Sometimes the interval between coal charges was so long that the fire became very thin, thereby allowing a large amount of cold air to pass through the grates. This is decidedly bad practice, but no trouble was experienced in keeping up steam due to the small amount of steam that happened to be demanded.

Whether the fire is thick or thin the grates should be well covered with fuel as any holes in the fire, and more especially those near the bridge-wall, are not conducive to good combustion or fuel economy. In the above case, with the light demand for steam, if the grates were kept properly covered with a fire of medium thickness and the dampers and ashpit doors were wide open it would be impossible to keep the boiler from blowing off, but the cooling effect of the air passing through the grates avoided this at the expense of the coal pile. Better results would be obtained by running a fire of medium thickness, paying particular attention to that part of the fire at the rear; the front part is usually taken care of pretty well, as most firemen are inclined to devote more attention to that part.

The fire should be given good draft for a few minutes immediately after firing, so as to give the fresh coal a good start and insure good combustion. The damper should then be partly closed; it will usually be found that quite a small opening at the damper will give very good results. This method will enable one to keep the grates well covered and also allow of a reasonable time between fires.

The previous paragraph may appear to indicate that I am not in favor of good draft. With a good strong draft and the proper amount of grate surface, the best fuel economy can be realized, but many boilers have a large excess of grate surface for the load that they are required to carry.

In a certain small town a new electric-light plant was started with a 100-hp. boiler supplying steam to a Corliss engine, but the peak load did not reach 50 hp.; most of the time the load was little more than the friction load of the engine. The rear half of the furnace grates was bricked over, resulting in a marked increase in fuel economy. This is good when practicable, but some similar plant may run along all day with little or nothing and have a peak load that will crowd

the boiler to its fullest capacity and then every square inch of grate surface is needed.

One capable engineer experimented with covering part of the grate surface with a slab of cast iron which was to be removed when the load came on. The experiment was not successful as difficulty was experienced in removing the slab because the slag and clinker from the coal had almost welded it to the grate bars.

One very good practice for such load conditions is to keep a very heavy bed of coal on the rear part of the grates, thereby effectively closing that part, and to do the firing on the front end. In this way the maximum draft can be used, and it has an advantage in that the fire at the rear can be broken, making the entire surface available at a moment's notice.

Another means for cutting down the grate surface, and still have it all available for peak loads, is to run a firebrick wall on the grates, lengthwise of the boiler, extending from between the fire-doors to the bridge-wall. This wall need not be over two or three bricks high and not more than one brick wide; a similar wall is constructed in the ashpit, only it should extend from the floor to the under side of the grates. By this arrangement the furnace will be divided into two distinct compartments. Only one side is used at a time as the other side is kept banked. It will be readily seen that with the ashpit door shut on the banked side there will be very little burning on that side, and in practice this plan has proved very satisfactory. When it is necessary to clean the fire or to use the entire furnace, the ashpit door on the banked side can be opened, the bank broken and in a short time the fire will be as hot as required. One advantage of this practice over that of keeping the rear end of the grates banked is that it admits of easily inspecting the fire; there is little excuse for permitting the banked fire to burn out. There may be some objection to having all the fire on one side during the light-load period but in practice this does not appear to cause any trouble.

It is so easy to maintain the necessary steam during these light-load periods, that the engineer is inclined to neglect to give the fire the proper attention. Much has been said and written about the detrimental effect of the infiltration of air through the cracks and crevices of the brickwork of a boiler setting. This is a good point and deserves attention, but more attention should be paid to the excessive amount of air that finds its way through the boiler flues by way of the holes in the fire.

The effect of excessive air passing through the fire was effectively illustrated to me during a recent visit to a large plant. This plant was equipped with steam flow meters on each boiler; at the load then being carried each meter indicated a steam flow of 10,000 lb. per hr. One of the boilers was not working properly and the flow dropped to about 2500 lb. These boilers were all equipped with chain grates, and investigation revealed that about 10 in. of grate surface was not getting coal, due to an obstruction in the coal hopper. This bare space extended the length of the grate, which was about 10 ft. As soon as the obstruction was removed the flow quickly came up to normal. This was no exceptional case and it would be difficult to tell how often it may have happened before, but with the indicating instruments it was at once apparent. In this case the turbines were getting all the steam they needed and no difficulty was experienced in maintaining the pressure. This condition may have continued for a number of hours and all that time this furnace was using nearly as much coal as the others with a very much smaller output.

G. S. SPRAGUE.

Geneva, Neb.

Stopping Leaky Cylinder Head Joints

A 20 and 38 by 42-in. compound engine developed a leak in the front head of the high-pressure cylinder, and as it meant much work and time to disconnect the parts and piping, loosen the foundation bolts, and move the cylinder back to put in a gasket, we decided to make a rather novel repair. The chief suggested that we remove the piston and put a strand or two of packing and some fine copper wire in the 0.025-space between the counterbore and the head and then caulk it tight. This was done and the engine was then put on the head end center and steam at 145 lb. pressure admitted; no leaks were noticeable.

The tool used for caulking was a 12-in. hacksaw blade with the teeth ground off, inserted in a plug and firmly secured by pins. This tool was driven by a small air hammer.

FRANK BUTTERWORTH.

Utica, N. Y.

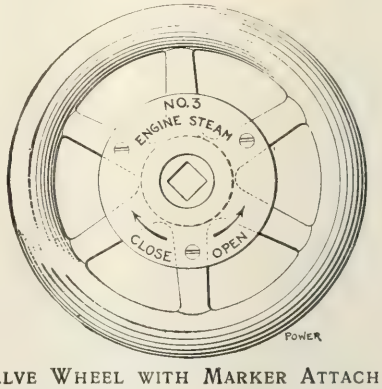
Indentification Markers for Valves

In large power or pumping plants where innumerable valves are used in connection with air, gas, steam, water and exhaust-steam lines, there is always chance, especially when there are many valves close together, that in a sudden emergency the wrong valve may be closed or opened.

This might lead to a serious accident, to both property and human life. To lessen this danger, throughout the plant I have charge of, I have installed safety markers on all the valve wheels.

The marker is in one piece, made of brass with the letters cast on. The letters have been buffed and polished while the background is painted dark blue to set off the letters.

Since some valves have rising stems while others have stationary stems, I designed the marker so that it would have a round hole in the center through which the stem could rise. The marker is securely fastened to the hub or spokes of



VALVE WHEEL WITH MARKER ATTACHED

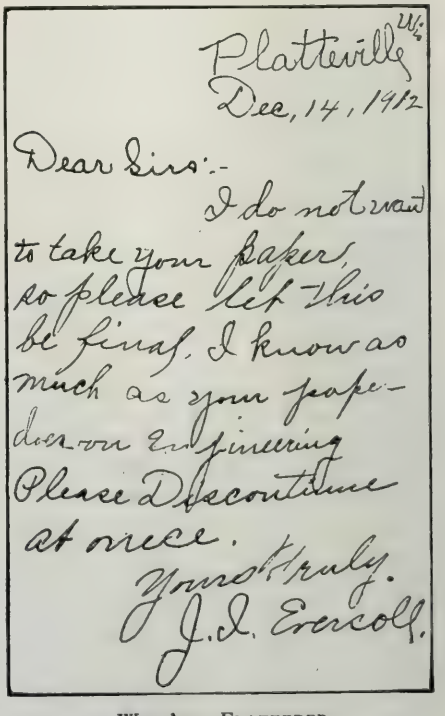
the wheel with countersunk machine screws.

In addition to the wording on the brass plate as to what the valve is for, I have also put on arrow marks, showing which way to open or close the valve. The sketch shows how this marker looks attached to a wheel.

A. RAUCH.

Swissvale, Penn.

A Reader with Something to Say



Platteville Wis.
Dec, 14, 1912

Dear Sirs:-
I do not want to take your paper, so please let this be final. I know as much as your paper does on Engineering Please Discontinue at once.

Yours truly.
J. D. Evercoll.

WE ARE FLATTERED

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Cause of Explosion in Air Line

Referring to H. H. Delbert's trouble and inquiry in the Oct. 29 issue, I would suggest that, first, he use the best grade of air-compressor oil obtainable for lubricating the cylinders of his compressors, and, second, place an oil separator in the discharge line between the compressors and the air receivers, which will separate most of the oil that is being carried over into the receiver with his present arrangement. A separator similar to that used in ice-making and refrigerating plants for separating oil from the discharge, or high-pressure gas from ammonia compressors, would be very effective for this purpose.

The separator should have a gage-glass to show the amount of oil in it, and so indicate when it should be drawn off. Also, the separator should be placed as far from the compressors as possible to allow the vaporized oil time to condense so it will easily separate from the air. It is important that the separator should be large enough to allow the air to pass over the baffle-plates slowly to effect a more complete separation of the oil from the air.

I would also suggest that it is well to place drain elbows at all low points in the air-distributing mains and branches for drawing off water where necessary.

THOMAS BAKER.

York, Penn.

H. H. Delbert asks what caused the explosion in the discharge pipe of the air compressor and requests suggestions as to how to prevent a recurrence of this trouble. The receiver of an air compressor should be large, and placed as close to the compressor as possible, not over 20 ft. away. One chief function of an air receiver is to reduce the pulsations of the compressor, which are similar to water-hammer in a water line, and the greater the distance between the compressor and the receiver, the greater the shock due to the pulsations.

I was called upon to remedy a trouble in a certain plant, where the receiver was 300 ft. from the compressor, and the noise due to the pulsations of the air entering the receiver, sounded similar to striking the receiver with a sledge hammer. A number of fittings on this line had been broken. The trouble was eliminated by placing the receiver close to the compressor.

A second function of an air receiver is to cool off the air as soon as possible after it leaves the compressor and before it enters the pipe lines. In cooling off, the air gives up moisture, and the less moisture that enters the air lines the better. The air, in expanding through the different machines, cools in proportion to the expansion, and any moisture in the air freezes and chokes the exhaust ports. The only remedy then is to reheat the air as it enters the different machines, so that the larger the cooling system, the less water will go into the pipe lines.

Another precaution is to use a cylinder oil made purposely for the work. Poor oil has been the cause of a number of receiver explosions. If the compressors are properly designed and set up, and if Mr. Delbert will follow these instructions, he will have no further trouble with explosions in the discharge pipe.

DAVID BILLSON.

North Chelmsford, Mass.

Replying to Mr. Delbert's inquiry, I would suggest that one or more of the air-compressor valves leaked, causing the air to be compressed over and over until it became of such a high temperature that the carbon deposit ignited, and in burning heated the fitting until it could not hold the internal pressure and so let go. I would suggest that Mr. Delbert put a fusible plug that will stand 400 deg., and a thermometer, such as is used in economizers, in the discharge line. If the trouble occurs again, the fusible plug should prevent an explosion.

E. A. YOUNG.

Isabella, Tenn.

More About Siphon Condenser Troubles

Mr. Marier's letter on siphon-condenser troubles, in the Oct. 22 issue, reminds me of my experience with them.

Some years ago I took charge of a plant where there was a 20-in. siphon condenser, the water supply for which was taken from a canal leading to a large waterwheel. The injection water-supply pipe entered the canal directly in front of the waterwheel and back of a grating which was placed in the canal to catch the foreign matter in the water. This arrangement was the source of various troubles.

In the spring when the warm rains melted the snow and raised the water level in the river, much dead grass and other material would be brought down and lodge on the strainer, seriously interfering with the water supply to the condenser, making it necessary to close the bulkhead gates and draw down the canal through the waterwheel to clean the strainer.

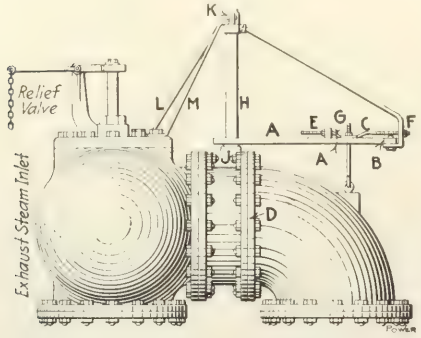
This sometimes had to be done twice a day, and it was frequently necessary to go down and cut the grass away with a stiff wire brush. At times in winter anchor ice formed in the river and in the morning we would suddenly be without water for the condenser, as the ice completely blocked the strainer, making it impossible to run until the ice was removed.

Whenever the river became low, the rush of water to the wheel churned air into the water, so that it became a serious menace, and if when the wheel was not running the water became too low, whirlpools would form at the end of the pipe and admit much air. In summer when the supply was insufficient to permit running the wheel, the water in the canal, on account of being of small volume, would get very warm, the temperature rising to as high as 80 or 85 deg. F. These troubles are not all peculiar to a siphon condenser, but are found with any type.

Previous to my taking charge of this plant there had been an accident to the water supply of the condenser just mentioned. A 2½-in. injector took its water supply from the pipe from the canal. I do not know whether there was a check valve on the line between the injector and the pipe at the time, but there was one after, and it was kept in good order. The steam blew back through the injector one night and somehow burst the 12-in. water pipe. The break occurred out in the yard where the earth contained many small gravel stones, some of which got into the pipe while repairs were being made, and I had the pleasure of removing them from the condenser cone and circulating pump.

Where I am at present there is a siphon condenser with a handy arrangement for lifting the elbow over the cone and swinging it out of the way for cleaning the cone. This arrangement may be easily and inexpensively made. It has

the advantage of being a part of the condenser and is always in place ready for use. A bar of flat iron *A*, 1x3½ in. in this case, has a slot 1 in. wide cut into the center from the points *B* to *C*, so that after the bolts are taken out of the flanges the elbow can be carried away from the joint *D* far enough to swing past the flange by manipulating the nuts



CRANE FOR SWINGING ELBOW TO ONE SIDE TO CLEAN CONDENSER CONE

E and *F* on the threaded ends of the carrying piece *G* and it may be raised by adjusting the nut on the hook in the eyebolt.

The mast *H*, which is of round iron, is pivoted in the block *J* and is free to turn in the collar at *K*. The braces *L* and *M* are attached to the studs on the flange as shown.

F. F. COWDEN.

New Bedford, Mass.

Economical Riddance of Turbine Scale

The article in the Nov. 19 issue, by Morgan C. Johns, on the above subject, has come to our attention. We experience trouble of the same kind as was referred to in our turbines and would like to have further details of the conditions under which Mr. Johns' turbines operate.

What is the steam pressure and superheat? What is the size of the turbine that can be cleaned with six barrels of coal oil? What type of boilers are in use? What is the analysis of the feed water? Does he have any trouble with his condenser from its heating up when the turbine is allowed to exhaust through it to the atmosphere? This question of scale in turbines is a very serious one with us.

R. J. C. WOOD.

Los Angeles, Calif.

In regard to scale in steam-turbine blades, I can say little, but I have had a 300-kw. turbine get badly choked with a fine substance that is sticky enough to close up the passages between the blades. There seemed to be no way to get this substance out, but to use small hooks and dig each blade free. We tried boiling the interior of the turbine with soda ash. We tried to soften the coating with oil, and then by applying a good strong air pressure to blow out the substance, but nothing had any effect. This trouble resulted from paper mill refuse being dumped into the water supply.

In summer, when the canal from which the water is taken is full, it does not harm, but in winter there is little water and much refuse, which causes the boilers to foam, and this fine dust seems to go with the steam. It lodges in the superheaters and traps and in any other place where there is a pocket in which it can lodge.

It is obvious, therefore, that there is danger of burning the superheater drum; even the tubes have to be watched, for they fill up with this stuff quickly. I do not understand how scale can form in the blades of a turbine unless there is some moisture in the steam, but even when a superheater is used there can be some water present at times. The turbine I read about in POWER could not have been coated with a very hard scale if it was softened and worked out so easily. I am watching for more letters about this scale question, for this turbine business is very interesting.

WALLACE JOHNSON.

Utica, N. Y.

Thermometer and Mercury Column Reading on High Vacuum Condensers

In the article under the above title in the Nov. 19 issue, I note on page 738 that the writer states that whenever possible a thermometer should be installed to give the temperature of the exhaust steam as this temperature can be used as a check on the reading of the mercury column.

I think that this point deserves a little further discussion because from the article one would be led to believe that the pressure shown on the mercury column should correspond to the exhaust steam temperature according to the regular tables of pressure and temperature relations of saturated steam. As a matter of fact the thermometer reading will always be that corresponding to some pressure lower than the mercury column reading. This difference is due to the fact that we always get some air mixed with the steam entering the condenser and the condenser pressure is the sum of the vapor pressures of both the air and steam present. The temperature reading for the exhaust, however, will be that corresponding to the vapor-pressure of the steam alone.

Thus in considering the temperature reading of the exhaust as a check for the vacuum reading, the check will be only to show the relative amounts of air and steam which are present in the exhaust.

The relation between the relative amounts of air and steam and their respective vapor pressure is one of direct proportionality of the relative volumes of the two to their separate vapor pressures.

JAMES L. HAYNES.

Milwaukee, Wis.

Split Pulleys

H. A. Jahnke's article on this subject in the Nov. 5 issue is interesting and full of facts. I had charge of the machinery and shafting in a factory for several years and found it hard to persuade the manager to stop getting solid pulleys. I remember once where we had to take off a 3-ft. diameter solid pulley. I could not drive it off or turn it on the shaft, so I put a chain on the rim and fastened the other end to a beam and put on the power slowly. I got the pulley to move along like a screw. I showed the manager next day the marks on the shaft and talked about split pulleys. I got them at last as we had so much changing of machines and pulleys that the advantage of split pulleys was apparent.

In the same factory I had quite a discussion with the foreman machinist. When the shafting was being installed the hangers were being put up with bolts through the floor beams. I proposed trying ¾-in. lagscrews. I made four screws as I could not get them quickly in that locality in those days—42 years ago. I tried the four screws on one of the hangers and a length of 2½-in. shaft, and had the boss machinist and the manager examine and test the screws. They were satisfied, so we had all the hangers put up with lagscrews, and made all the screws at the place—and there were 1600 of them.

Mr. Jahnke mentions split-loose pulleys. I had quite a number of them in an electric-light station in the early days of small units. The shaft was on the lower floor, with the belts coming up through the next floor to the machines. We were running the machines at different hours for incandescent and street lighting. Some of them had to be taken off the line at midnight. I did not wish to have the loose pulleys on the shaft, so I had a stand made with a loose pulley fitted on it. The loose pulleys were running only when the belt was thrown on or off.

In this same issue, Nov. 5, I noticed a picture of an Edison dynamo. We have three of them here in the factory which were made in 1884. These machines are still doing good work lighting the factory at 104 volts, which was the standard in the days when they were manufactured.

A. SANGSTER.

Sherbrooke, Que., Canada.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Effect of Lime on Pitting

When boiler-feed water contains a small quantity of lime, is it more or less likely to cause pitting of a boiler than when all traces of lime are removed from the feed water?

C. F. M.

Pitting is less likely to occur when the water contains a sufficient quantity of lime to form a thin internal scale of carbonate of lime.

Coils with Continuous Discharge

If coils are kept filled with steam at a given pressure, can water be heated by them as rapidly when the condensation from the coils is discharged by traps acting intermittently as when the discharge is continuous?

W. H. N.

No; a continuous discharge insures better removal of air and more complete contact of the steam with the interior surfaces of the coils.

Horsepower Constant

What is the horsepower constant of a 12x16-in. engine running 200 r.p.m., and what would be the horsepower developed with 19 lb. mean effective pressure?

R. S.

The area of the piston would be $12 \times 12 \times 0.7854 = 113.1$ sq.in. and the piston speed would be

$$\frac{16 \times 200 \times 2}{12} = 533.33 \text{ } ft. \text{ } per \text{ } min.$$

The horsepower constant being the power that would be developed for 1 lb. m.e.p. is

$$\frac{1 \times 113.1 \times 533.33}{33{,}000} = 1.828$$

and with 19 lb. m.e.p. there would be developed $1.828 \times 19 = 34.73$ hp.

Advantage of Vacuum System of Steam Heating

What are the advantages of a vacuum steam-heating system over an ordinary gravity return system?

W. S.

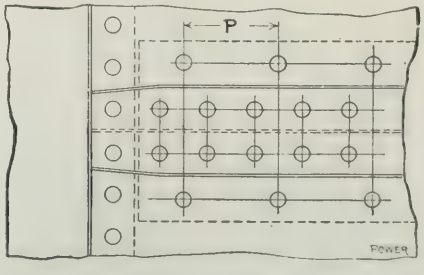
In a vacuum system, the discharge pressure being reduced, circulation and supply of steam to the heating surfaces are obtained by the use of lower initial pressure, which is especially desirable when the system is supplied with exhaust steam. A vacuum system also has better means for adjusting the activity of circulation and consequently the amount of heat radiated by the apparatus, as the discharging pressure and rate of air removal are both under control. It also possesses the advantage of requiring no air valves on coils or radiators.

Efficiency of Butt and Double Strap Joint, Double Riveted

What would be the efficiency of a butt-and-double-strap joint, double-riveted, with the tensile strength of the plate 55,000 lb. per sq.in.; thickness of the plate, ⅜ in.; thickness of the butt straps, $\frac{5}{16}$ in.; pitch of the rivets on the row having the greatest pitch, 4¾ in.; diameter of the rivets after driving, $\frac{13}{16}$ in.; cross-section of the rivets after driving, 0.5185 sq.in.; strength of the steel rivets in single shear, 42,000 lb. per sq.in. of cross-sectional area, and in double shear, 78,000 lb. per sq.in. of cross-sectional area; and the crushing strength of the plates, 95,000 lb. per sq.in. of cross-sectional area?

E. S.

The figure shows a portion of a butt-and-double-strap joint double riveted in which the dimension *P* illustrates the unit



of pitch which is generally taken, viz.: the pitch of rivets on the row having the greatest pitch.

The efficiency would be the strength of the joint for a unit of length divided by the strength of the solid plate for a unit of length. The pitch being 4¾ in., if we take this as the unit of length then:

(A) The strength of the solid plate will be

$4¾ \times ⅜ \times 55{,}000 = 97{,}969$ *lb.*

The strength of the joint may depend on: (B) the strength of the plate between the rivet holes in the outer row, which will be

$(4¾ - 1\frac{3}{16}) \times ⅜ \times 55{,}000 = 81{,}211$ *lb.*

or (C) the shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear, which will be

$(2 \times 78{,}000 \times 0.5185) + (1 \times 42{,}000 \times 0.5185) = 102{,}663$ *lb.*

or (D) the strength of the plate between the rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row, which will be

$$[[4¾ - (2 \times 1\frac{3}{16})] \times ⅜ \times 55{,}000] + [1 \times 42{,}000 \times 0.5185] = 86{,}230 \text{ } lb.$$

or (E) the strength of the plate between rivet holes in the second row, plus the crushing strength of the butt strap in front of one rivet in the outer row, which will be

$$[4¾ - (2 \times 1\frac{3}{16})] \times ⅜ \times 55{,}000] + [1\frac{3}{16} \times \frac{5}{16} \times 95{,}000] = 88{,}574 \text{ } lb.$$

or (F) the crushing strength of the plate in front of two rivets, plus the crushing strength of the butt strap in front of one rivet, which will be

$(2 \times 1\frac{3}{16} \times ⅜ \times 95{,}000) + (1 \times 1\frac{3}{16} \times \frac{5}{16} \times 95{,}000) = 82{,}012$ *lb.*

or (G) the crushing strength of the plate in front of two rivets, plus the shearing strength of one rivet in single shear, which will be

$(2 \times 1\frac{3}{16} \times ⅜ \times 95{,}000 + (1 \times 0.5185 \times 42{,}000 = 79{,}668$ *lb.*

For the assumed proportions and strength of materials it is seen from the above that the least strength of joint comes from consideration (G), i.e., 79,668 lb., then as the strength of the solid plate per the same unit of length would be (A), equal to 97,969 lb., the efficiency of the joint would be

$$\frac{(G)}{(A)} = \frac{79{,}668}{97{,}969} = 81\frac{3}{16} \text{ } per \text{ } cent.$$

Reducing Diameter of Eccentric

If ¼ in. is turned off an eccentric of an engine, what difference would it make to the valve travel?

H. A. G.

The throw of the eccentric and consequently the amount of the valve travel depend upon the amount of eccentricity, i.e., the distance from the center of the shaft to the center of the eccentric and the reduction of the diameter of the eccentric, as by turning off ¼ in., would therefore have no effect on the travel of the valve because it would not effect the eccentricity or throw of the eccentric.

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Square Root

As we learned in our last lesson, a number is easily squared by simply multiplying it by itself. If the square is given, the process of finding the number which multiplied by itself will produce that number is not so simple, but it is not so difficult as it might at first appear. To find that number would be the process of finding the root and when we are dealing in squares it would be the process of finding the *square root* of that number.

Extracting square root is really nothing much more than long division and having learned long division, square root will come easy to you.

The first step in this process is to determine how many figures there will be in the root, and this is easily found by pointing off the given number into Periods of two figures each, commencing to point off at the right of the given number, i.e., the number the square root of which we are to extract.

We must always begin to point off at the decimal point.

The number of periods in the number indicate the number of figures there will be in the root.

Thus the square root of 65,536 must contain three figures for, dividing the number into periods of two figures each, there are (6'55'36) three periods. The last period contains only one figure, but it is nevertheless a period. The square root of 65,536 is

$$\sqrt{65,536} = 256$$

Notice the three figures in the root. Now to prove that the root 256 is correct, multiply

$$256 \times 256 = 65,536$$

When a number, the root of which is to be extracted, contains a common fraction, reduce the fraction to a decimal. If it is a decimal to begin with, then we do not need to reduce it, but in either case we start to point off each way from the decimal point. The number of periods in the integer and fractional parts of the number indicate the number of figures in the integer and fractional parts of the root respectively.

Thus the square root of 692.2161 is 26.31. Here there are (6'92.21'61) four periods in all, two in the integer part and two in the fractional part, so there must be two figures in the integer part and two in the fractional part of the root.

Example: If a square room contains 1156 sq.ft. of floor space, what is the length of one of its sides.

As the room is square the sides must be of equal length, and the area being their product, is the square of the sides. The length of either side, therefore, will be the square root of the area.

We first point off the number, 1156, into periods of two places each, commencing at the right hand. There will be two places or figures in the answer.

3	11'56) 34 root
3	9
60 trial divisor	256 dividend
4	256
64 complete divisor	

We find by trial the greatest number the square of which does not exceed 11, or the first left-hand period. This number is 3, the square of which is 9. We place 3 in the quotient, as in long division, and we also set it over to the extreme left, as shown. The quotient in square root examples is the root. Now we say 3×3 is 9 and place 9 under the 11, or the first left-hand period, and subtract, getting 2 as a remainder. Now we bring down the next period, which is 56, and place it alongside of the remainder 2. We now have 256 as a dividend.

We next place the 3 in the quotient under the 3 at the extreme left and add, getting 6, to which we add a cipher. This gives us 60 and is our trial divisor for the dividend 256. Now 60 will go into 256 four times, so place 4 in the quotient and add 4 to the *trial divisor* 60, which gives us a *complete divisor* of 64. Now we find 64 will go into 256 just four times. Hence 4 was the correct figure to be placed in the quotient and our answer is 34 as the *square root* of the floor area 1156 sq.ft. This 34 represents feet and is therefore the length of a side of the square room the area of which is 1156 sq.ft.

The square root of a number that is entirely a decimal will be wholly decimal.

Example: What is the square root of 0.000676?

Pointing off into periods we have

2	0.00'06'76) 0.026 ans.
2	4
40	276
6	276
46	

As the first period consists entirely of ciphers, the first figure of the root will be a cipher. Suppose the number, the root of which we are to extract, is not a perfect power, then the root will contain an indefinite number of decimal places.

But we can fix the number of decimal places by adding as many periods of two ciphers each as we wish decimal places in the root.

Example: Find the square root of 13 to three decimal places.

3	13.00'00'00') 3.605 ans.
3	9
60	400
6	396
66	40000
6	36025
7200	3975
5	
7205	

Here we added three periods of two ciphers each to the 13. The first figure 3 of the root is a whole number because its square 9 is less than 13. The rest of the process is readily understood.

As was said before, we begin to point off both ways from the decimal point in finding the square root of a mixed number.

Example: Find the square root of 324.7204.

1	3'24.72'04) 18.02 ans.
1	1
20	224
8	224
28	7204
8	7204
3600	
2	
3602	

Notice that we pointed off toward the left from the decimal point in dividing the whole number part into periods, and to the right from the decimal point when dividing the decimal part into periods.

When the last period of a decimal contains only one figure, add a cipher to complete the period.

In some examples in square root a peculiar condition arises, as shown in the following:

Example: Find the square root of 15,202,201.

3	15'20'22'01) 3899 ans.
3	9
—	—
60	620
8	544
—	—
68	7622
8	6921
—	—
760	70101
9	70101
—	—
769	
9	
—	
7780	
9	
—	
7789	

Referring, in this example, to the dividend 620, we find that the complete divisor 68 goes into this dividend 8 times, but subtracting $68 \times 8 = 544$ from 620 we have

$$620 - 544 = 76$$

for a remainder *which is greater than the divisor.*

But if we had said 60 as a trial divisor would go into 620, nine times, then we would have had

$$69 \times 9 = 621$$

which is 1 greater than the dividend and so cannot be used.

From this we learn that whenever the divisor, 68 in this case, goes into the dividend and leaves a remainder greater than itself when a divisor of one unit greater would more than equal the dividend, we must, nevertheless, use the lesser number, i.e., we must always have the product of the divisor and the figure in the root less than the dividend.

Example: What is the square root of 113?

1	113) 10.63 ans.
1	1
—	—
200	01300
6	1236
—	—
206	6400
6	6369
—	—
2120	31
3	
—	
2123	

In this example the trial divisor 20 will not go into 13, so bring down a period, place a cipher in the root and also add a cipher to the 20, making it 200. The rest is easily understood.

Rules

(1) *Begin at the decimal point, or, what is the same thing, units place, and point off the number into periods of two places each.*

(2) *Find the greatest number the square of which does not exceed the first or left-hand period. This number will be the first figure of the root and should be written in the root and also at the extreme left of the given number. Multiply the number at the left by the figure in the root and subtract the product from the first period. Now bring down and annex the second period to the remainder.*

(3) *To the number in the column at the extreme left add the first figure of the root and annex a cipher to the sum. This figure will be the trial divisor.*

Divide the dividend by the trial divisor to obtain the second figure of the root. Now add the second figure in the root to the trial divisor to obtain the complete divisor. Multiply the second figure in the root by the complete divisor and subtract the product from the new dividend.

Bring down the third period and annex it to the remainder resulting from the subtraction. Now add the second figure in the root to the last divisor and annex a cipher to form a new trial divisor.

(4) *Work out the example according to the above rules until the last period, and then, if any more figures are required in the root annex periods of two ciphers each until the root contains the number of decimal places desired.*

(5) *Should the trial divisor not go into the dividend, place a cipher in the root, add a cipher to the trial divisor and bring down another period, then proceed as before.*

(6) *Should the root contain an indefinite number of decimal places and it is desired to have it contain a certain number of places, work the example to one place farther than desired, and if the last figure is 5 or greater, increase the last figure in the root by 1.*

Examples for Practice

(1) Find the square root of the following numbers: (a) 10.791, (b) 326.-1636, (c) 0.000784, (d) 15,202,201.

(2) (a) Find the square root of 7 to four decimal places, (b) of 13 to five decimal places.

(3) What is the length of wall required for a square engine room which must have 1764 sq.ft. area?

(4) To find the diameter of a circle that will inclose a given area, divide the area by 0.7854 and extract the square root of the quotient.

What would be the diameter of a cylindrical tank, the cross-section of which is 5674.515 sq.in.?

Answers to Last Week's Problems

(1) 20.449; 283.5856; $\frac{4}{5}$; $351\frac{9}{10}$.

(2) 2744; 599,077.1; $\frac{125}{729}$; $14.041\frac{65}{512}$.

(3) (a) That the square root is to be extracted. (b) it indicates what root is to be extracted.

(4) 576; 110,592; 104,976; $\frac{1}{512}$.

(5) That 43 is to be raised to the third power and its square root extracted. That 16 is to be raised to the fourth power and its square root extracted.

(6) 324 sq.in.; 315 sq.ft.

(7) 729 cu.in.

Correction

A serious error appeared in the Engineers' Study Course for Dec. 17. The rule for ratio should have been expressed

$$Ratio = \frac{antecedent}{consequent}$$

The antecedent and not the consequent is the dividend.

No. 1 answer to last week's problems should be 12,325 instead of 12,609. No. 4 answer is for 18 per cent. less coal received; 21.32 tons for 18 per cent. more.

Study Course Exchange

The following readers are desirous of exchanging Study Course papers with others who are following the Course:

T. A. McGrath, New Alexandria, Penn., William H. Phillips, 218 East Genesee St., Buffalo, N. Y., J. P. Colton, Ohio City, Ohio; Henry Gronbach, Jr., 2409 Silver St., Brooklyn, N. Y.; Hans Frederiksen, Box 46, Nerrington, Conn.; William L. Keil, 1836 Etting St., Philadelphia, Penn.; J. G. Lee, Box 15, Goldfield, Nev.

Heating and Ventilation Course

A course in heating and ventilation will be commenced by the Harlem (New York City) Young Men's Christian Association on Jan. 9, 1913. The class is provided especially for engineers, draftsmen, mechanics and salesmen of heating and ventilating appliances, and will last 22 weeks. During the first part, gravity, steam and hot-water apparatus will be studied, including the proper proportioning of boilers, mains, radiators, chimneys and flues, and all special methods of steam and water circulation applicable to small buildings. The second part takes up methods of ventilation. This will include appliances adapted for office buildings, hotels, factories, etc., as well as such parts of the power plant as may be intrusted to the care of the heating engineer.

Some of the topics to be considered: Properties and nature of heat; loss of heat from buildings; heat given off from radiating surfaces; properties of steam; sizes of steam mains and return mains; design of heating system (high- and low-pressure, hot-water, vacuum and vapor systems); high- and low-pressure boilers; chimneys, breeching and grates; power-plant piping and equipment; ventilation of buildings (lofts and factories, stores and office buildings, schools and theaters); air treatments (filters and screens, air washers, humidity and humidifying apparatus).

Dalton's Law in the Steam Plant

By George F. Fenno

Air and steam pressure in the condenser and boiler and examples to show that a liquid at a given temperature continues to give off vapor until the pressure of its own vapor equals the maximum for that temperature.

In the realm of steam engineering there is probably no law more important than Dalton's. A knowledge of this law enables one to properly understand the behavior of mixed gases or vapors and the effect of the pressure of mixed gases on the formation of vapors. The engineer is principally concerned with the behavior of various mixtures of air and steam, or water vapor. A mixture of these gases is found in condensers, cooling towers, dry rooms and sometimes in boilers. Dalton's law may be stated as follows:

When two or more gases are mixed in a vessel the total pressure is equal to the sum of the pressures that each component gas would exert if it occupied the vessel alone, provided there are no temperature changes and there is no chemical action between the two gases.

In Fig. 1, if *A* contains 1 cu.ft. of dry air under an absolute pressure of 20 lb. and *B* contains 1 cu.ft. of water vapor at

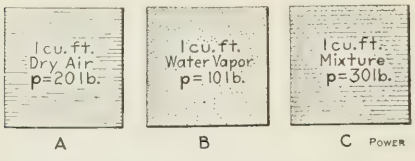


FIG. 1. PRESSURE OF COMBINED GASES

the same temperature as *A* and under an absolute pressure of 10 lb., then if these two gases are combined in a vessel *C* of 1 cu.ft. capacity, the total pressure will be 30 lb. Or, stating this in another way, the total pressure of 30 lb. in *C* is made up of 20 lb. air and 10 lb. steam pressure.

Of course, air itself is a mixture of different gases, thus the total pressure of the atmosphere is made up of the separate pressures exerted by the oxygen, nitrogen, carbon dioxide, etc., so that in reality in the above experiment, several gases were mixed and each one contributed its individual pressure to produce the total.

Every engineer knows that there is a definite relation between the pressure and temperature of water vapor. These relations are given in numerous steam tables, so that a ready method of determining the water-vapor pressures in any gaseous mixture containing steam is at hand. The curve in Fig. 2, giving these temperature and pressure relations of steam, is merely a graphical representation of the steam table, but it will show at a glance how to separate the partial pressure of air and steam. For convenience the pressures have been plotted in inches of mercury. Now suppose a condenser to have a vacuum of 27.5 in. of mercury (referred to a 30-in. barometer), then the absolute pressure in the condenser is 2.5 in. of mercury. If a thermometer were placed in any part of the condenser and showed a temperature of 80 deg. F., it would be known at once that the pressure the steam exerts at this place is 1 in. of mercury, as shown by the curve or a steam table, and the other 1.5 in. pressure must be due to some other gas in the condenser, presumably air.

That part of Dalton's law relating to evaporation, which perhaps is not so generally understood, may be stated as follows:

A liquid at a given temperature continues to give off vapor until the pressure of its own vapor equals the maximum for that temperature.

Vaporization, according to this law, will go on even though the space above the liquid is filled with some other gas at a higher pressure. Suppose when a boiler has been shut down for cleaning and is just being started up again, an air compressor is connected to the steam space and pumps up a pressure until the

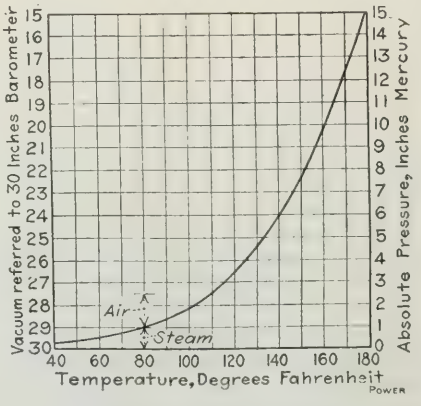

Vacuum referred to 30 Inches Barometer
15 16 17 18 19 20 21 22 23 24 25 26 27 28 29 30
Absolute Pressure, Inches Mercury
15 14 13 12 11 10 9 8 7 6 5 4 3 2 1 0
Air
Steam
40 60 80 100 120 140 160 180
Temperature, Degrees Fahrenheit
POWER

FIG. 2. TEMPERATURE-PRESSURE CURVE

gage showed 40 lb.; then suppose steam is raised until the gage shows 100 lb. What would be the temperature of the water in the boiler? Many engineers would say 337.9 deg. F., the temperature corresponding to 100 lb. gage pressure. But this is not the case, for there was 40 lb. gage, or 54.7 lb. absolute, air pressure to start with, so that, when 100 lb. showed on the gage, there would be only

$$100 - 54.7 = 45.3 \text{ lb.}$$

steam pressure present, and the temperature corresponding to this is 292.7 deg. F.

Nothing was said about the temperature of the air put into the boiler, but in order not to complicate the problem, suppose that at a temperature of 292.7 deg. it had a pressure of 40 lb. gage as assumed above. Of course, a change in the temperature of the air alters the pressure which it will exert.

If Dalton's law were not true, water could not evaporate into the atmosphere at a temperature of less than 212 deg. F., where its vapor pressure is 14.7 lb., or equal to that of the atmosphere. Yet all know that when clothes are hung on a line on a comparatively cold day they will dry, that is, the water evaporates. If the temperature of the air is 60 deg., the clothes will generate "steam" at a pressure of 0.5 in. of mercury. See Fig. 2. If this "steam" pressure of 0.5 in. were to remain pressing on the clothes they would never dry, as a greater vapor pressure could not be generated to push away the vapor already formed. Fortunately, air currents carry away the vapor and more "steam" is continually being generated, but always at a maximum pressure of 0.5 in. of mercury. If the day is very humid and "muggy," drying takes place slowly because the air already contains moisture exerting a pressure probably nearly up to the limit of 0.5 in. for the assumed temperature of 60 deg. F. If the pressure of the water vapor in the air itself before contact with the clothes were 0.5 in., the air would be saturated, that is, at that temperature it could carry no more water vapor. Strictly speaking, the air is not saturated, for, as Dalton's law tells us, the air itself makes no difference as to the amount of vapor or moisture present. In reality it is the space that the air occupies, that contains the moisture. However, for convenience, we usually speak of the amount of water that a cubic foot of air carries at a given temperature.

In a recent article showing how to calculate the quantity of heat necessary to carry on drying, the writer assumed that 966 B.t.u. were necessary to supply the latent heat of evaporation for each pound of water evaporated irrespective of the temperature at which the drying was done, 966* B.t.u. being the latent heat of vaporization at 212 deg. F. and atmospheric pressure. In the case just mentioned the temperature of drying happened to be 150 deg. and from what has just been said, it is evident that it is only the vapor pressure of the water itself which has any affect on the process of evaporation, so that the maximum vapor pressure on the substance being

*In accordance with the Marks and Davis steam tables the value 970.4 has now been generally adopted.

dried depended on the temperature. The steam table shows this pressure to be about 3.7 lb. abs. for a temperature of 150 deg., and the latent heat at this point is 1007.4 B.t.u., which is the figure that should have been used.

In a certain book on lubrication one finds the following: "The flash point is determined in the laboratory under atmospheric conditions. If the cylinder oil were to be tested under the high pressure carried in the steam pipe the flash point would undoubtedly be shown to be considerably higher, just as the boiling point of water, which at atmospheric pressure is 212 deg. F., increases with any pressure above that of the atmosphere (for instance, at 150 lb. per sq.in. the boiling point of water is 366 deg. F.). This will explain why it might be possible to use a cylinder oil successfully for lubrication under superheated-steam conditions where the temperature of the steam might even be a good deal higher than the flash point of the oil, measured under atmospheric conditions."

This author shares the quite common misunderstanding that the temperature of evaporation increases with any pressure. It only increases with the partial pressure of the vapor of the liquid itself.

Now the flash point, or the temperature at which ignitable vapors are given off, of most steam-engine cylinder oils is usually between 500 and 600 deg. F. They evidently then have to be heated to this temperature before their vapor pressure equals that of the atmosphere, so that at 366 deg. F. their vapor pressure must be quite low and the 150-lb. steam (a different gas) pressure can have no deterrent effect on the formation of the oil vapor. If the oil were tested in a closed room where the air pressure is 150 lb. the same flash point should result.

There are many other cases of the mixture of different vapors, such as in carburetting volatile liquids for internal combustion engines, which are interesting engineering examples of the application of Dalton's law. One more illustration, showing an unusual example, will suffice.

In cases of carbon monoxide poisoning, such as occur in mines, it is found that recovery is greatly quickened by placing the victim in oxygen at a pressure of two atmospheres. Due to the increase in pressure, the amount of oxygen in free solution in the blood, is increased immediately in accordance with Dalton's law and the tissues are more quickly supplied with the necessary oxygen. This oxygen is really forced into the blood and is quite independent of the normal exchange of oxygen that goes on.

Measuring Air from Compressor

By E. A. Fessenden*

Simple and accurate method of measuring air discharge from compressor. Only one tank is required and this may be the receiver forming part of the equipment.

*Associate professor of mechanical engineering, University of Missouri.

E. M. Ivens' article on "Air Compressor Efficiency" in the Oct. 15 number of POWER recalls to the writer a method of measuring the volume of air delivered by a compressor which he has often used successfully. While not any better than the orifice method or the method using two tanks suggested by Mr. Ivens, it may be of interest to readers.

Only one tank is required and this may be the receiver tank which often forms a part of the compressed-air equipment. If such a receiver is not a part of the equipment, any reasonably tight tank will answer the purpose if it is strong enough to hold the normal delivery pressure of the compressor safely. A few small leaks will not affect the results in any manner whatever.

The arrangement of the tank is shown in Fig. 1. The outlet pipe contains a tee for a thermometer cup *T* and two valves *A* and *B*. A carefully calibrated gage is provided to read the pressure of the air

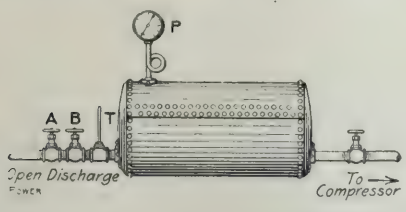


FIG. 1. AIR TANK FOR MEASURING DISCHARGE

in the tank. If the regular receiver is used the outlet pipe may be the ordinary service pipe, but when a test is to be made it must be disconnected from the service lines so as to discharge directly into the atmosphere through the two valves *A* and *B*. There should be a valve between the tank and the compressor. All other outlets from the tank should be closed off.

When ready for a test, first open valves *A* and *B* wide. Then with the compressor running at its usual speed gradually close valve *B* until the pressure in the tank remains constant at the normal working discharge pressure of the compressor. The compressor will then be running under normal operating conditions. After allowing the compressor to run in this condition for a short time to insure that all conditions have become constant, close valve *A* tight and allow the pressure in the tank to rise to 10 or 15 lb. above the normal working pressure. Then close the valve between the tank and the compressor and stop the latter.

Open the valve *A* quickly and take simultaneous readings of time and pressure as the pressure in the tank falls due to the escape of air through the valve *B*. The best way to do this is to have an assistant hold a watch and call "read" at uniform intervals of 15 or 30 sec. The exact pressure should be read at the instant the signal is called. Continue this until the pressure has dropped considerably below the working pressure.

Plot the readings just taken on cross-section paper so as to obtain a curve of pressure against time as illustrated in Fig. 2.

From the thermodynamics of gases we have the general equation for any gas,

$$PV = WRT \qquad (1)$$

in which

P = Absolute pressure in pounds per square foot;

V = Volume of gas under the pressure P, in cubic feet;

W = Weight of gas present in pounds;

T = Absolute temperature of the gas in deg.F. ;

R = Constant = 53.35 for air.

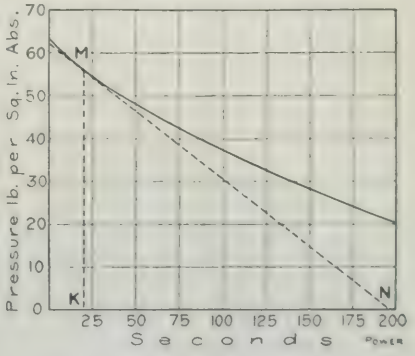


FIG. 2. PRESSURE-TIME CURVE

Equation (1) can be rewritten as

$$W = \frac{PV}{RT} \qquad (2)$$

In the case under consideration W, P, V and T represent the weight, pressure, volume and temperature of the air in the tank at any time. The volume and the temperature remain constant and the

pressure and weight decrease as the air escapes through the partly open valve B. The volume considered is the entire volume of the tank and all connections between valve B and the valve between the tank and compressor. This can be determined by filling the tank and connections with water, weighing the water required and calculating the volume by dividing by 62.355, the weight of 1 cu.ft. of water at 62 deg. F.

Differentiating equation (2) with respect to time gives

$$\frac{dW}{dt} = \frac{V}{RT} \times \frac{dP}{dt} \qquad (3)$$

in which $\frac{dW}{dt}$ is the rate of decrease of weight of air in the tank, or in other words, the rate of discharge, in pounds per second, while $\frac{dP}{dt}$ is the rate of decrease of pressure.

In Fig. 2 let M be a point on the pressure-time curve corresponding to the normal working pressure of the compressor. Draw the line MN tangent to the curve at M (this must be done carefully). Draw the line MK parallel to the pressure axis. Then

$$\frac{dP}{dt} = \frac{MK}{KN} \qquad (4)$$

The values of MK and KN are read in pounds per square foot and seconds directly from the figure.

If the value thus obtained for $\frac{dP}{dt}$ is introduced into equation (3) the rate of discharge in pounds per second is found directly. Since the weight discharged must be the same as the weight drawn into the compressor, the free air capacity is found by the formula

$$V_0 = \frac{WRT_0}{P_0} \qquad (5)$$

where V_0 is the volume in cubic feet of air per second at the intake pressure P_0 (in pounds per square foot absolute) and at the absolute temperature T_0, and W is the weight of air discharged per second just found from equation (3).

To illustrate this method consider Fig. 2 to be the pressure-time curve of a small compressor delivering air into a tank which with its connections has a volume of 47.7 cu.ft. The temperature of the air in the tank at the time the test was made was 87 deg. F. or 547 deg. abs. It was desired to find the amount of air discharged against a pressure of 56 lb. per sq.in. abs. Thus the line MK represents 56 lb. per sq.in. or 8064 lb. per sq.ft. From the figure KN is 194 — 20 = 174 sec. Then

$$\frac{dP}{dt} = \frac{MK}{KN} = \frac{8064}{174} \text{ lb. per sq.ft. per sec.}$$

From equation (3)

$$\frac{dW}{dt} = \frac{47.7}{53.35 \times 547} \times \frac{8064}{174}$$

$$= 0.0758 \text{ lb. per sec.}$$

At atmospheric pressure and 32 deg. F. the volume of air passing into the compressor is

$$V_0 = \frac{WRT_0}{P_0} = \frac{0.0758 \times 53.35 \times 492}{14.7 \times 144}$$

$$= 0.9391 \text{ cu.ft. per sec.} = 56.346 \text{ cu.ft. per min.}$$

When accurate results are required, it is well to repeat readings for the pressure-time curve several times and take the average value of $\frac{dP}{dt}$ thus obtained. It is, of course, important that the pressure gage and thermometer used be carefully calibrated, and that the temperature of the air remain the same when the readings for the pressure-time curve are taken as when the valve B is adjusted. After once adjusting valve B it must not be disturbed until the test is finished.

This same scheme can be used to provide a constant load on a compressor for an economy test. It is only necessary to adjust the valve B to give the discharge pressure desired and then run the compressor continuously at the proper speed, allowing it to discharge into the tank and the air escaping through valve B. As long as the pressure and temperature of the air in the tank are constant the rate of discharge is constant. The compressor governor should be adjusted to keep the speed constant and the pressure will remain constant as long as the valve B is unchanged. The temperature of the discharged air is practically dependent upon the amount and temperature of the jacket-cooling water. If these are kept constant the air temperature will not change appreciably. The compressor is then allowed to warm up thoroughly so that all these conditions become constant and the test is begun by adjusting valve B, thus fixing the rate of discharge as determined from a pressure-time curve. The load will now remain constant and the readings of steam pressure, steam weight, indicated horsepower, etc., are taken in the ordinary way.

The method outlined above has several advantages: It is rapid, simple, practically and theoretically correct and can be made very accurate if care is taken in making the readings and in plotting the pressure-time curve. It is flexible and can easily be adapted to widely different conditions of operation. All leakage and friction losses are taken into account. No expensive apparatus is required and often not even a change in piping. The manipulations and calculations are easy.

A Pacific Coast company, now erecting a large hydro-electric plant, will use a transmission voltage of about 150,000 volts on a line about 275 miles long, terminating at Los Angeles. Climatic conditions in California are such as to permit of voltage transmission as high as 250,000.

Notes on Oil Burning

By William Kavanagh

Whenever oil is burned in the liquid state or in conjunction with coal, smoke is sure to form especially when the furnaces are being heated up or steam is first being generated. When oil is burned in the gaseous form, smoke is less likely, but even then some will form if the men attending the furnaces are not accustomed to burning oil, or if the burners are poorly proportioned or designed.

In general, oil burners of the spraying type will produce smoke; the amount of air required for smokeless combustion of sprayed oil is usually so large as to greatly lessen its calorific value. Within the last few years the newspapers have been strenuously urging the use of oil on our battleships, claiming "the use of oil will prevent the detection of our warships on account of the absence of smoke, especially when the warships are below the horizon or hidden behind a projection of land." This argument is usually followed by the assertion that the speed of the ships will be greatly increased from its use.

Those not conversant with oil burning generally believe the newspapers' contentions and argue that the universal use of oil on warships is only a matter of time.

Two kinds of oil are in general use for steam generation. One known as "fuel oil," and the other as "crude oil." Fuel oil is a product of the oil refinery while crude oil is taken direct from the wells. The calorific values of the two oils do not vary much, but crude oil has the highest. The difference in favor of crude oil can be easily lost if it is poorly handled. It will take in round numbers 185 gal. of fuel oil to equal one ton of coal of good quality, allowing 7.5 lb. to the gallon and 20,000 B.t.u. per pound. One ton of good coal will contain 28,000,000 B.t.u. and dividing this by the heating value of one gallon of oil gives about 187 gal. as the result.

The rapidity with which smoke can be formed when oil is improperly burned is one reason why some means should be adopted for generating steam before oil is used in the furnace. Starting with every boiler cold and steam yet to be raised and with no way of atomizing the oil, smoke will roll off in great clouds, and continue to do so until sufficient steam is generated to atomize the oil.

Compressed air, if available, may be used as an initial means of smoke prevention, but neither air nor steam will stop smoke forming if the oil is improperly handled or fed to the oil burners.

Oil has its advantages and disadvantages as compared with coal. Among the first are quick steam generation, less space occupied, easier handling or moving from place to place and, when used on ships, a greater steaming radius for

the same bunker capacity. Its disadvantages are greater danger from fire and explosion, due to gas formation, and if it is not properly fed to the furnaces and the flame should happen to die out there will be no heat in the furnace. Then an explosion is likely, for after the flame dies out the oil or gas may appear again at the burner nozzles and, accumulating, explode when the burner is again lighted. Usually this results in blowing off the furnace doors, back connection doors, etc., and the boiler-room attendants are likely to be injured.

Another disadvantage of oil is the rapid formation and deposition of carbon on the heating surfaces of the boiler which retards heat transmission. Coal also will deposit ash or carbon on the boiler surfaces, but this deposition is slow when compared with that of carbon from improper oil burning. Ever since oil has been used for generating steam the promotors of some particular oil burner have advocated air as the atomizer arguing "when air is used it is burned," and in this way the calorific value of the oil is increased. Even were it true that air could be burned and that its use augments the heating value of the oil, special arrangements would have to be made to obtain the air at sufficient pressure to act as an atomizer. But air is noncombustible, and its use will not add any heat to the oil. Its supply is universal and it can be used when and wherever one pleases if one has the means to store and compress it.

Numerous attempts have been made to construct an oil burner that would produce its own air pressure, giving to the air sufficient velocity for it to act as an atomizer. Other oil burners have been constructed with a view to self-gas production. In this type of burner the flame issuing from the nozzles strikes the burner, thus raising its temperature and breaking up the liquid oil into finely divided particles easily gasified.

Oil has been burned in the gaseous form directly. This manner of oil burning is effected by pumping the oil into an oil boiler, the furnace of which is fed with coal or oil. The oil is thus gasified and the gas is then allowed to flow through the burner in the different furnaces until it issues and is ignited by a pilot flame or jet. The oil boiler is similar to a steam boiler and is handled after the same manner. The oil is pumped to the boiler and is carried at a certain height in the gage-glass. When a sufficient pressure is generated the gas is used as above mentioned.

One great objection to using oil in the gaseous state is its tendency to carbonize or choke up the burner nozzles and prevent satisfactory burning of the gas. All attempts to burn oil in the gaseous form have resulted in this carbonization and stopping of the openings in the nozzles, thus requiring constant attention on the part of the firemen to keep the slits or openings clear to permit the emission of the gas to the furnace.

Points in Buying Mechanical Stokers

By H. S. Knowlton

Many important clauses are incorporated in specifications and contracts, the purpose of which is to safeguard the purchaser.

A number of these points are brought to the reader's attention in this article.

Among the functions of the operating engineer none is more important than advising the plant owner in the purchase of new equipment. The more able an engineer is in selecting the best apparatus, the greater his value to his employer and the more likely will the right types of machinery go into the plant. Today the plant owner expects more of his engineer than doing routine operating work, and often studying machinery contracts, specifications and correspondence in connection with former purchases has well repaid the engineer.

Suppose the plant owner becomes convinced that it will pay to buy a mechanical stoker, and he asks his operating engineer's aid in drawing up the specifications and contract. Usually a consulting engineer will do all this work, but if the operating engineer is well informed and his judgment respected, he is likely to be consulted at least on the important engineering points which should be included in the specifications. The legal points to be covered are less his province, but even these are occasionally familiar to the operating man.

Legal Points

The legal phase of the stoker contract need not be discussd here in detail, but its general subjects and their significance are worth knowing. They include the date of the agreement, names of the parties concerned, definitions of any terms subject to false interpretation, description and quality of work required, date of completion, prices, payments and notice of shipment of the equipment as well as clauses covering the responsibility for accidents and damages incurred in connection with the work, method of securing alterations and amendments, guarantee of good workmanship and operating efficiency, abrogation of previous communications inconsistent with the terms of the agreement, and possibly a provision for arbitration in case of a dispute.

Sometimes there is a bonus or penalty clause for performing the work within a specified period, or based upon the efficiency of the equipment when tested under stated operating conditions. Ordinarily, the local engineer is not concerned with these points, but none of them should be overlooked.

Engineers' Problem

The engineering elements in a purchase are typified in a recent installation of stokers under three water-tube boilers, these involving general requirements, drawings, description of work, rating of boilers and operating results. The engineer selecting the apparatus required bidders to submit detailed drawings to scale. He gave each the boiler heating and superheating surface, operating steam pressure, superheat and the width of the furnace and its depth from the dead plate to the bridge-wall. The purchaser agreed to provide labor for handling the stokers from the railroad siding to the boiler room, but the maker was required to provide all labor for erecting, including the stoker parts and supports, driving shaft and air-controlling doors.

The purchaser was further to provide the premises and furnaces for the stokers, build the brickwork and furnish the air equipment to a stated point, besides connecting the driving engines with the line shaft. The engineer in charge defined the term normal boiler rating, giving the bidders the amount of evaporation per hour from and at 212 deg. and the superheat and steam pressure, based on 0.56 as the specific heat of superheated steam.

Operating Results

Probably the most important thing done by the engineer is what the owner particularly expects of him, to specify the operating results. In this case the engineer required the stokers to hold the steam pressure within 5 lb. of the desired working pressure when the boilers were normally operated at not over 100 per cent. in excess of their rated capacity.

Another important specification was that the stokers, when burning semi-bituminous coal of not less than 14,000 B.t.u. per pound of dry fuel, should give with the boilers and furnaces a combined efficiency of not less than 65 per

cent. when the boilers were operating at not less than 175 per cent. of their normal rating. The brickwork, however, was to be in good condition and free from cracks, and the baffling so arranged and the tube surfaces kept so clean that the temperature of the flue gas would not exceed 550 deg. It was also required that the stokers, when operated according to the maker's instructions and using the fuel specified, should burn 3300 lb. of coal per boiler per hour.

Conformation to the local smoke laws was required and the power for operating the three stokers and blower equipment was limited to 120 hp. when burning coal at the rate of 3300 lb. per boiler per hour. One fireman was to suffice for the three stokers when operated up to and including 200 per cent. of the boiler rating. The stoker manufacturer had to guarantee to replace, free of charge for one year, all repairs in excess of $350 or 5 per cent. of the stoker investment not caused by neglect or abuse. The dates and manner of delivery were also specified, the engineer recommending the purchase.

Obviously the detailed practice in purchasing stokers or other power-plant equipment would depend on the conditions; the figures cited were given merely as a concrete illustration. In another case, very different requirements might be proper, but in general the same ground should be covered by the operating engineer wishing to make a comprehensive recommendation when his employer asks his advice.

Reference in detail to a large amount of preliminary work has been omitted, since any recommendation would naturally be based upon a thorough study of competitive stoker designs, comparison of prices and guarantees, observation of the behavior of equipment in other plants and as wide reading as possible upon the general subject.

Care of High Vacuum Apparatus

Some practical suggestions on detecting and preventing leaks, the importance of using thermometers to show operating defects and the use of chemicals to detect contamination in water.

In plants carrying a high vacuum a large supply of cold water is absolutely essential. This may be curtailed by mechanical troubles in the pump or its prime mover, pipes too small, dirty strainers, pipes or tube sheets, and air leaks in the suction piping or at the shaft glands on the circulating pump.

Leakage at the glands is a most prolific source of trouble which can be overcome by proper ways of packing, such as the following which does not require frequent renewals.

Fill the stuffing-box for about one-third its depth with a soft metal packing, another third with a flanged metal ring and the rest of the box with the same soft metal packing. Then screw the gland up fairly tight. Drill down through the casing into the box near the inner flange of the metal ring and make a ¼-in. pipe connection to the discharge side of the pump. This forms an effective water seal that will fill any groove that may be cut in the packing and prevent air from flowing in.

The wet-air pump should be set with its suction inlet slightly below the condensed-water outlet in the condenser and a drain connection provided in the suction chamber, which should be opened when the air pump is stopped. Cast-iron flanged pipe and fiber gaskets are satisfactory on the air-pump line, which should be as straight as possible and as large as the pump outlet. The hotwell line should have a controlling valve near the pump and means for discarding contaminated condensate. If the piping be tight and the pump packing receives ordinary care, little trouble will arise. The water valves should be of composition to withstand the rise in temperature that will occur should the vacuum fail. Where there are more than one the air pumps should be interchangeable.

Dry-air pumps are generally of the rotative type with water-jacketed air cylinders and metal valves. If the piping is tight they give little or no trouble, the attendance required being only that needed by any reciprocating engine of the size and speed.

The hotwell pump, usually located about 4 ft. below the outlet of the condenser, is controlled by a float located in a chamber at the base of the condenser and adjusted to stop the pump while there is still a little water in the chamber. These pumps should be in duplicate to guard against failure.

CONDENSER LOG

Condenser No. 3 — Date July 13, 1911

Hour	Vacuum	Kw.	Temperature: Inlet water	Outlet water	Rear head	Steam inlet	Air pump discharge	Hotwell line	Remarks
6:30 p.m.	28.4	900	72	82	73	90	89	89	
7:00 p.m.	28.0	900	72	88	75	98	97	97	Strainer partially clogged.
7:30 p.m.	28.4	1160	72	82	73	90	89	89	
8:00 p.m.	27.8	1960	72	96	77	101	100	100	Insufficient circulating water.
8:30 p.m.	28.3	2260	72	83	73	92	91	91	
9:00 p.m.	28.4	2030	72	78	72	90	89	89	Too much circulating water.
9:30 p.m.	27.9	1860	72	82	73	101	100	100	Leak at glands plunger rods of air pump.
10:00 p.m.	28.4	1860	72	82	73	92	90	90	
10:30 p.m.	28.4	1800	72	81	73	92	91	91	
11:00 p.m.	28.4	1800	72	82	73	92	91	91	

Use of Thermometers

As a small change in the vacuum on a turbine makes a comparatively large difference in the economy, engineers have ceased to depend upon the spring vacuum gage as an indicator and have turned to the more accurate mercury column which in turn is checked by thermometers, for no change can occur in the degree of vacuum without a corresponding change in temperature. The connection of the mercury-column gage to the steam-inlet neck of the condenser is made as short as possible. Thermometer sockets are also placed in this neck and in the circulating-water inlet and outlet, the rear head of the condenser and the hotwell line. All thermometers should have a range at least up to 240 deg. F. Another thermometer is hung in the room adjacent to the mercury-column gage by which to correct the readings for temperature differences.

The accompanying table is a condenser log, taken from a 2000-kw. Curtis horizontal turbine unit, covering a period of about 5 hr. At 6:30 p.m. the load was extremely light and the vacuum apparatus in normal condition. At 7 o'clock the vacuum had dropped to 28 in. without any change in the load. The increase in the temperature of the discharge water indicated an insufficient supply of circulating water. Investigation showed that the strainer on the intake was partially clogged. This was evidently cleaned, for at 7:30 the apparatus was in normal condition despite an increase in the load. At 8 o'clock the vacuum had fallen to 27.8 in. and the

thermometer again showed a deficient supply of circulating water. This was corrected by throttling at the circulating-water inlet until the thermometer indicated the normal 10-deg. rise.

At 9:30 the vacuum had fallen to 27.9 in. The thermometers indicated that the circulating water was at normal temperature; hence the cause had to be an air leak. This was located at the glands on the air-pump plunger rods and corrected. The log shows normal conditions throughout the balance of the run. The table serves to show what valuable aid the thermometers render to the operator.

If centrifugal pumps are used to supply the circulating water, means are necessary for priming them quickly. A separate pump for this service, is generally provided, although any one of the wet-air pumps may be used, if means exist to discard the water that may be picked up in the operation.

Air Leaks

If the turbine-shaft glands are steam sealed, a gage should be mounted upon each packing case. The usual pressure maintained on these glands is 10 lb. gage and the steam just shows at the glands when the vacuum is normal. If any leaking occurs a greater pressure must be maintained to effectively seal the gland. If the glands are water sealed, there should be a water meter on the line supplying the sealing water or else an indicating valve should be used to control it. The normal amount of water for sealing purposes being known or the normal opening of the valve, a leak may be observed by the increased amount required.

Air leaks in the balance piping show by a marked rise in the degree of vacuum between no-load and full-load conditions. These leaks may or may not show when running noncondensing.

Leaks at the automatic atmospheric exhaust valve may be found by laying a temporary wooden cover over the end of the atmospheric exhaust, connecting a manometer gage to the pipe and working up a vacuum upon the unit. This may be done while the unit is in operation. Another method used where exhaust heads are placed on the end of the pipe, is to admit steam to the pipe on the atmospheric side of the valve and when the pipe is thoroughly heated, as shown by steam appearing at the end of the pipe, the steam is throttled until it just shows at this point. A vacuum is worked up and a leak at the valve will cause the small amount of steam to disappear entirely.

Leaks in the air-pump valves will show by the pump failing to maintain a vacuum within itself when the suction valve is closed. Air leaks in the condenser shell or in the piping may be found by the smoke test or by testing with a candle flame.

Testing Water for Contamination

When the circulating water is taken from a stream, be it either salt or fresh, there is always danger of contaminating the condensate through leaky tubes or tube packing. This may be guarded against by taking half-hourly chemical tests of the condensate. The necessary outfit consists of a dozen clear-glass 2-dm. phials, a dark blue or brown 2-dm. phial, a 4-oz. dark-brown bottle, a medicine dropper and a tray to hold them. If the circulating water is salt or brackish the chemical used is nitrate of silver, which is prepared as follows:

Weigh out 50 gm. of the silver nitrate crystals and place in the 4-oz. bottle, cover about 1 in. with distilled water (condensed steam), and shake until the crystals are entirely dissolved. Then fill the bottle to the top with distilled water. Next fill the dark-brown 2-dm. bottle two-thirds with the solution and put the medicine dropper in it. Cork up the balance and stand it aside until needed.

For a test take a 1½-dm. sample of the condensate from a convenient tap in the hotwell line by filling one of the clear-glass 2-dm. phials three-fourths full. With the medicine dropper held well above the mouth of the bottle containing the sample, press three drops of the solution into it. If a minute quantity of salt is present the sample will take on a bluish-white cast, but if much is present the sample will turn milky white. With a solution of this strength any water showing a trace of salt should be discarded until the leaky tube can be located and replaced. Care must be taken not to allow the dropper to touch the sample being tested, for if salt is present the small amount that adheres to the dropper will spoil the balance of the solution in the dark-brown 2-dm. bottle.

If the circulating water is fresh the method of procedure is the same, but the chemical to be used and the strength of the solution will depend upon the injurious ingredient in the water and the percentage that may be admitted to the system without giving trouble. The nitrate of silver test may be used if the water contains chlorides and litmus if acids or alkalies are present. Oxalate of ammonia will show organic matter.

Locating Leaky Tubes

Having ascertained that some tube leaks, the defect may be located in two ways. One is by flooding; that is, by closing the valve between the condenser and turbine and admitting water at about 10 lb. pressure to the steam space of the condenser after the water-chamber heads have been removed. The water finds its way through the leak and will be seen running out at the ends of the defective tubes. It is well while the heads are off to blow all the tubes with an air jet to remove any sediment that may have collected.

Often a tube will split and the flooding process fails to indicate the leak due to the pressure on the outside of the tube closing it sufficiently tight to prevent the water coming through. Then the condenser must be drained and the vacuum method applied by starting the air pump. The pressure of the atmosphere will force the split tube open and it can be located with a candle flame or by blowing a cloud of smoke against the tube sheet. Sometimes split tubes are difficult to locate and one end of the tubes may have to be blanketed by pasting wet newspapers over the tube sheet. This requires a little patience and skill to avoid tearing.

Before closing the condenser all the ferrules should be gone over to see that they are tight. It will save the trouble of taking off the heads at some future time only to find that the leak is at the packings.

Corliss Governor Troubles

By F. C. Holly

When putting a new governor in service or having trouble with an old one, knowing where trouble is most likely to be found, enables the engineer to quickly reason from the visible effect in the engine to the cause at the governor.

Probably the most frequent trouble in governors is binding in the governor-rod bearings, because the pins are too short; when the nut and washer are drawn up tight, side-binding results. This will cause a variation of engine speed in proportion to the friction induced, and a very little friction will cause a very noticeable variation at the switchboard; excessive binding will cause severe racing. This fault applies to the governor connections proper and to the cam-lever connections. The remedy is to adjust every rod so that it can be freely moved sideways on the pin.

In one instance, several years ago, an expert made a trip of 300 miles and back to loosen one little nut on one of the governor pins, this being all that was necessary to cure a bad regulating trouble. The engine was in a municipal lighting plant and at least one of the engineers was experienced in Corliss-engine operation.

The practice of placing the knockoff-cam lever bearing between the steam arm and the double arm, thus exposing it to a side thrust from each, is fundamentally wrong. The knockoff cam should operate on a separately flanged bearing.

Also important are the setscrews, or keys if any, in the governor gears. Serious accidents have occurred from these gears working loose on their shafts. On

one occasion an erector installed and started an engine and left it in charge of the regular attendant for a few moments. When he returned to the plant he found the engine racing badly. An investigation showed that the governor-driving gear was alternately slipping and catching on the shaft. The setscrew had not been properly tightened. Inspecting the governor gears and pulley occasionally is advisable.

Racing Causes

Sometimes, under extreme variations of load and even under less exacting conditions, a stiff governor will allow the engine to race. This is often due to too low speed in the governor, and is corrected by decreasing the size of the governor pulley and adding weight to the governor until the engine picks up its normal speed. The governor can be weighted by hanging small weights on temporary wire hooks to the projecting yoke of the governor, afterward weighing them and adding the equivalent weight permanently to the governor in whatever form its design best allows.

Before doing this, one should make sure there is no undue friction in the governor parts. This may be determined by removing the weights and balls and disconnecting the valve-gear, and seeing if the arms, after being raised as high as they will go, drop freely of their own accord. If, on moving them up and down, no binding or sticking is evident, there is no undue friction.

Dashpot Oil

A trouble every engineer experiences is that of oil getting low in the governor dashpot. This should be the first place to look for the cause of a racing engine; but much valuable time is often lost investigating other possible causes before looking to this most likely and obvious one. To make up for evaporation, a little oil should be added to the dashpot occasionally. When all governor parts are properly adjusted to eliminate friction and all bolts, setscrews and keys are tight, the possibility of trouble is narrowed down to slipping belts or the remote possibility of broken parts.

That all possible friction must be eliminated may seem unimportant but cannot be too strongly emphasized.

Counterweight Required

A leading pumping-engine builder contracted to furnish a high-duty pumping engine. The specifications required a governor capable of regulating the engine to a fixed speed. Previous to this contract no such regulating attachment had ever been furnished by this company, and none of the engineering force was familiar with governor design. However, a governor was made and applied. When the engine was started, and the governor put into commission, only the prompt action of the engineer in stopping the engine prevented a wreck. A more cautious trial was made and showed that the governor would lengthen the cutoff to the running away point and then go to the other extreme and shut the engine down.

Throughout the following months experiments were tried. Belt transmission was changed to gear drive and back again. Speeds were altered and cam attachments applied and taken off again; the best and oldest engineers of the company exercised their wits on the problem to no avail.

Extension rods ran from the governor to the valve-gear 25 ft. above, transmitting the motion of the governor arm through bell cranks and shafting. Hundreds of dollars were spent to say nothing of the loss of time and prestige before someone thought to counter-balance the weight of these rods and cranks and eliminate the friction from various pins and shaft bearings. Although these same pins and bearings were originally erected in a manner to satisfy all ordinary mechanical requirements, they were not perfect enough for governor work, a fact a good governor man could have pointed out in a few moments.

Sarco Automatic Lubricator

This lubricator is made in two types, one actuated by a pendulum, for moving bearings, such as crossheads, cranks, etc.; the other actuated by clockwork in a dustproof casing and used for stationary and slow moving bearings. Both lubricators feed grease similarly.

In the pendulum type, the pendulum, instead of a spring, actuates the clock movement and consequently when the bearing stops moving the pendulum the feeding of the grease stops simultaneously.

The body of the cup is of gun metal and screws to the bearing. Grease is pressed down into the bearing by a piston, operated by the clockwork, which is placed in a hermetically sealed casing. The speed of the piston depends on the the thread of the spindle, six different sizes being provided to suit conditions.

The clock is wound up daily. If more grease is delivered than required the clock stops automatically. It restarts as soon as the bearing requires lubrication. A lever on the cover of the lubricator releases and stops the clock. These lubricators are manufactured by the Sarco Engineering Co., 116 Broad St., New York City.

The Bureau of Mines has a copy of bulletin 49, "City Smoke Ordinances and Smoke Abatement," by S. B. Flagg, 55 pages, for distribution. Address the Director of the Bureau of Mines, Washington, D. C., and order by number.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Gold per fine ounce is valued at $20.671834625323. *Engineering News* asks why the computer "stopped at twelve places." For one thing, brother, this is cold weather. But twelve places! If we once stopped in this many, we would clamber up the H_2O vehicle and stick to celery tonic.

We take it, you have never yet tried to die a poor man. A. Carnaygie expects to do just this thing, leaving his heirs a paltry twenty-five million—and he is being severely criticized. When a man has a good impulse, encourage him; don't grab your 16-lb hammer and start the "Anvil Chorus." Even the millionaire is entitled to some consideration.

Somebody in sinking a shaft into the rich lodes of antiquity has discovered that old King Coal was a useful old soul even in Aristotle's halcyon days, about 238 B. C. Perhaps this is the reason we pay so much for it; it's a genuine antique. Johnny, run down into the cellar and chase up another hodful; mother is going to bake some biscuits.

A lady with the suggestive name of Catt urges woman suffragists to escape the thralldom of skirts and don trousers. They (the women) will break into the boiler room yet, and then you'll hear: "Why in ——! Beg pardon, Phyllis (bowing and opening the firedoor); when you have arranged your coiffure to your satisfaction, will you kindly fire up on No. 4? We're losing steam."

Who said corporations are soulless? The Hotel La Salle, Chicago, gave a Thanksgiving dinner to its 1260 employees. Because all the engineering Mahomets could not come to the mountain, a mountain of "big eats" came to them in the engine room. Gee whiz! Think of anybody discovering that there are human beings in an engine room!

In a South American town is a corking good machinist when he steers wide of the demon Rum. Red-eye and humor, mixed, is a sore combination in a power plant. One morning the master mechanic found the c. g. m. feeding a fine white powder to a compressor crankpin that was running hot.

"Shay, boss! Crankpin (hic) got a high fever. Givin' her quinine; 's only dope for fever!"

Economy in Steam Generation

By D. F. Nisbet

Calculation by formulas and approximate methods of determining the amount of air theoretically required to burn one pound of coal.

It is suggested that the volatile combustibles and fixed carbons be burned separately.

A recent trend is to increase boiler capacities far beyond their normal rating. While undoubtedly a move in the right direction, to be permanent, it must be carried out so that at least fair economy is obtained in operation. The furnace and its conditions must have more consideration than heretofore, and the proper burning of the fuel also will need to be treated as an engineering problem, and be studied as carefully and intelligently as has been the use of the steam. Putting a certain mount of coal into the furnace and an uncertain volume of air, leaving all other functional conditions to fortuity, will not be sufficient.

Most important is getting away from the need of admitting so much more air than the fuel theoretically requires, and, next, devising means for realizing more completely the heat value in the hydrocarbon distillates. Many other sins of omission and commission will need correction, but these two evils predominate and should receive first attention.

Many authorities on combustion maintain that 1 lb. of coal requires theoretically about 12 lb. of air, and, that practically 18 to 19 lb. gives the greatest efficiency. These statements have been widely disseminated, and would seem to assume that all bituminous coals require the same amount of air for proper combustion, and that 50 per cent. excess air is always necessary.

The weight of air theoretically necessary to completely burn a pound of any coal may be calculated from an ultimate analysis of the coal by the formula:

$$A = 11.52\,C + 34.56\left(H - \frac{O}{8}\right) + 4.32\,S$$

where A = air; C = carbon; H = hydrogen; O = oxygen; S = sulphur.

A close approximation to the heat value of the coals in British thermal units can be found also from the ultimate chemical analysis by Dulong's formula.

$$14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\,S$$

where C, O, H and S represent the percentage of the constituents of the coal.

To show that coals of variant chemical characteristics differ greatly in their theoretical air requirements, the ultimate analyses of four coals from widely separated localities are here given, each with the amount of air theoretically necessary for its perfect combustion and the thermal value.

Case 1. A Missouri coal, with an ultimate analysis of C = 52.55, H = 3.75, O = 7.10, S = 4.73. The air theoretically required is 7.22 lb. per pound of coal having a thermal value of 9465 B.t.u. Supplying 18 lb. of air per pound of this coal, the excess air would be 149 per cent.

Case 2. An Illinois coal analyzing C = 64.10, H = 4.54, O = 8.56, S = 4.87. The air theoretically required is 8.78 lb. and the heat value 11,705 B.t.u. With 18 lb. of air per pound of this coal the excess would be 105 per cent.

Case 3. A Hocking Valley coal having C = 68.3, H = 5.29, O = 15.64, S = 1.59. The air theoretically necessary is 9.09 lb. and the heat value 12,180 B.t.u. Per pound of coal this 18 lb. of air would show a 98 per cent. excess.

Case 4. A high-grade West Virginia coal analyzing C = 86.14, H = 4.41, O = 2.56, S = 0.60. The air theoretically required is 11.36 lb. and the heat value 15,135 B.t.u. With this coal an air supply of 18 lb. would be only 58.4 per cent. excess.

The four coals cited, show the fallacy of apportioning any arbitrary weight of air to all coals alike. That there is a well-defined relation between the thermal value of any given coal and the air required is also shown. This relation is very manifest when both the air supply and the thermal value are calculated from the ultimate analyses by the usual formulas, and where the heat units are derived by calorimetric determination, only a slight difference is observable.

Ultimate analyses, however, from which to determine the chemical minimum of air, are not always available, but the growing practice of buying coal on a heat unit basis has made quite easy procuring the proximate analyses and thermal values. When the heat value of any bituminous coal is known, a very close approximation of the theoretical air requirement can be found by multiplying this value in heat units by the factor 0.00075.

A more easily remembered rule is: every 1000 B.t.u. requires ¾ lb. of air. To find the air required then, simply point off three places from the right of the number of heat units and multiply by the decimal 0.75.

For instance: the coal cited in case 1 has a heat value of 9465 B.t.u. and

$$9.465 \times 0.75 = 7.098 \text{ lb.}$$

of air, which is 0.122 lb. less than 7.22 lb. found by calculatiing from the analysis.

The coal in case 2 has 11,705 B.t.u.,

$$11.705 \times 0.75 = 8.778 \text{ lb.}$$

which compares with 8.78 lb. found by calculation from the analysis.

Similarly in case 3 the value of the coal in B.t.u. is 12,180;

$$12.180 \times 0.75 = 9.135 \text{ lb.}$$

against 9.09 lb. by the usual formula.

In case 4 the value of the coal was 15,135 B.t.u. and

$$15.135 \times 0.75 = 11.351 \text{ lb.}$$

which is only 0.009 lb. less than the weight of air found by the formula.

Although only four coals are cited, they represent widely divergent analyses, and fairly cover the entire range of bituminous coals most used. The writer has applied this approximate method for estimating the theoretical air requirements to hundreds of analyses, and nearly always has obtained results agreeing with the amounts calculated from the ultimate analyses as closely as those here shown.

Since every excess pound of air passing through the furnace reduces the efficiency and these excesses in ordinary practice will range anywhere from 50 per cent. to 200 per cent. evidently, to secure economy in furnace operation, this is the first evil to be corrected. The use of such enormous surplusage of air is due to the construction of furnaces, both hand-fired and largely also, stoker-fired.

With present constructions, the air cannot be so evenly distributed that each combustible element will receive its quantity of oxygen and no more, nor can the intimate mixture be secured of air and gases, essential to complete combustion of the gaseous distillates from the coal. No matter how great affinity the gaseous combustibles have for oxygen, no chemical union can take place unless they are in intimate contact; certainly the greater the area of contact, or the more numerous the points of contact, the more rapid and complete the chemical union.

This inability to supply just the proper amount of air in just the right place, furnishes an excuse for using such excesses.

As a remedy, the writer can only suggest that in burning a fuel of dual characteristics like bituminous coal, its combustible elements being partly gaseous and partly solid, the volatile combustibles and the fixed carbons should be burned separately, with separate air supplies, each under control.

In such a furnace the element of chance could be largely minimized, the volatile combustibles more fully utilized, and excess air reduced, perhaps, to a negligible quantity.

Mixed Pressures in Boiler Plants

By T. M. Giles

Widely varying mixed pressures should not be allowed in plant operation unless the boilers on the low-pressure side can safely withstand the higher pressure for a limited time.

It is fairly common practice where conditions warrant, to operate a boiler plant at two different pressures. Generally reducing valves are used to maintain the pressures desired in the two parts of the piping system. Sometimes reducing valves are not used, but an ordinary stop valve is placed in the steam line to separate the high- and low-pressure parts of the pipe system, and this valve opened when it is desired to operate at one pressure. Can safety valves be arranged to adequately protect the low-pressure boilers in such a plant, when the pressures carried are considerably different?

The Massachusetts Board of Boiler Rules recognized the necessity of providing for the use of mixed pressure in boiler-plant operation, and has made rules governing the safety-valve equipment for such plants, but there seems to be very little reason for adopting some of the rules. They recognize three conditions of operation: First, where all the boilers are connected to a common steam main and are allowed the same pressure. In this case the pressure allowed and the grate area of each boiler determine the size of safety valve that is to be used for that boiler. This rule is perfectly proper and open to criticism in no way.

In the second place, the board has provided a rule to govern the equipment of a plant where the boilers are connected to a common steam main and are allowed different pressures, but are to be operated at one pressure. This rule provides that the safety-valve area on each boiler shall be at least that required for the grate area under it and the pressure that is allowed (not working pressure), and it is further stated that the aggregate valve area must be at least equal to that which would be required for the aggregate grate area, the lowest or operating pressure being used in calculating this valve area. The last part of this rule is intended to prevent the use of less valve area than would be required by the lower pressure in order to adequately relieve the pressure on all the boilers and prevent an excess of pressure on the weaker boilers. This rule would be logical if it also specified that the plant could not be operated unless all the boilers were connected to the system, but such a requirement would not be practical.

Apparently the effect of the discontinuance of one or more of the boilers on the rest of them was not considered. For example, under the present rules, if a plant containing ten boilers was allowed 160 lb. pressure, and equipped with 31.5 sq.ft. of grate area each, and two boilers allowed 95 lb. and with 32.5 sq.ft. of grate surface each, the safety-valve equipment would be as follows: The part of the rule referring to the minimum size of safety valve required for each boiler, would be met by using 3-in. diameter valves on the high-pressure boilers and 3.5-in. valves on the others; but nearly 29 sq.in. of total valve area would be lacking to comply with the latter part of the rule, referring to the aggregate valve area required.

This could be remedied by placing an additional 4-in. valve on each of the low-pressure boilers. If, for any reason, one of these low-pressure boilers was cut off the line, there would be a shortage of about 10 sq.in. of valve area for the remaining boilers, if they were considered as a complete plant under this rule.

Does the closing of the stop valve on a boiler or boilers supplying the excess valve area required under this rule, constitute a violation of it? This question may seem a little harsh and fine drawn, but safety valves have been condemned in Massachusetts on account of an excess of grate area beyond that specified of as little as ½ sq.ft., while under the above assumed conditions there would be a total excess of about 30 sq.ft. of grate area, or nearly 3 sq.ft. per boiler.

The third condition under which safety valves are installed where the boilers are connected to a common steam main, is where both the operating and allowed pressures are different. In this case, the lower-pressure boilers are to be additionally protected by safety valves placed on the pipes connecting them to the steam main, and such valves are to have an area at least equal to that of the connecting pipe. This rule seems to have been adopted on the assumption that a safety valve of a given diameter would be capable of discharging all the steam capable of passing through a pipe of the same diameter. This, of course, is by no means true, as may be shown by the Massachusetts rules for the determination of the size of safety valves, which are based on Napier's formula for the weight of steam discharged from an opening of a given area in a second of time at a fixed pressure. There is assumed an arbitrary rate of evaporation per square foot of grate surface to determine the ability of a boiler to generate steam. For pressures between 150 and 200 lb., the valve areas are based on the assumption that 200 lb. of steam might be generated per hour per square foot of grate surface, and since a valve of 4 in. diameter is considered capable of caring for 56 sq.ft. of grate surface under these conditions, it is estimated that it will successfully discharge about 11,200 lb. of steam per hour, or say 186 lb. per min. Tests of well designed safety valves show that this discharge rate is about what could be reasonably expected. At a pressure of 175 lb. gage, 186 lb. of steam per minute would represent a velocity of about 5000 ft. per min. when delivered through a 4-in. pipe. Such velocity in pipe lines is considered conservative for regular operating conditions, and can be greatly exceeded if a considerable drop in pressure is allowed. It will therefore be seen that it would be an impossibility for a 4-in. safety valve to discharge all the steam that could be transmitted through a 4-in. pipe.

To see how the safety of a plant complying with the Massachusetts law might be effected, assume a case of widely different pressures, where a small horizontal tubular boiler is used to furnish steam for heating, or other purposes requiring low pressure, in a manufacturing establishment. Assume that this boiler has seen its best days for other uses, and is now regarded as being safe for only 50 lb. pressure, which is ample for the work required. While this boiler is sufficiently large to care for the low-pressure work most of the time it frequently requires some assistance, and is therefore conneted to the high-pressure power boilers through a reducing valve. The regular boiler-plant equipment is composed of eight 500-hp. boilers, operating at 175 lb. pressure, so there is a difference of 125 lb. in the steam pressure carried on the two portions of the system. Assume that the low-pressure boiler is connected to the steam main by 20 ft. of 4-in. pipe which has placed on it a 4-in. safety valve, as required by the Massachusetts rule; also assume that the safety valve on the boiler is capable of caring for the steam generated by the boiler alone, and that both ends of the plant are working at rated capacity when the reducing valve becomes inoperative and allows the high-pressure steam to flow into the low-pressure side of the piping system. What chance is there under these conditions that the pressure on the heating boiler will not exceed 150 lb.? The eight 500-hp. boilers will contain about 384,000 lb. of water at a temperature of 377 deg. F., and assume that the pressure is to drop

15 lb. This would give a final temperature of 370 deg., or a drop of 7 deg. With the quantity of water given this drop would represent the liberation of about 2,680,000 B.t.u. or would produce the evaporation of 2250 lb. of water. A 4-in. diameter safety valve would require about 12 min. to discharge this amount of steam. A drop of 15 lb. from the initial entering pressure through the 20 ft. of 4-in. pipe connecting the low-pressure boiler with the steam main, would cause a flow through this pipe of about 770 lb. per min., or over four times the amount that the safety valve would be capable of discharging.

It should be remembered that the total flow of steam that has been considered is produced by stored energy in the heated water which is liberated by the reduction in pressure, and the furnace heat is left to supply the regular draft of steam to be used for operating the plant. It would seem that the explosion of the low-pressure boiler under the assumed conditions would be inevitable, unless it successfully withstood about three times the pressure that was considered safe for it. Apparently mixed pressures should not be allowed in plant operation, unless the boilers on the low-pressure side are considered safe to withstand the pressure allowed on the high-pressure side for a limited time. The fact that widely varying pressures are seldom used is perhaps one reason that the use of mixed pressures has not more often caused serious accidents in the boiler plant.

Procedure in Coal Specifications

By Mervin K. Baer

Pro rata corrections for heat values and deductions for excess volatile matter and ash. A convenient tabulation for comparison of bids.

That the great majority of consumers of fairly large quantities of coal purchase their fuel under specifications is well known, but how the various bids are tabulated so that they may be readily compared with each other is not generally understood. There are no hard-and-fast rules for writing specifications; they will vary with each individual consumer, according to the conditions which obtain. However, all such specifications take one of several general forms, the trend of which may be more easily followed by taking up a definite example under each class.

The board of education of a certain southern Michigan city recently bought coal under a contract which illustrated one type. The specifications call for "2000 tons of New River Sewell seam smokeless bituminous coal, mined in West Virginia, run-of-mine size (50 per cent. over pea size)." The standard upon which all computations are to be based is arbitrarily chosen in this case as follows: Heat value, 15,000 B.t.u.; volatile matter, including moisture, 22 per cent.; ash, 5 per cent.; these percentages being upon the basis of air-dried coal. No bids are to be accepted upon coals containing less than 14,250 B.t.u.; containing more than 25 per cent. volatile matter, including moisture; or containing enough sulphur to produce clinkers or give other unsatisfactory results. The correction factors to be used in arriving at the comparative value of the coal are given as follows:

1. A direct proportionality between the heat values.

2. Five per cent. of the price for each 1 per cent. difference in volatile matter, including moisture.

3. Two cents per ton for each 1 per cent. difference in ash content.

A set of specifications of this nature places the burden of the computations entirely upon the bidder; he must reduce his coal to the arbitrary standard set up by the specifications, and quote his price on this standard rather than upon his coal. Since all bids are on the basis of this standard coal, the purchaser can by inspection pick out the cheapest bid, or the bid he is willing to accept. It remains for him, however, to have an analysis of the fuel made upon its delivery, from which to figure the actual price to be paid, as seldom does the coal delivered have the same composition as the contract standard.

Suppose the analysis shows that the coal actually has the following composition: Volatile matter, including moisture, 23 per cent.; ash, 6 per cent.; heat value, 14,500 B.t.u.; and suppose the contract price to have been $3 per ton. The purchaser would arrive at the price he would have to pay in the following manner:

Correction for heat value:

$$\frac{14,500}{15,000} \times \$3 = \$2.90$$

Correction for volatile matter:

$$23 - 22 = 1 \text{ per cent.}$$
$$\$2.90 \times 0.95 = \$2.755$$

Correction for ash:

$$6 - 5 = 1 \text{ per cent.}$$
$$\$2.755 - 0.02 = \$2.735$$

the cost of the coal per ton.

Adopting a standard set of values is not very common at present. The more frequent practice is merely to set an upper limit for the various constituents, and have the bidders submit the prices and analyses of coals which will fall within these limits. The purchaser then tabulates the bids to have a comparative basis upon which to work.

This method is illustrated by the manner in which coal is purchased for a certain court house and post-office building. Bids are called for on 8500 long tons of Illinois Nos. 1 and 2 washed nut, whose upper limits in per cent. of dry coal are: Ash, 10 per cent.; volatile matter, 38 per cent.; sulphur, 2.5 per cent. The corrections in price are:

1. A *pro rata* correction for the heat value.

2. A 2c. per ton correction for each 1 per cent. difference in ash.

With this sort of a specification, the purchaser, upon opening the bids, constructs Table 1.

TABLE 1. TABULATION OF BIDS

Bid	(1) B.t.u. in coal as received	(2) Ash in dry coal per cent.	(3) Price per ton, Dollars: Bid	(4) Price per ton, Dollars: Plus ash difference	(5) Cost in cts. per million B.t.u.
A...	11,400	10	3.02	3.100	12.140
B...	14,800	6	4.10	4.100	12.367
C...	11,350	10	2.80	2.880	11.328
D...	12,136	8.9	3.04	3.098	11.396

The first three columns can be filled out directly, since these items are all specified in the bids. To compute column four, that coal which has the lowest ash content is taken as a basis, so that the corrections on all the other bids will be of the same sign. Thus, coal "B" is used as the standard; consequently it has no correction for ash. Coal "A," on the other hand, has an excess of ash over coal "B" of 4 per cent.; hence there must be added

$$4 \times 0.02 = \$0.08$$

per ton to the price of coal "A."

In column five, the corrections for heat values are made. Instead of using any one coal as a basis, however, the prices of all the coals are reduced to cost in cents per 1,000,000 B.t.u.:

$$\text{Cost per 1,000,000 B.t.u.} = \frac{\text{price per ton} \times 1,000,000}{\text{B.t.u. per lb.} \times 2240}$$

For coal "A," this amounts to

$$\frac{310 \times 1,000,000}{11,400 \times 2240} = 12.14 \text{ cents}$$

The last column reveals the comparative costs of the fuels: Coal "C" is the cheapest, at 11.328c. per 1,000,000 B.t.u.

Recently the method used in these two cases of correcting for ash differences has been much discussed, the consensus of opinion being that the price of the coal should not vary at a uniform rate throughout the range of its ash content, since coal containing 40 per cent. ash has no value for steam generation. In view of this, certain large consumers have adopted a general form for their ash specifications which is entirely different from those previously mentioned. The standard is a certain percentage of ash (depending upon the kind of coal required). For each 1 per cent. of ash less than this amount, a bonus of 2c. per ton is paid; for ash in excess of this standard percentage, the following penalties are deducted:

TABLE 2. PENALTIES FOR ASH

1 per cent. excess	No penalty,
2 per cent. excess	No penalty
3 per cent. excess	2 cents per ton,
4 per cent. excess	4 cents per ton
5 per cent. excess	7 cents per ton
6 per cent. excess	12 cents per ton,
7 per cent. excess	18 cents per ton,
8 per cent. excess	25 cents per ton,
9 per cent. excess	35 cents per ton,

In addition to this ash correction, they use the *pro rata* correction for heat values.

It would be impracticable to cover all the different varieties of coal specifications now being used. The foregoing examples are given because they are thought illustrative of good practice in this rapidly grow' field of specifications for the purchase of coal, and because they also show that the work involved is not nearly as formidable as the title might imply.

Boiler Efficiency

The efficiency of a boiler is the ratio of the amount of heat which it absorbs and uses in making steam to the amount of heat furnished to it.

The heat which is furnished to the grate is all the heat contained in all the coal as fired.

But the heat furnished to the furnace and boiler is that in the coal which is *burned.* If a part of the good coal drops through the grate the boiler and furnace should not be charged with its heat.

The Committee on Tests of the American Society of Mechanical Engineers, therefore, recommends the consideration of two efficiencies.

The efficiency of the grate would equal the first of these formulas divided by the second or

$$\frac{\textit{Lb. combustible}\ \textbf{burned}}{\textit{Lb. combustible}\ \textbf{fired}}$$

Suppose a boiler burned 482 lb. of coal and evaporated 3856 lb. of water from feed of 130 deg. into dry-saturated steam of 150 lb. absolute pressure. The coal had a calorific value of 13,050 B.t.u. per lb. and 10 per cent. of ash, and there were 82 lb. of refuse taken out of the ashpit. What are the efficiencies of the combined boiler, furnace and grate; of the boiler and furnace; and of the grate alone?

To make one pound of steam of 150-lb. pressure from feed water of 32 deg. would take 1193.4 B.t.u. To raise a pound of water from 32 deg. to 130 deg., the temperature of the feed water requires 97.89 B.t.u. To make a pound of water already at 130 deg. into dry-saturated steam at 150 lb. would therefore require

$$1193.4 - 97.89 = 1095.51\ \textit{B.t.u.}$$

There are $3856 \div 482 = 8$ lb. of water evaporated per pound of coal, and since 1095.51 B.t.u. are absorbed for each pound of steam made, there are $1095.51 \times 8 = 8764.08$ B.t.u. absorbed for each pound of coal fired. The calorific value of 1 lb. of coal is 13,050 B.t.u. Then the efficiency of boiler furnace and grate is

$$\frac{\textit{Heat absorbed per lb. of coal fired}}{\textit{calorific value of 1 lb. of coal}} = \frac{8764.08}{13{,}050} = 67.16\ \textit{per cent.}$$

The portion not taken out of the ashpit was presumably combustible and burned. This is the difference between the amount of coal as fired and the amount of ashpit refuse, or

$$482 - 82 = 400\ \textit{lb.}$$

The water evaporated per lb. of combustible burned was therefore

$$3856 \div 400 = 9.64\ \textit{lb.}$$

And the heat absorbed per pound of combustible burned is

$$1095.51 \times 9.64 = 10{,}560.7+\ \textit{B.t.u.}$$

One pound of coal contains 13,050 B.t.u. Of this coal 10 per cent. or $\frac{1}{10}$ is ash. The other $\frac{9}{10}$ is combustible, and must contain all the heat units. If nine-tenths contain 13,050 B.t.u., one-tenth would contain $\frac{13{,}050}{9}$ and $\frac{10}{10}$ or a whole pound $\frac{13{,}050 \times 10}{9} = 14{,}500$ B.t.u.

$$\textit{Efficiency of boiler, furnace and grate} = \frac{\textit{heat absorbed per lb. of coal}\ \textbf{fired}}{\textit{calorific value of 1 lb. of coal}} = \frac{\textit{heat absorbed per lb. of combustible}\ \textbf{fired}}{\textit{calorific value of 1 lb. of combustible}}$$

$$\textit{Efficiency of boiler and furnace} = \frac{\textit{heat absorbed per lb. of combustible}\ \textbf{burned}}{\textit{calorific value of 1 lb. of combustible}}$$

Then the efficiency of the boiler and furnace exclusive of the grate is

$$\frac{\textit{Heat absorbed per lb. of combustible burned}}{\textit{calorific value of 1 lb. of combustible}} = \frac{10{,}650.7164}{14{,}500} = 73.45\ \textit{per cent.}$$

The number of pounds of combustible fired was $0.90 \times 482 = 4338$. The number of pounds of combustible burned was $482 - 82 = 400$. The efficiency of the grate was

$$\frac{400}{433.8} = 92.2\ \textit{per cent.}$$

And the product of the efficiency of the boiler and furnace and that of the grate equals

$$\frac{10{,}650.7164}{14{,}500} \times \frac{400}{433.8} = 67.16\ \textit{per cent.}$$

as before determined.

Boiler Explosion at Keene, N. H.

The accompanying photograph shows the results of a boiler explosion on Dec. 5 at the plant of the Keene Glue Co. Fortunately there was no loss of life,

RESULTS OF BOILER EXPLOSION AT KEENE, N. H.

but considerable property damage resulted, as is evident in the photograph.

The boiler was a return-tubular, 16 ft. long, 66 in. in diameter and was made up of ⅜-in. plate. It was 20 years old and at the time of the explosion was carrying only 40 lb. pressure, its principal duty being to supply steam to the kettles and for other purposes in the factory.

The rupture of the boiler was caused through a lap crack which extended under the lap of the longitudinal seam for the full length of the middle course, then continued around the girth seams at the end of this course. The plate tore along the rivets for about one-third the circumference of the boiler and then tore through the solid plate diagonally from seam to seam. The shell plating was free from corrosion or pitting.

The boiler was insured and according to report had been inspected twice within the past nine months.

NEW PUBLICATIONS

HYGIENE FOR THE WORKER. By William H. Tolman and Adelaide Wood Guthrie. American Book Co., New York City. Cloth, 7½x5¼ in.; 231 pages; illustrated.

While this book is primarily designed for boys and girls from 13 to 18 years old, it is also adapted to the needs of all workers, old and young. Its purposes are twofold: "to equip the worker to care for himself under actual working conditions and to add to his happiness and efficiency."

The hygiene of the workroom, its proper conditions and ventilation, is carefully treated by the authors and many helpful suggestions are offered. Occupational dangers are also considered and recommendations made for the employee's protection. "First Aid to the Injured" is a valuable chapter, and "What the Worker Has a Right to Expect" makes clear the laws regarding hours of labor, what provisions are necessary for safeguarding the employee's rights and the sanitary regulations the state insists shall be carried out. There is also an interesting chapter on tuberculosis, how it develops and what precautions should be taken against contagion. The book is written in simple English and may be read with profit.

ANALYSIS OF METALLURGICAL AND ENGINEERING MATERIALS. By Henry Wysor, B. S. The Chemical Publishing Co., Easton, Penn. 10½x8 in., 82 pages, 9 illustrations, 2 tables. Price, $2.

The purpose of this manual is to furnish the student and chemist "the most practical methods compatible with accuracy and authoritative recognition for the analysis of such materials as are commonly brought to chemists in the field of metallurgical and engineering chemistry." Opposite each page of text is a blank page for the insertion of notes, equations, and the interpretation of results relative to the work to which the text refers.

Directions are given for the care of laboratory apparatus and the orderly arrangement of the chemicals. The book is devoted to practical direction for the sampling and analysis of fuels, gases, lubricating oils, ores of many kinds and the chief industrial metals.

The working chemist will find this book a valuable addition to his laboratory equipment.

The Elektrotechnischer Verein in Wien, the leading professional electrical society of Austria, announces the early appearance of a work which under the title "Statistik der Elektrizitätswerke in Österreich" will give a list of all the electric stations in Austria, Bosnia and Herzeogovina, with the names and addresses of the owners, the builders and furnishers of apparatus, number of inhabitants of the district served, and data regarding the operation, load, rates, etc. The work is published as one of the activities of the society and is sold practically at cost which is 4 kronen (eighty-odd cents) plus the postage. The address of the society is 12 Theobaldgasse, Vienna, Austria.

BOOKS RECEIVED

SOLDERING AND BRAZING. By James F. Hobart. D. Van Nostrand Co., New York. Cloth; 190 pages, 4½x7 in.; 62 illustrations. Price, $1.

PRIMARY BATTERY IGNITION. By C. Wadsworth, Jr. D. Van Nostrand Co., New York. Cloth; 78 pages, 4¾x7 in.; 26 illustrations. Price, 50c.

STEAM ECONOMY IN THE SUGAR FACTORY. By Karl Abraham; translated by E. J. Bayle. John Wiley & Sons, New York. Cloth; 5¼x7¾ in.; 104 pages; tables.

ALTERNATING-CURRENT MACHINERY. By William Esty. American School of Correspondence, Chicago, Ill. Cloth; 7¼x9¾ in.; 467 pages; 410 illustrations; tables and formulas.

SOCIETY NOTES

The semiannual meeting of the American Institute of Steam Boiler Inspectors, of Boston, began its season's educational work on Dec. 16, 1912. A dinner followed the meeting, the speakers being J. M. Winter, president American Institute of Steam Boiler Inspectors of New York City; John A. Stevens, chairman Massachusetts Board of Boiler Rules, and P. H. Hogan, past national president N. A. S. E. The organization, now in its third year, will continue to hold monthly meetings during the winter, at which papers and addresses of practical value to boiler inspectors will be presented. Social and educational advancement is the aim of the organization. Membership of boiler inspectors employed by the federal government, by state, cities and insurance companies is solicited. Communications should be addressed to E. R. Doherty, 31 Milk St., Boston, Mass., care of Mutual Boiler Insurance Co. of Boston.

Charleston Convention of Marine Engineers

The convention of the Marine Engineers' Beneficial Association will convene at Charleston, S. C., on Jan. 20, 1913, with headquarters at the Charleston Hotel. It is expected that about a hundred delegates will attend.

The local committee has its arrangements well in hand for the comfort and entertainment of the visiting engineers and their ladies, and promises to give them a hearty Southern welcome. The program will include a ball and reception, an oyster roast, a sail along the harbor and visits to near-by places of interest. A smoker under the auspices of the National Marine Engineers' Supply Men's Association, to be held in German Artillery Hall, on Wentworth St. will be a big feature, to which all prominent engineers and city officials will be invited.

The New York City delegation will leave for Charleston on Saturday, Jan. 18, by the Pennsylvania R.R., and every effort is being made to insure a full attendance.

OBITUARY

George B. Carpenter, founder and head of the railroad, mill and contractors' supply house which bears his name, died at his home in Park Ridge, Chicago, on Dec. 11. Mr. Carpenter was born in Conneaut, Ohio, in 1834 and moved to Chicago with his parents in 1850. In 1857 he acquired an interest in the concern of Gilbert Hubbard & Co. and in 1861, following the death of Mr. Hubbard, he obtained full control and reorganized the business under its present name. After the death of his wife in 1905 he retired from active business, although he retained the presidency of his company and continued as a director of the Great Lakes Supply Co. Mr. Carpenter is survived by four sons, Judge George A. Carpenter, and Benjamin, Hubbard F. and John A. Carpenter. The three last have been in active charge since their father's retirement.

PERSONAL

John J. Swan, formerly with the Chicago Pneumatic Tool Co., New York, as manager of the New York office and later as mechanical engineer in the compressor department, has become associated with the Oil Power Engineering Corporation, New York.

James F. Hobart, familiar to the mechanical reader as contributor to various technical journals, has recently resigned his position as inventor and designer in the research department of the Diamond Match Co., of Barberton, Ohio, to become superintendent of the Hanna-Breckenridge Co., Fort Wayne, Ind.

Moments with the Ad. Editor

Some men are superstitious enough to believe that '13 will, perforce, be an unlucky year.

It probably will be—for them.

We usually get things that are in harmony with our minds. If we conclude that a straight line and a wiggley one following 19 is going to send us to the Limbo of the unlucky, we'll get there no doubt.

For one mellow note from a flute there will be a dozen replies, some full and near, some vague and dream-like, but all in the identical key first ringing from the flute.

This principle, which is one of the most fundamental truths in music, is also one of the greatest and most valuable truths in life.

Since Nature controls the law of the echo business, you will get a tone in harmony with the one you send out.

If we are hopeful and optimistic, 1913 will bring us our share of its reward. If we persist in a belief that there is no hope in life and a bottomless pit for its end, we shall probably find no hope and a pit.

Following out the idea, if you strike a certain note in life—or in business—if the note is distinct, you are simply bound to get a response from every human being in key with that note.

In advertising, if you strike a note of sincerity, sincere people will listen to you. If the tone that runs through your business talk is genuine and honest, it is going to repeat itself in every genuine mind it reaches.

And the same rule holds when the tone sounded is false or tawdry or affected, for the first to respond will be the superficial and the veneered. They are the people invited, why shouldn't they come? People have realized this in a vague, instinctive way for ages but they haven't thought it out much.

The whole intention of modern business is to attract a favorable audience, to win the ear of the public, to establish good will between the world that buys and the world that sells.

Advertising is the main channel through which this is accomplished. No man can afford to float his business enterprize on any little side stream. And Advertising is the good, weather beaten old navigator who is willing and able to bring the bark to port.

Since advertising plays so important a part in the world's business the quality of it is a very important consideration.

In the first place, advertising was nothing more than a direct, honest foreword to the public. Then it was abused as every good thing is. The cheap and superficial, the insincere and the dishonest crept in. They sounded a morbid note—with its own answer—not a widely satisfactory one.

Folks found that a permanent business couldn't be built on anything but a fundamentally honest product.

And so the new school of advertising and publishing came into being—founded on the principle that there is no more potent argument for a good article than the truth about it and dedicated to the belief that the interests of subscribers must be considered above all things else.

1913 will see new things in advertising, greater progress, more intelligent copy, closer adhering to the square deal.

Incidentally, it will see the ad. editor located on page two, nearer to the beginning of things. We suspect our good friends will think enough of us to look us up at the new address every week. And you'll find us "at home."

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